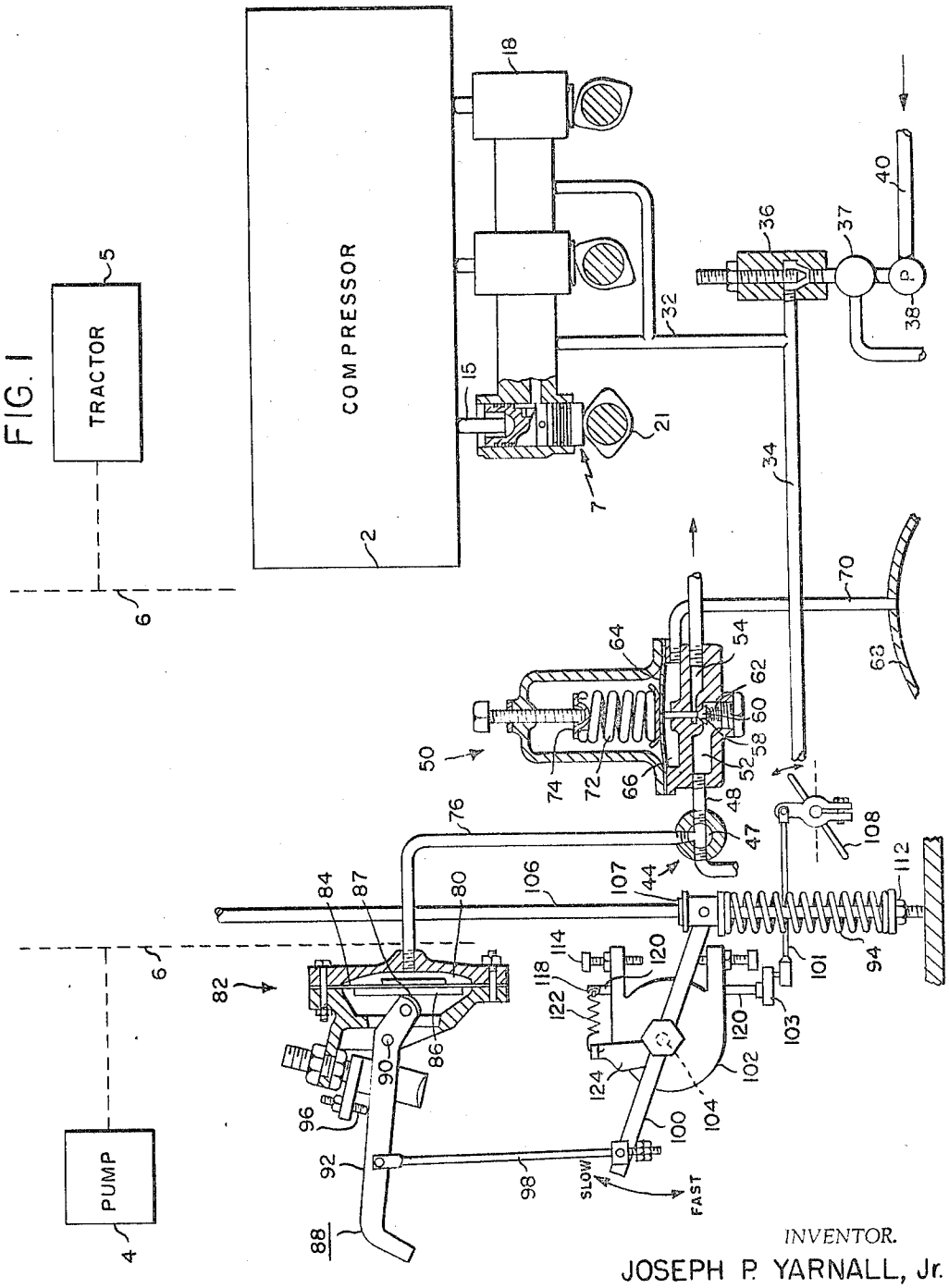
Dec. 13, 1966 J. P. YARNALL, JR 3,291,378

COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 1

FIG. 1



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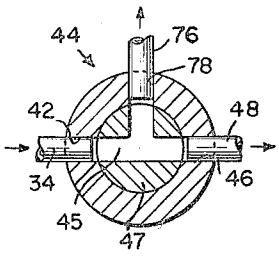


FIG. 2.

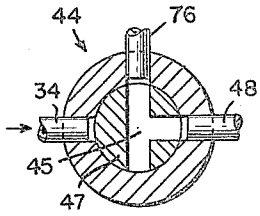


FIG. 3.

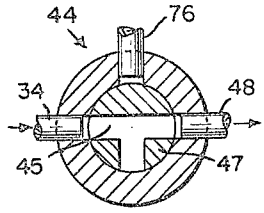


FIG. 4.

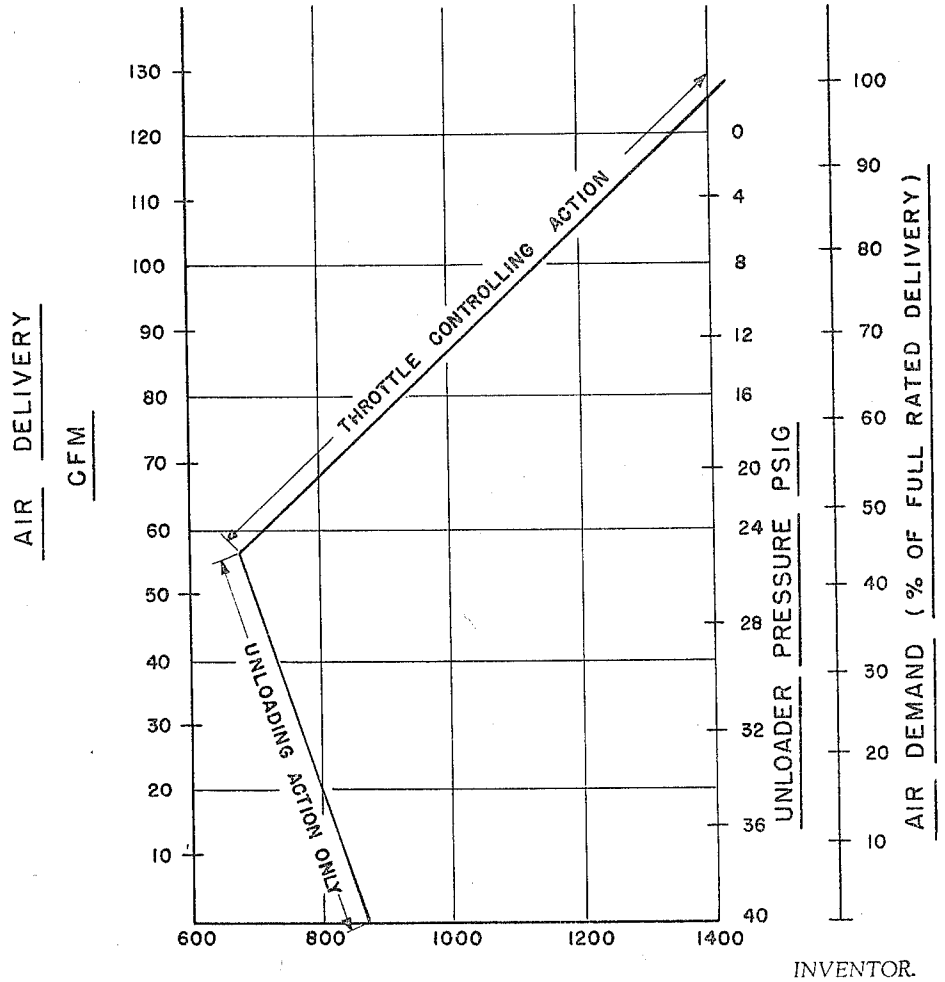


FIG. 5.

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Dec. 13, 1966 J. P. YARNALL, JR 3,291,378

COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 3

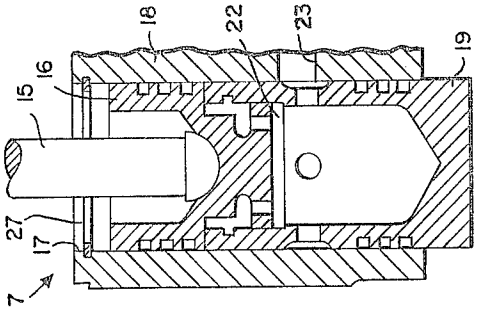


FIG. 7.

FIG. 6.

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COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 4

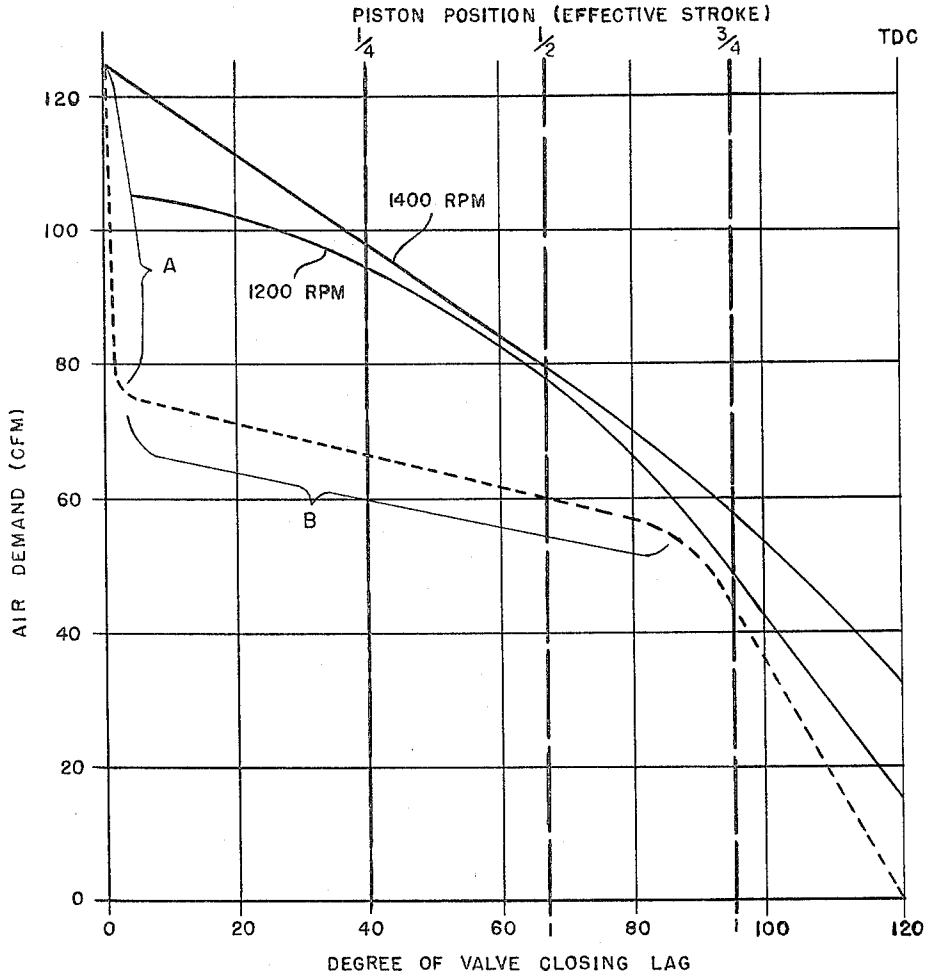


FIG. 10.

RECEIVER PRESS (PSI)	RPM	AIR DEMAND (CFM)	UNLOADER PRESS (PSI)	DEGREES OF VALVE LAG
100	1200	20	55	115
"	"	35	52	105
"	"	55	49	90
"	"	75	40	70
"	"	105	12	0
"	1400	20	58	125
"	"	35	55	120
"	"	55	52	105
"	"	75	40	80
"	"	105	35	30
"	"	125	15	0

FIG. 8.

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COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 5

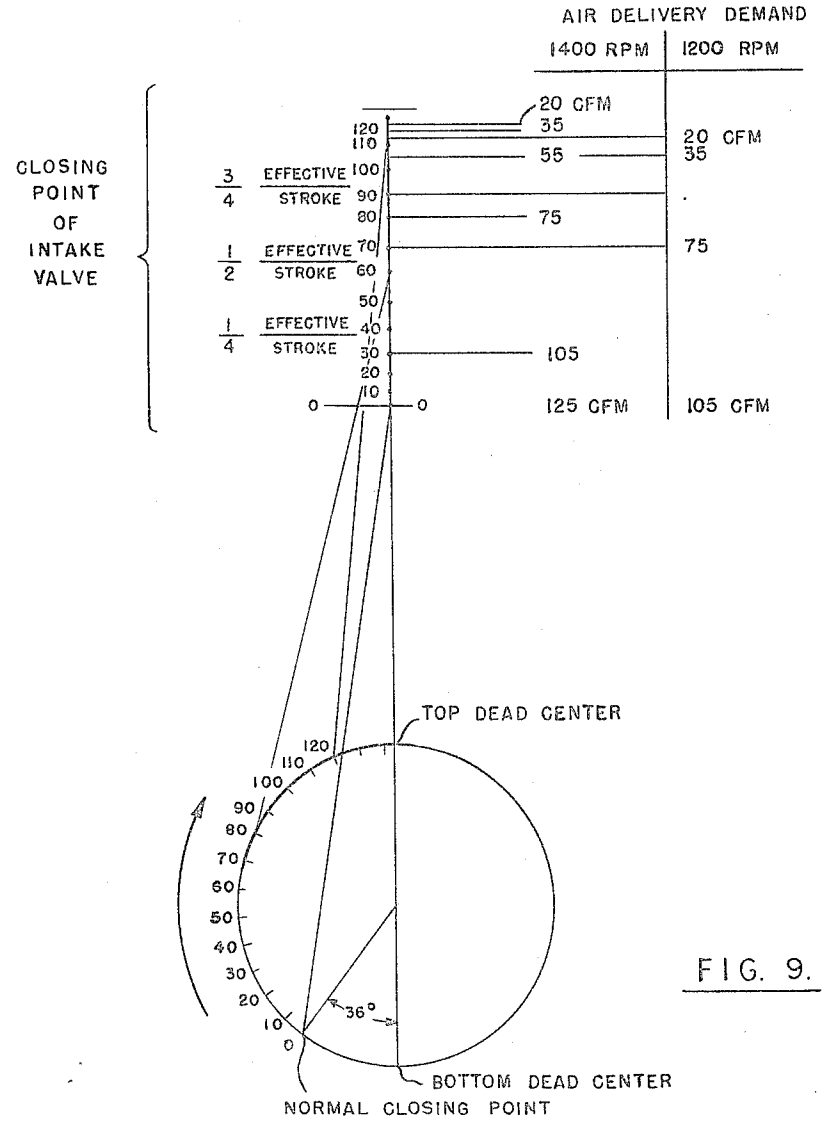


FIG. 9.

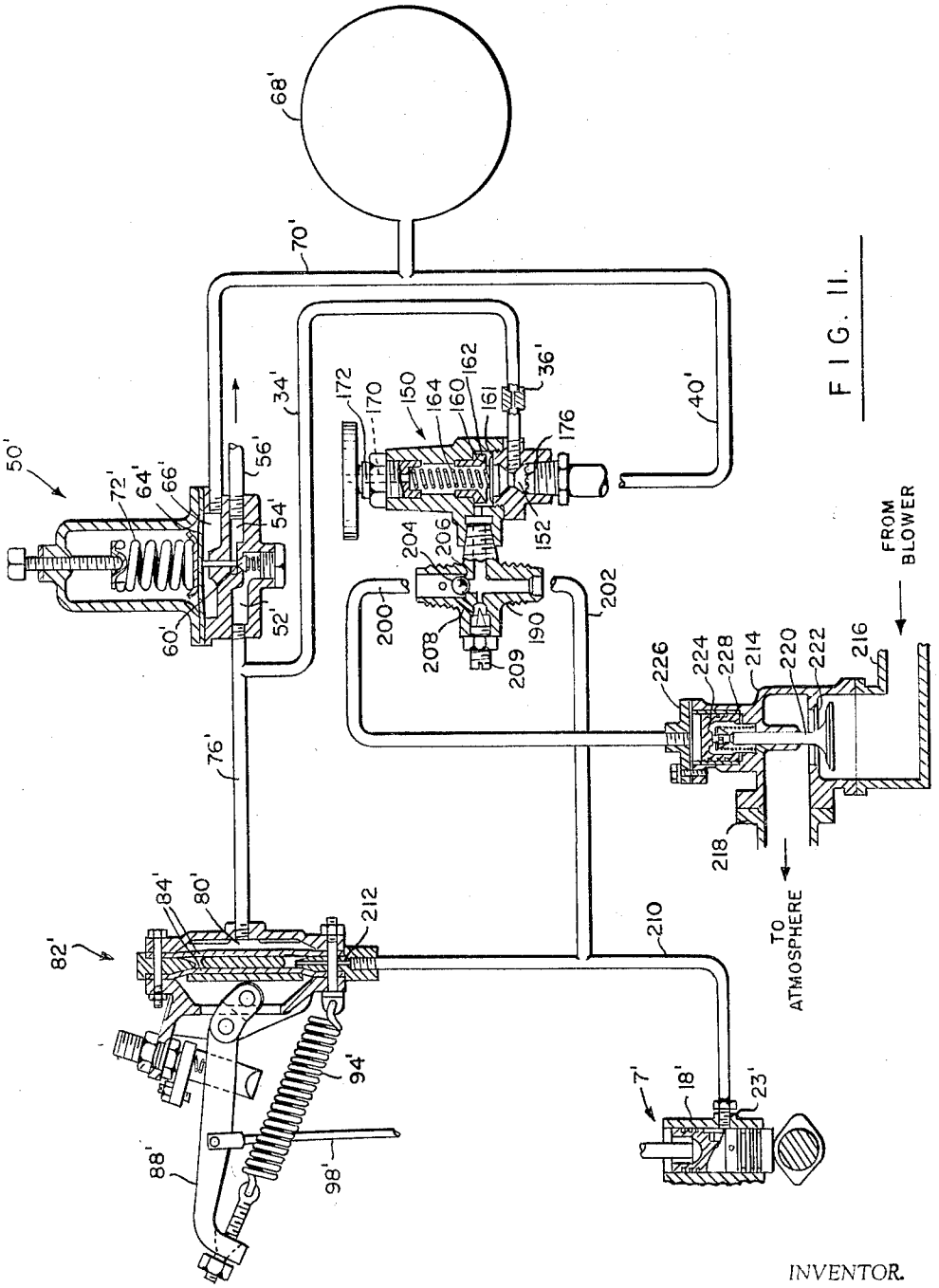
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COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 6



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Dec. 13, 1966 J. P. YARNALL, JR 3,291,378

COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 7

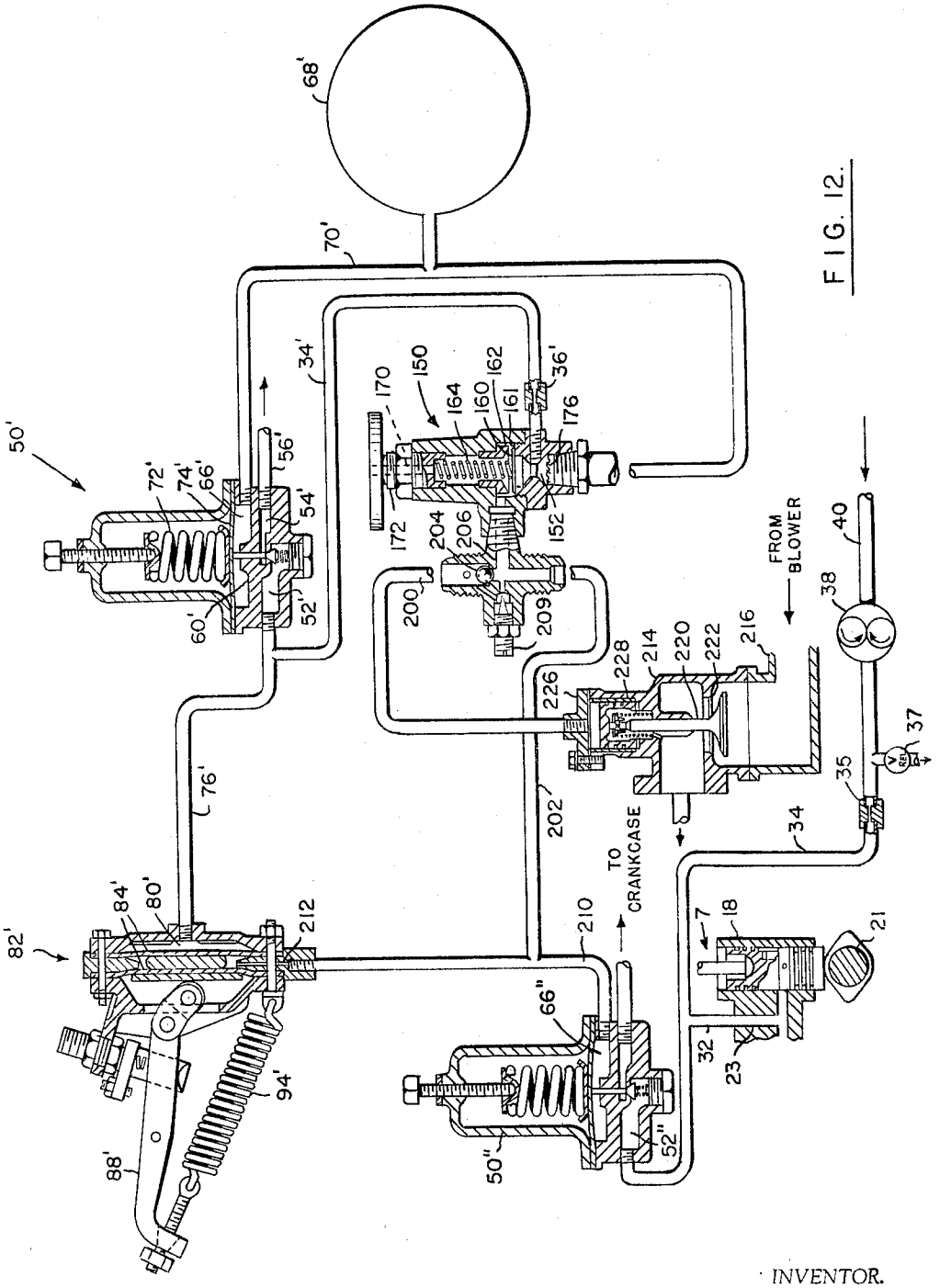


FIG. 12.

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COMPRESSOR CONTROL SYSTEM

Original Filed March 1, 1963 8 Sheets-Sheet 8

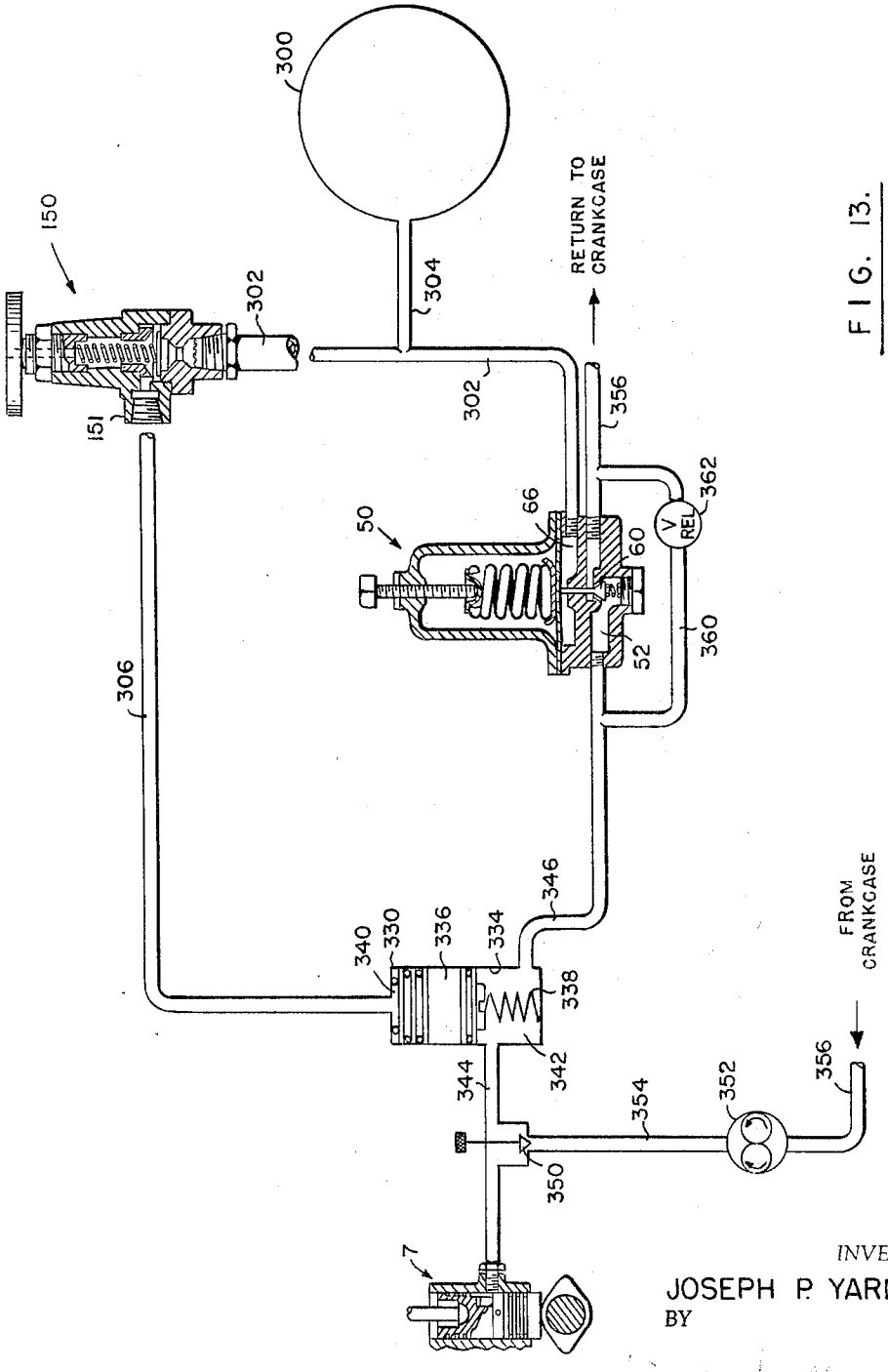


FIG. 13.

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United States Patent Office

3,291,378

Patented Dec. 13, 1966

3,291,378

COMPRESSOR CONTROL SYSTEM

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Continuation of application Ser. No. 262,112, Mar. 1, 1963. This application Mar. 8, 1965, Ser. No. 437,906

17 Claims. (Cl. 230—9)

This application is in part a continuation of my application, Serial No. 262,112, filed March 1, 1963, now abandoned.

This invention relates to a control system for compressors and has particular reference to a control system wherein the delivery of a reciprocatory air compressor is controlled by partial unloading of the compressor cylinders in accordance with the demand for air.

Although the control system in accordance with this invention will be disclosed as incorporated in a combined compressor-tractor unit wherein an internal combustion engine drives both an air compressor which delivers compressed air to a receiver and a pump for hydraulically powered equipment and in other engine-compressor combinations, it will be apparent that the invention has broader application. For example, the invention is broadly applicable to the control of compressors alone. Moreover, the compressor may be driven by an engine, a motor or other suitable drive means.

One type of compressor delivery control system in use today involves a technique known as step-unloading. This system involves a selector valve which is set for the continuous unloading of one or more cylinders. In accordance with this invention, the load-unload action of these prior systems is eliminated and there is achieved an automatic matching of delivery to demand throughout the entire range from 0 to 100% of full rated delivery. In the prior systems, such matching of delivery to demand is only from full rated delivery to 60% of full rated delivery.

Another object of the invention is to provide a compressor delivery control system of the indicated type in which there is a reduction in the air receiver pressure change needed to cause a receiver pressure operated speed control device to effect a reduction in engine and compressor speed. Briefly stated, this is achieved by an arrangement whereby a pressure change in the receiver is amplified when applied to speed control device.

In accordance with this invention the air delivery is controlled by only partially unloading the compressor cylinders in accordance with the demand for air. Briefly stated, this is achieved by regulating the pressure in the unloaders in proportion to the receiver pressure. The control system in accordance with this invention will control the compressor to match the air demand automatically throughout the entire range of the rated delivery of the compressor for any given speed. Moreover, the system in accordance with this invention automatically matches compressor delivery to demand continuously and without loading and unloading even at high compressor speeds where normal delivery would far exceed the demand.

Another feature of this invention is the use of lubricating oil under pressure for actuating the unloaders. Hydraulic unloading involves less noise because the oil cushions and dampens the unloader movements.

The above and other objects and features of the invention will become apparent from the following description read in conjunction with the accompanying drawings, in which:

FIGURE 1 is a diagrammatic view of the control system in accordance with this invention illustrating certain mechanical parts in section;

FIGURES 2, 3 and 4 are fragmentary views showing a selector valve in various operative positions;

FIGURE 5 is a performance curve illustrative of the operation of the control system disclosed herein in which the volume of air delivery and the unloader pressure are plotted against the engine speed;

FIGURE 6 is a sectional view of the compressor valves and unloaders;

FIGURE 7 is a sectional view of an unloader;

FIGURES 8, 9 and 10 are views illustrating performance tests of control systems in accordance with the invention; and

FIGURES 11, 12 and 13 are diagrammatic views of other forms of control systems in accordance with the invention illustrating certain mechanical parts in section.

Referring to FIGURE 1, a compressor indicated at **2**, a hydraulic pump indicated at **4** and a conventional tractor **5** are driven by an internal combustion engine indicated at **6** through suitable transmissions. Unloading apparatus **7** for the compressor is illustrated in FIGURES 6 and 7 as comprising an unloading mechanism substantially of the type disclosed in Gustafson Patent No. 2,667,299, to which reference is made for details of construction, unloading being achieved by opening the intake valve or valves **8** which control the entry of air from an intake manifold **10**.

For simplicity of description the unloading will be described in detail with respect to a single cylinder, it being understood, of course, that duplicate arrangements of the valve assemblies will be provided in conjunction with all of the cylinders if all of the cylinders of a particular unit are compressor cylinders, or will be applied to some of the cylinders which are utilized for compression in a unit which is in part engine and in part compressor.

As shown in FIGURE 6, there is provided the usual cylinder **9** in which the piston **11** reciprocates, the inlet valve **8** being biased closed by a spring **12**. A shaft **13** carries a rocker arm **14** which engages the upper end of the valve stem and the upper end of the push rod **15**. The lower end of the push rod **15** is received in a socket in a piston **16** slidable within a cylinder **17** formed in an unloading bracket **18**, there being provided an unloader cylinder and accompanying unloading mechanism for each compressor cylinder to be unloading. Three unloaders are shown in FIGURE 6, each being constructed as shown in detail in FIGURE 7. The piston **16** of each unloader cylinder is normally acted upon by a cam follower piston **19** engageable by a cam **21** on a conventional cam shaft **20**. The arrangement of the cams **21** on the cam shaft **20** in the case of a typical engine-compressor unit comprising three engine cylinders and three compressor cylinders is shown in FIGURE 6, the direction of rotation of the cam shaft being shown by the arrows. The arrangement is in accordance with conventional practice. It will be noted that the double lobed cams **21** are positioned angularly 120° apart whereby for each rotation of the cam shaft, each inlet valve is opened twice, one inlet valve being opened each 60° of rotation.

Under operating conditions during which the compressor is loaded, the upper end of piston **19** engages piston **16**, as shown in FIGURE 7 so that these pistons **16** and **19** operate as a unit under the action of the cam **21** to produce normal movements of the inlet valve **8**. The unloader mechanism comprises a chamber **22** between the pistons **16** and **19**, this chamber **22** being arranged to receive oil through passageways **23** in bracket **18**. Oil is introduced into chamber **22** through passageways **23** when unloading is to be accomplished. The admission of oil into the chamber **22** lifts the piston **16** away from the piston **19** to thereby elongate the connection between the cam **21** and the valve **8** with the result that the valve **8** continues to reciprocate under the action of the cam **21**

but its range of movement is increased for its normal operating range so that, for example, during complete unloading, it remains open at all times. A snap ring **27** is mounted in the upper end of cylinder **17** to limit the upward movement of piston **16**. Valve **8** is also opened to produce partial unloading in proportion to the pressure of the oil in passageways **23** in a manner to be described in detail hereafter.

The discharge valve assembly is a low lift, spring cushioned, disc type valve located in the cylinder head above each piston and retained by individual valve caps. The disc valve **24** seats on a seat **25** in the cylinder head and opens when the air pressure in the cylinder becomes sufficient to overcome the combined pressure of the air above the valve disc **24** and that of the valve spring **26**. The discharge valve assembly is the same as that shown in said Patent No. 2,667,299 to which reference is made for details of construction and operation.

Passageways **23** have a line **32** connected thereto which line communicates with another line **34** connected to the output of a needle valve **36**. The inlet of the needle valve receives oil delivered from a pump **38** which receives oil through a line **40** connected to the crankcase of the engine **6**. The flow restricting orifice of needle valve **36** is adjusted to provide a flow just enough to get steady regulation of the unloaders without any surging. While the unloaders should be moved gradually, the valve **36** should not provide a flow opening so small that the compressor unloads too slowly on a rapid pump up. The pump **38** is provided with a conventional bypass or pressure relief valve **37** to maintain a desired output pressure.

The line **34** from the needle valve **36** is connected to an inlet port **42** of a three-way selector valve **44** having a T-shaped chamber **45** in a valve member **47**. Another port **46** of the selector valve **44** has one end of a line **48** connected thereto, the other end of line **48** being connected to the inlet of a pilot regulator valve **50**.

The pilot regulator valve **50** comprises an inlet chamber **52** connected to the selector valve **44** by line **48**, an outlet chamber **54** connected to the engine crankcase by a line **56**, a valve seat **58** defining a passageway between said inlet and outlet chambers, and a valve member **60** cooperable with the valve seat **58** to regulate flow through the valve **50**. The valve member **60** is guided for axial movement in the valve housing. A spring **62** biases valve member **60** upwardly so that the upper end of the valve stem contacts a diaphragm **64** whereby valve member **60** is movable along with diaphragm **64**. The diaphragm **64** defines a portion of a control chamber **66** which is connected to the compressed air receiver **68** through a line **70**. A pilot spring **72** is mounted in compression between the upper side of the diaphragm **64** and an adjustable spring retainer **74**. The pilot spring **72** acts through the diaphragm **64** to bias the valve member **60** to its open position. The valve member **60** is actuated toward its closed position in response to pressure in the control chamber **66** which overcomes the bias of spring **72**.

A line **76** connects a port **78** of the selector valve **44** to the control chamber **80** of a pressure responsive control device **82**. A diaphragm **84** is arranged to be responsive to the pressure in the control chamber and has a backing member **86** on the outer side thereof in contact with an end **87** of a control lever **88** pivoted at **90** to the control device **82**. The leg **92** of the control lever **88** is biased by a spring **94**, through linkage to be described hereafter, to urge the lever **88** in a counterclockwise direction about pivot **90** so that the end **87** contacts the backing member **86** and tends to move the diaphragm **84** to the right as viewed in FIGURE 1. An adjustable stop **96** is provided to limit the clockwise movement of the control lever **88**.

The pressure responsive control device **82** is arranged to control the speed of the engine in accordance with the

pressure within the control chamber **80**. To this end, a link **98** connects the leg **92** of the control lever **88** with the throttle lever **100** of a conventional variable speed type governor **102** which controls the speed of the engine in conventional manner as described hereafter. Throttle lever **100** is pivotally mounted at a medial point on a pivot **104** and is connected at one end to the throttle rod **106** of the tractor. The tractor throttle rod **106** is arranged in accordance with conventional practice to operate the throttle lever **100**. The control spring **94** for the control device **82** engages the throttle lever **100** and is arranged to bias the throttle lever **100** in a counterclockwise direction to thereby urge it toward movement to the high speed position. The bias exerted by spring **94** may be varied by the adjustment of a nut **112** or by movement of the throttle rod **106** from the driver's seat. A pair of adjustable set screws **114** and **116** are mounted on the governor housing for limiting the pivotal movement of throttle lever **100**. Screws **114** and **116** provide the high and low speed stops, respectively, for the engine speed controlling mechanism. The stops **114** and **116** regulate the range over which the engine **6** may operate. The link **98** is adjustable in length and is adjusted so that the governor throttle lever **100** may reach the correct minimum speed during compressor operation by the action of control lever **88**. The low speed during tractor operation, is set by adjusting the low speed stop screw **116** which limits the movement of throttle lever **100** when urged toward the slowest speed by rod **106**. The low speed stop screw **116** controls the low speed of the engine during manual operation as will be described hereafter.

A stop washer **107** is provided on the throttle rod **106** to urge the throttle lever **100** against the low speed stop screw **116** to permit lower engine speed during tractor operation than would normally be achieved during compressor operation. To this purpose the link **98** is slideably connected to the throttle lever **100** so the lever **88** can be against the stop **96** and not prevent throttle lever **100** from reaching stop screw **116**.

An adjustable screw **118** engages the usual yoke shaft **120** of the governor **102**. The governor spring **122** is mounted in tension between the head end of screw **118** and an arm **124** mounted on throttle lever **100**. Accordingly, movement of the throttle lever **100** is transmitted to yoke shaft **120** by arm **124**, spring **122** and screw **118**. During speed controlling operation of governor **102**, movement of yoke shaft **120** is transmitted to the engine throttle lever **101** by a connection **103**. The engine throttle lever **101** operates the throttle valve **108** in the intake connection. The governor operation is in accordance with conventional practice. It will be evident that the actuation of throttle lever **100** by control lever **92** controls engine speed by acting on the governor spring **122**.

The operation of the control system in accordance with this invention will be in accordance with the setting of the selector valve **44** which may be set to three operating positions, which may be termed "compressor," "tractor," and "compressor-tractor." These terms denote components of the unit which are to be used.

With the selector valve **44** set in the "compressor" position as is illustrated in FIGURE 2 (i.e. in cases where only the compressor loads the engine), the control system functions as a compressor control. This condition of the system illustrates how the invention may be used for controlling compressors only. In this position of valve **44**, the valve chamber **45** interconnects lines **34**, **48** and **76**. Accordingly, the pressure of the oil being pumped through line **34** by pump **38** is applied to both the unloaders **7** and control device **82** for controlling operation thereof. The communication between line **34** and the unloaders is provided by line **32**. The communication between line **34** and control device **82** is provided by selector valve **44** and line **76**. Selector valve **44** also

provides communication between lines 34 and 48 whereby oil is pumped into inlet chamber 52 of pilot regulator 50.

In describing the operation of the control system when the selector valve 44 is in the "compressor" position, let it be assumed that initially the various parts of the system are as illustrated in FIGURE 1. In this condition of the system, the compressor 2 is being unloaded. The control system is actuated to the unloading condition in response to the occurrence of the high receiver pressure sufficient to meet the demand. The receiver pressure is transmitted through line 70 to the control chamber 66 of the pilot regulator 50. The diaphragm 64 of the pilot regulator 50 is responsive to the occurrence of this high pressure and positions the valve member 60 to shut off the flow of oil through the pilot regulator 50. Since the pump 38 provides a constant delivery of oil through the line 34, the blockage of this flow by the valve member 60 of the pilot regulator 50 causes a high oil pressure to be applied through line 32 to the unloaders 7 and the flow of oil into chamber 22. The unloaders 7 are responsive to the flow of oil through passageways 23 into chamber 22 to acuate the intake valve 8 of the compressor 2 to an unloading position.

The same oil pressure is transmitted to the control chamber 80 of the control device 82. This pressure in the control chamber actuates the control lever 88 to the low speed position in contact with the stop screw 96. In this position the link 98 has actuated the throttle lever 100 to the low speed position as shown in FIGURE 1.

As the pressure in the receiver 68 decreases the pilot regulator 50 will control the unloaders 7 to match the compressor delivery to the demand at the receiver 68. This function is achieved because the pilot regulator 50 will position the valve member 60 to regulate the flow of oil therethrough in accordance with the receiver pressure which is applied to the control chamber 66. The movement of the valve member 60 away from the valve seat 58 in response to a decrease in control pressure permits a greater amount of the oil delivered by the pump 38 to pass through the pilot regulator 50 and accordingly the pressure applied to the unloaders 7 is reduced. This reduction in the unloader pressure decreases the amount of unloading and accordingly increases the volume of compressor output.

It will thus be apparent that the control system in accordance with this invention controls the air delivery by only partly unloading the compressor cylinder according to the demand for air, the details of this operation being described hereafter. This is achieved in response to the action of the pilot regulator 50 which regulates the pressure in the unloaders 7 in proportion to the receiver pressure. As the receiver pressure increases, there is a corresponding rise in unloader pressure to thereby increase the volume of air delivery of the compressor. Conversely, when the receiver pressure falls the pressure regulator 50 reduces the unloader pressure to thereby increase the volume of air delivery of the compressor.

Although the air delivery can be controlled simply by regulating unloader pressure, the apparatus has better performance if speed is reduced as air demand is reduced. This is accomplished by means of the control device 82 which combines with the unloading control to achieve control over the entire range of rated delivery. Accordingly, the control device 82 is also responsive to the receiver pressure and functions to vary the engine speed, and accordingly the compressor speed, to regulate the air delivery of the compressor. The same oil pressure applied to the unloaders 7 is applied to the control chamber 80 of control device 82. Accordingly, as receiver pressure decreases and the flow of oil through the pressure regulator 50 increases, the pressure in the control chamber 80 of the control device 82 decreases whereby the diaphragm 84 is moved to the right, as viewed in FIGURE 1. This movement is followed by the control linkage under the bias of spring 94 so that the throttle lever 100 is moved in a counterclockwise direction to thereby actuate the governor yoke shaft 120 to increase the engine speed through lever 101. It will be evident that the control device 82 actuates the throttle lever 100 to decrease the engine speed in response to an increase in receiver pressure. Moreover, the arrangement in accordance with the invention is such that changes in the pressure applied to control device 50 from the air receiver produce amplified changes in the controlling pressure applied to the control device 82. This pressure change amplification results from the arrangement whereby the valve 60 blocks the flow through the control line from the restricted orifice 36 so that a very small movement in the valve member 60 can produce large pressure changes in chamber 52 and lines 34 and 76 which communicate with the control chamber 80. To illustrate how this amplification occurs, let it be assumed that a condition exists in which valve member 60 is completely closed in response to the occurrence of a high pressure in chamber 64 whereby a very high pressure is applied to the control chamber 80. If the receiver pressure should drop a very slight amount so that a valve member 60 is opened a small amount, the control pressure in chamber 80 will drop much more than the pressure drop in chamber 64 since the oil in chamber 52 can flow relatively quickly past the valve member 60. Of course, a reverse action will occur in response to increases in pressure. Tests performed on action embodiment indicated in FIGURE 1 indicate that changes in the pressure applied to the control device 82 are as high as five times that of the receiver pressure changes which cause the first-mentioned changes.

The control device 82 is operative to control the engine speed throughout any desired range as is determined between stop screws 114 and 96 cooperating with the throttle lever 100. While the ranges of operation of the engine speed control device 82 and of the unloaders 7 may be set to various extents, the type of operation found to be highly satisfactory in the case of combined compressor-tractor units is illustrated by the performance curve shown in FIGURE 5. As is illustrated in this curve, the speed control mechanism is operative to control the volume of air delivery throughout an upper range of air delivery. The unloaders 7 are also operative over this range. The unloaders 7 take over the sole control of the volume of air delivery throughout a lower range. It will be apparent that the unloading control can control air delivery even though compressor speed varies in a random manner as it would during simultaneous operation of the compressor and a tractor mounted accessory such as a pump 4 for powering a hydraulic backhoe.

Referring to FIGURE 5, the operation of the typical compressor system which operates in accordance with the curve shown in this figure will be described. When the maximum pressure exists in the receiver, i.e., 106 p.s.i., there is no air demand on the compressor. In this case, the compressor is unloaded fully and the engine operates at the unloaded speed of about 900 r.p.m.

If the pressure in the receiver drops slightly the compressor will load slightly and the engine speed will decrease. This condition will exist as the pressure varies from the maximum pressure of 106 p.s.i. to about 105 p.s.i. with the air demand varying between 0–44 percent of full rated delivery. The engine speed will vary between about 900 r.p.m. and 700 r.p.m. This condition is shown in the lower portion of the curve of FIGURE 5 designated "unloading action only." With the receiver pressure at 105 p.s.i. at 44 percent of air demand, the compressor is loaded partially with the engine running at the minimum speed of about 700 r.p.m. and the throttle remaining closed at the idle position thereto.

As the receiver pressure drops below 105 p.s.i. and the air demand increases above 44 percent, the throttle

is opened to increase the speed of the engine and the loading of the compressor is increased. This action is continued up to 100 percent of air demand as shown in FIGURE 5. In this condition of the system, the receiver pressure is 100 p.s.i., the compressor is loaded fully, and the engine operates at maximum speed of about 1400 r.p.m.

The selector valve **44** is placed in the "tractor" position when the only load on the engine **6** is the hydraulic pump **4** and the tractor **5** and there is no demand on the compressor **2**. In this position, the selector valve **44** shuts off any flow through the line **34** to either the pilot regulator **50** or the speed control device **82**. Since the flow through the line **34** is blocked, the pump pressure is delivered through line **32** to the unloaders **7**. Accordingly, the compressor **2** is continuously unloaded and does not pump any air. The control device **82** is also inoperative to control the engine speed since the pressure through the line **34** is shut off from the control chamber **80**. Accordingly, the engine speed is regulated by manual actuation of the control rod **106** which controls the throttle setting by way of lever **100** as discussed above.

When the engine is loaded by both the hydraulic pump **4** and the compressor **2**, the selector valve **44** is set to the "compressor-tractor" position. In this position, the selector valve chamber **45** provides communication between the line **34** and the pilot regulator **50** and shuts off the control pressure from the speed control device **82**. Accordingly, the pilot regulator **50** controls the unloaders **7** in response to the receiver pressure to match automatically the compressor delivery to the demand thereon. As the receiver pressure increases the pilot regulator **50** operates to increase the unloader pressure and thereby unload the compressor **2**. Conversely, as the receiver pressure decreases, the pilot regulator **50** operates to decrease the unloader pressure to thereby reduce the amount of unloading and so that a greater volume of air is pumped by the compressor **2** to the receiver **68**. Since the speed control device **82** is shut off and is not responsive to the oil pressure in line **34**, the engine speed is controlled by the manual operation of the control rod **106** which is connected to the throttle lever **100** for setting the throttle.

An analysis of what actually makes a variation in the the unloader pressure cause a corresponding variation in air delivery has been made along with various tests on control systems embodying the invention to illustrate the analysis presented. Briefly stated, the variation of air delivery is a result of providing a varying yielding force which is applied to the intake valve to bias the same to an open position, the yielding force being provided by the pressure applied to the unloaders which serves to vary the time at which the intake valve closes during a compression stroke. In a reciprocating compressor there is provided a dynamic force, or a combination of dynamic and static or kinetic forces, acting to close the intake valve during the compression stroke, and in accordance with the invention, there is also provided a yielding force acting to open the intake valve during the compression stroke. The dynamic force is provided by a combination of many things, such as (1) fluid flow through the intake valve causing a drag which tends to close the valve, (2) fluid flowing at a higher velocity on one side of the intake valve than on the other, (3) the impingement of fluid on the valve acting to close the same, and (4) the inertia of the moving parts of the intake valve tending to close the same. Of course, spring **12** also provides a force acting to close the intake valve. The yielding force can be due to various factors, such as, in the present case, the yielding pressure applied to the unloaders. More specifically, oil under pressure is forced through an orifice (needle valve **36**) into a "reservoir" where it builds up pressure when flow from the "reservoir" is restricted by a control valve **50** actuated in response to the receiver pressure. The "reservoir" comprises the various lines and chambers where the pressure varies in accordance with the movement of control valve **50** and includes line **32**, passageways **23** and unloader chambers **22** in addition to line **34**, chamber **45**, line **48** and chamber **52**. As long as the control valve **50** is the least bit open, the pressure applied to the unloaders **7** is yielding.

In operation, if the yielding force acting to open the intake valve exceeds the dynamic force acting to close the same throughout the entire compression stroke, the intake valve will remain open and no compression can occur. However, if at any time during the compression stroke the dynamic forces tending to close the intake valve exceed the yielding force tending to open the same, the intake valve will close and compression will take place from that time for the remainder of the compression stroke. The instant at which the intake valve will be closed during the compression stroke can be varied by varying the yielding pressure applied thereto. By increasing the yielding pressure the intake valve can be made to close at a later time during the compression stroke. By decreasing the yielding pressure the intake valve can be made to close at an earlier time during the compression stroke. The effect of varying the yielding pressure is to vary the length of the effective compression stroke and consequently the air delivery volume.

It will be evident that what occurs in each of the unloaders is that when the unloader pressure is sufficient to overcome the spring **12** and cause a separation of the unloader pistons **16** and **19**, the intake valve **8** will be maintained off its seat beyond the time in the cycle when it would normally seat for a complete compression stroke. The normal closing point occurs about 36° of camshaft rotation after the bottom dead center point. However, as the piston **11** moves toward the intake valve during the compression stroke, the dynamic forces acting to close the intake valve will build up. When these dynamic forces become greater than the yielding pressure provided by the oil in the unloader chamber **22**, the intake valve will move against its seat and oil will be forced out of the unloader chamber **22**. Oil may be expelled from the unloader chamber **22** as long as the control valve **50** is open since this permits an outlet flow from the "reservoir" described above to accommodate the amount of oil expelled from the unloader chamber. Other means accommodating the outlet flow from the "reservoir" are the flexible diaphragm **84** which can yield to accommodate additional oil and the other unloaders which are not in the compression stroke portion of the cycle. It will be evident that as the amount of opening of control valve **50** is decreased, there will be a greater resistance to flow therethrough and a higher pressure in the unloaders so that the dynamic forces tending to close the intake valve will not be effective until a later time in the compression stroke. Accordingly, there will be less compression since as long as the intake valve is open there will be no compression because the piston **11** merely forces air backward through the intake valve rather than through the discharge valve. Of course, a reverse action will take place as the control valve **50** is opened a greater amount. It will thus be apparent that by merely providing a variable and yielding pressure on the unloaders, it is possible to vary the amount of compression, or conversely the amount of unloading, in proportion to this unloader pressure. It is in this manner that the partial unloading in accordance with the invention is achieved.

To demonstrate the above-discussed operation of the control system, actual tests were conducted on a Model 125 "Pneumapower" compressor manufactured by Schramm, Inc., this compressor being constructed in accordance with the invention. The results of these tests are shown in FIGURES 8, 9, and 10. In FIGURE 8 there is shown a table of test data illustrating what occurred when the compressor was operated at constant speeds of 1200 r.p.m. and 1400 r.p.m. This table illus-

trates the relationship between the unloader pressure and the air delivery demand with respect to the time of intake valve closing in degrees of crank shaft rotation after the normal closing point. These test results are also shown in diagrammatic form in FIGURE 9 and are plotted on the graph of FIGURE 10, the effective compression stroke also being shown in these figures. The tests were performed with a timing light test arrangement whereby when the control valve closed, a circuit means was actuated to effect the flashing of a timing light onto the flywheel which was provided with markings to indicate the time lag of the intake valve closing. As FIGURES 8, 9 and 10 illustrate, the test results demonstrate that as the air demand decreased and unloader pressure increased, the time lag of the intake valve closing increased. This demonstrates that by variation of the unloader pressure in a system constructed in accordance with the invention, there will be produced a corresponding variation in the amount of unloading and the air delivery resulting therefrom. Referring to FIGURES 9 and 10 there will be noted that the time lag is plotted from beyond the normal closing point of the intake valve which occurs approximately 36° after bottom dead center.

It will be apparent that in the case of a control system wherein the engine speed is reduced at the same time that unloader pressure is increased, the effect is somewhat different than in the case of constant speed as discussed above. The curve shown in dotted lines on FIGURE 10 illustrates generally what occurs when the speed is reduced along with an increase in the unloader pressure. As shown in upper portion A of the dotted curve, the valve timing remains substantially normal, that is without lag, as long as the air delivery is being reduced by reducing the engine speed only. In this case, the speed was reduced from 1370 r.p.m. to 800 r.p.m. by a speed control device such as 82 in FIGURE 1. When the speed control lever comes against the low speed stop, any further reduction of air delivery is by unloader action only. What occurs as the air delivery is decreased further by unloading with a fixed throttle, is shown by the remainder of the dotted curve in FIGURE 10, the portion B involving a speed reduction from 800 to 700 r.p.m.

In FIGURE 11 there is shown another control system in accordance with the invention, which system is similar to that shown in FIGURE 1. The system shown in FIGURE 11 differs from that shown in FIGURE 1 mainly in that it involves use of air for actuating the unloaders, although there are other differences which will appear hereafter. In view of the similarity of the systems, parts of the system shown in FIGURE 11 corresponding to parts of the system shown in FIGURE 1 will be designated with the same reference numeral with primes added.

In the system shown in FIGURE 11, the air receiver 68' is connected by a line 70' to the upper chamber 66' of a control device 50' which has the same construction as the control device 50 shown in FIGURE 1. The regulator valve 50' has an inlet chamber 52' connected to the control chamber 80' of a speed control device 82' by a line 76'. The inlet chamber 52' is in communication with the receiver 68' by means of a line 34' connected between line 76' having an orifice defining means 36' therein. The receiver pressure is supplied to line 34' upstream of the orifice 36' by way of a line 40' connected to a lower chamber 152 of a pilot valve 150, which chamber is also in communication with the upstream end of line 34'.

The regulator valve 50' has an outlet chamber 54' which is connected to a bleed line 56'. There is provided a valve member 60' cooperable with a seat to regulate flow through the valve 50' and a spring 72' acting on the diaphragm 64' to bias the valve member 60' to its open position, the valve member being actuated to the closed position in response to the pressure in the control chamber 66' which pressure is actually the receiver pressure.

The pressure responsive speed control device 82' is essentially the same as the speed control device 82 shown in FIGURE 1, the only essential differences being the provision of a double diaphragm arrangement 84' instead of the single diaphragm of the device 82 and the mounting of the control spring 94' in direct engagement with control lever 88'. The control device 82' is provided with a control lever 88' which is pivoted in response to the movement of the diaphragm arrangement 84'. A spring 94' biases the lever 88' to the high speed position. The control device 82' is arranged to control the speed of the engine in accordance with the pressure within the control chamber 80' by way of the link 98' which is connected to control the engine speed in the same manner as the corresponding link in the system shown in FIGURE 1.

The line 40' from the receiver 68' is connected to a standard unloader pilot valve 150 of the snap action type illustrated in U.S. Patents Nos. 2,023,418 and 2,160,860. This valve comprises a casing in which is located a disc 160 operating between a lower seat 161 and an upper seat 162, the disc being urged downwardly by a spring 164. The space above the lower seat is connected to a ball check valve and the space below this seat, i.e., chamber 152, is connected to the air receiver supply line 40'. Reference to the last-mentioned patents will reveal that the unloader pilot valve 150 is essentially as indicated in these patents. A hole 170 extending axially through the spring adjusting screw 172 furnishes communication between the interior of the casing above the upper seat 162 and the atmosphere. A filter means 176 serves to filter the air entering the valve from the line 40'. The upstream end of line 34' is connected to the valve 150 and communicates with a chamber 152 therein downstream of the filter 176 and below the lower valve seat 161.

A T fitting 190 communicates with the interior of the casing of the pilot valve 150 at a location between the valve seats 161 and 162. The aligned ends of the fitting 190 are connected to air lines 200 and 202, the end communicating with the line 200 being provided with a ball check valve 204 resting on a seat 206 and retained in the fitting by a cross pin. There is provided a bypass passage 208 for the ball check valve assembly, the bypass being provided with a needle valve 209 which is adjustable to regulate the flow around the ball check valve. The check valve 204 permits free flow of air through the valve seat 205 to the line 200 but prevents reverse flow therethrough, this reverse flow being confined to the bypass passage 208 and needle valve 209. The line 202 is connected by way of a line 210 to the compressor unloaders 7'. Line 210 is connected to the passageway 23' provided in the unloader bracket 18' which is constructed in a manner similar to the corresponding unloader structure described in detail with respect to the control system shown in FIGURE 1. The line 210 is also connected to a passage 212 in the speed control device 82', such passage leading to the space between the double diaphragm arrangement 84.

The line 200 is connected to an unloader 214 for the supercharger compressor. The supercharger unloader is a poppet valve type which is operated by air from the pilot valve 150 whenever the compressor is loaded and unloaded. The unloader 214 unloads the supercharger by discharging the air delivered therefrom to the atmosphere and thus preventing excessive discharge pressure from building up at the supercharge outlet when the compressor stops pumping air. To this end, the unloader 214 is connected to the supercharger by a connection 216 which communicates through a valve and valve seat arrangement with a connection 218 connected to atmosphere. The supercharger unloader 214 comprises a valve 220 cooperable with a valve seat 222 between the connections 216 and 218. The valve member 220 is actuated away from the valve seat 222 by a piston means

224 movable in a cylinder 226. The line 200 is connected to the cylinder 226 to apply air pressure to the end of the piston 224 to move the same downwardly and cause a corresponding movement of the valve 220 against the bias of the spring 228. In operation, the piston 224 on the valve stem opens the valve 220 whenever air is admitted to the cylinder 226. When the air receiver pressure is reduced to a point where the pilot valve 150 acts to load the compressor, it also causes the supercharger unloader valve to close. The ball check valve 204 is adjusted to allow a short period of time to elapse between the time that the compressor loads and the supercharger unloader valve 220 closes. This pause allows the compressor air demand to increase sufficiently before the supercharger "cuts in" to prevent popping a safety valve between the supercharger and the compressor.

A typical operation of the control system shown in FIGURE 11 will involve control of the engine and compressor speed by the speed control device 82' when the air demand is above 20% of total capacity and control by unloading of the compressor when the air demand falls below 20% of total capacity. In operation, when the air demand is above 20% of total capacity, small pressure changes in the air receiver 68' are amplified by the regulator 50' and applied to the control device 82' to match air delivery to demand. The air receiver pressure is applied to the control chamber 66' of the regulator 50' by way of line 70' and acts on the diaphragm 74' to cause it to raise or lower the valve member 60' toward and away from the valve seat to control flow through the regulator 50'. In this manner, by permitting a certain amount of air to bleed through line 56', the pressure acting on the speed control device 82' is raised or lowered. The air for actuating the speed control device 82' is supplied from the air receiver 68' through the restricted orifice 36' by way of line 40', the bottom chamber 152 of valve 150 and line 34'. It is noted that this air is filtered by the filter medium 176 provided.

It will be apparent that pressure change amplification will be provided in the same manner as in the control system shown in FIGURE 1. Since the control air is supplied to regulator 50' through an orifice provided by the means 36', small movements of the regulator valve member 60' in response to small pressure changes in the control chamber 66' will cause large pressure changes on the upstream side of the valve member 60', these pressure changes being applied to the speed control device 82'. For example, if the regulator valve member 60' were closed, a pressure approximating the actual receiver pressure will be applied to the speed control device 82'. However, as soon as the valve member 60' is moved slightly from the valve seat, the air in chamber 52' can flow rapidly therefrom to cause substantial pressure change in this chamber and in chamber 80'. Actual tests on a control system in accordance with that shown in FIGURE 11 have demonstrated the effectiveness of the very simple pressure change amplifier provided therein. In the actual performance, a receiver pressure change of about 5 p.s.i. caused an actual pressure change in the speed control device of about 70 p.s.i., which involves a ratio of 14:1. At the same time, the engine speed change was about 1000 r.p.m. and the air delivery varied from 600 to 120 cubic feet per minute. It is thus apparent that the system in accordance with the invention can be used to control a wide range of air delivery in response to a relatively small change in receiver pressure. Moreover, it is apparent that the pressure change amplifier system can be operated in various ways, such as entirely on air, entirely on hydraulic, or a combination of both. In other performance tests 1 p.s.i. pressure change in the receiver has been amplified to a 20 pound per square inch change in the speed control device. This is with the use of air over air type of control.

Continuing with the description of a typical operation of the system, when the air demand falls below 20%, the engine will run continuously at 600 r.p.m. while the compressor is pumping air. Since the air delivery exceeds the air demand, the air receiver pressure continues to rise above that needed to slow the speed to 600 r.p.m. When the pressure becomes high enough the pilot valve 150 will be opened to permit the air receiver pressure to enter the compressor unloader 7', the space between the diaphragms 84, and the supercharger poppet valve 214 to prevent the compressor from pumping air and to further reduce engine speed during unloading if desired. The further reduction of engine speed is possible with the same pressure required for reduction to 600 r.p.m. because the area of the front diaphragm acted upon by the same pressure is greater than that acted upon on the rear diaphragm. This is due to the shape of the spacer piece which separates the two diaphragms. For a more complete description of this feature refer to Patent No. 2,595,369.

The pilot valve 150 is adjusted by varying the force provided by spring 164 so that at the desired pressure, the disc 160 will be snapped upwardly against the upper seat 162 by the pressure beneath it with the result that the receiver pressure is applied through the pilot valve 150 into the T assembly 190. The snap action of the pilot valve 150 is as described in said prior patents and need not be repeated here. It need only be noted that the area of the disc 160 subjected to the receiver pressure is substantially increased when it rises from the lower seat 161 so that it will snap upwardly and will not return to that seat until the tank pressure drops quite substantially. A stable snap action is thus secured.

With the pilot valve 150 now in the snapped open position, receiver pressure is applied through the T assembly 190 and lines 202 and 210 to the compressor unloaders 7'. The unloaders 7' will function in a manner similar to the unloaders described in detail with respect to the system shown in FIGURE 1, the only essential difference being that air under pressure is supplied to the unloader chambers instead of oil under pressure. However, in this case the unloaders are subjected continuously to full receiver pressure and the intake valves are held open continuously, thus preventing the compressor from pumping air. As long as receiver pressure is high enough to overcome the dynamic forces tending to force the intake valve closed, the intake valve stays open and no air is delivered.

At the same time that air is applied to the compressor unloaders, air is also applied to the supercharger unloader 214 to open the poppet valve 220 and allow the supercharger to blow out to atmosphere.

When the air pressure in the receiver 68' drops somewhat, the disc 160 of the pilot valve 150 is forced back to its lower seat 161 by the spring 164 and air trapped in the unloaders 7' is released by way of the passageway 170 extending through the upper end of the pilot valve 150. In order to give the compressor time to start pumping before the supercharger does, the adjustable needle valve 209 restricts the flow of air from the supercharger unloader 214 for a short time in the manner described above.

It will be noted that the speed control device 82' will function to reduce the engine speed in proportion to the demand and its operation depends upon the variations of receiver pressure with the changing demand for air. Typically, the speed control device may be adjusted to maintain, at rated speed, 100 p.s.i., and as the air demand decreases there will be an increase in the pressure in the air receiver. When the speed is reduced to approximately three-eighths of rated speed due to a lessening of the demand for air, the pilot valve 150 will act to unload the compressor and the speed control device will reduce the engine to idle speed. The engine will continue to idle until air receiver pressure drops to a point at which the pilot valve is set to load the compressor. At this point, the compressor will load and the speed control device will simultaneously adjust the throttle to operate

the engine at a speed proportionate to the volume of air being used.

The compressor control system shown in FIGURE 12 is very similar to that shown in FIGURE 11, the main difference being that instead of unloading with air, a hydraulic unloading system is provided essentially the same as the hydraulic unloading system described with respect to the system of FIGURE 1. Accordingly, in FIGURE 12 the same reference numerals have been used as in the system of FIGURE 1 for the corresponding parts.

To achieve the hydraulic unloading, the line 210 is connected to a pressure regulator 50'' identical with the regulator 50 of the system shown in FIGURE 1. Accordingly, the receiver pressure is applied to the control chamber 66'' of this regulator when unloading is to occur as described with respect to FIGURE 11. The unloading is then effected in a manner the same as is described with respect to FIGURE 1. To this end, there are provided unloaders 7 mounted in the unloader bracket 18 and actuated by cams 21. Oil is introduced into the unloader chambers through the passageway 23 in the unloader bracket 18 when unloading is to be accomplished. Passageways 23 have a line 32 connected thereto which line communicates with another line 34 connected to the downstream end of an orifice defining means 35. The upstream end of the orifice 35 receives oil delivered from a pump 38 which receives oil through a line 40 connected to the crankcase of the engine. The pump 38 is provided with a conventional bypass or pressure relief valve 37. The line 34 passing from the orifice 35 is connected to the inlet chamber 52'' of the regulator 50''.

Accordingly, the compressor unloading is carried on in the manner identically to that described with the hydraulic system of FIGURE 1. In all other respects, the system shown in FIGURE 12 operates in a manner of the control system shown in FIGURE 11.

In FIGURE 13 there is shown a control system in accordance with the invention involving the use of a compressor running at a constant speed. The compressor may be driven at a constant speed by either a constant speed motor or an engine running at a substantially constant speed. The system shown in FIGURE 13 comprises a regulator valve 50 identical with that shown in FIGURE 1 and a pilot valve 150 identical with that shown in FIGURE 11 wherefore corresponding parts have been given corresponding reference numerals. The control chamber 66 of the regulator 50 is connected to the air receiver 300 by way of lines 302 and 304. The air receiver 300 is also connected to the lower chamber of the pilot valve 150 by way of line 304 and line 302. The outlet connection 151 of the pilot valve 150 is connected by a line 306 to a relay valve 330 having a cylinder 334 formed therein. A piston 336 is arranged to move within the cylinder 334 and is biased upwardly by a spring 338. The control chamber 340 for the relay valve is on the upper side of the piston 336. A lower chamber on the bottom side of the piston 336 is indicated at 342 and has a pair of lines 344 and 346 connected thereto.

The line 344 is connected to an unloader apparatus 7 which is the same as the unloader apparatus shown in FIGURE 1. Line 344 also has the output from the downstream side of a needle valve 350 connected thereto, the inlet side of the needle valve 350 being arranged to receive oil delivered from a pump 352 by way of line 354. The pump 352 receives oil through a line 356 connected to the crankcase of the engine.

The line 346 is connected to the inlet chamber 52 of the valve 50. The outlet chamber of the valve 50 being connected by a line 356 to return the oil to the crankcase.

There is provided a pressure relief arrangement for bypassing the valve 50 and for limiting the maximum pressure applied to the unloaders 7. This means comprises a bypass line 360 connected between lines 346 and 356 and having pressure relief valve 362 connected therein. In a typical system, the pump 352 is a type for delivering a relatively high pressure, such as 80 p.s.i. for example. The pressure relief valve 362 typically will be set to maintain a pressure of 35 p.s.i. in the lines 344 and 346 which are connected to the unloaders 7. Typically, the pilot valve 150 is set to open about 3 p.s.i. above the pressure at which the unloader pressure reaches 35 p.s.i. The pressure on the unloaders 7 below 35 p.s.i are controlled by the regulator valve 50 according to the valve setting and the air receiver pressure.

The operation of the control system described will be best understood by considering initially that the tank is at a minimum pressure. Under these conditions, the pilot valve 150 is closed so that the upper chamber is vented to atmosphere through passage 170 and flow through the pilot valve 150 is blocked. The atmospheric pressure applied to the control chamber 340 of the relay valve 330 is insufficient to overcome the force of spring 338 so that the piston 336 is in the upper position. In this position of the valve 330, flow is permitted through the chamber 342 between lines 344 and 346.

The low receiver pressure is also applied to the control chamber 66 of valve 50 and is insufficient to overcome the spring 72 so that the regulator valve 50 is also in an open position. Accordingly, oil is delivered from the pump 352 through the needle valve 350 and line 344, chamber 342, line 46 to the inlet chamber 52 of the control valve 50. The oil will pass through the control valve 52 to line 356 and return to the crankcase.

As the receiver pressure rises, and this rise is below the pressure sufficient to open the pilot valve 150, the pressure changes are applied to the control chamber 66 of regulator 50. The diaphragm 64 of the regulator 50 will be responsive to the increases in pressure to position the valve member 60 gradually closer to the valve seat to gradually increase the blockage of flow through the regulator 50. As the flow is blocked, the pressure in lines 346 and 344 which pressures are applied to the unloaders 7 will be increased so that a higher oil pressure is applied to the unloaders 7. It will thus be apparent that partial unloading of the compressor will occur in the manner described above in accordance with the pressures applied to the unloaders 7. It will be noted that the unloader pressure will be limited to an increase up to the setting of the relief valve 362, that is, 35 p.s.i. At this point, the unloading will be maintained at a desired amount. In a typical system, this condition is set to occur at approximately 50% of full rated air delivery.

If the receiver pressure continues to increase above the pressure sufficient to snap the disc of the pilot valve 150 to the upper position thereof, air and receiver pressure passes through the valve 150 to line 306 and to the control chamber 340 of the relay valve 330 and forces the piston 336 downwardly to block flow through the relay valve by covering the port receiving oil from the line 344. Unloader pressure will then rise to the pressure delivered by the pump 352, that is 80 p.s.i., whereby the compressor unloads fully.

This condition will exist until the air receiver pressure drops to the low pilot valve setting whereby the disc of the pilot valve is snapped to the position in which the disc engages the lower valve seat. This allows the piston 336 to return to the upper position under the bias of 338, air being bled from the line 306 by way of the passage in the valve 150. The control system is now again under the regulation of the regulator valve 50 and the relief valve 362 as described above.

In the use of the control system shown in FIGURE 13 with a constant speed motor, the line 306 may be connected to a pressure operated switch which operates to stop and start the motor automatically in response to the control of the pilot valve 150. This switch means may or may not be made operative, this being determined by the particular application which may or may not require starting and stopping of the motor.

It will be apparent that changes may be made in the control systems in accordance with the invention without departing from the scope of the invention. For example, the regulator **50** may be controlled by using air to control air pressure or by using air to control oil pressure. Moreover, the using of an air receiver is not necessary for the operation of the system in accordance with the invention since no substantial volume of air is necessary for the loading and unloading in accordance with the invention. The system may be operated by a connection directly to the output of the compressor which is another feature of the invention.

Accordingly, it is not desired to be limited except as required by the following claims.

What is claimed is:

1. In combination, a receiver for compressed gas, a compressor connected to said receiver to deliver gas thereto, said compressor being of the reciprocating type including a piston reciprocating in a cylinder, an inlet valve for the cylinder and an exhaust valve for the cylinder, an engine driving said compressor, pressure responsive means for unloading said compressor proportionately in response to a controlling pressure, pressure responsive means for operating the engine throttle to vary the engine speed in response to variations in said controlling pressure, means for supplying said controlling pressure including means for delivering fluid under pressure to a control line, said fluid supplying said controlling pressure, control means connected in pressure responsive communication with said receiver for varying said controlling pressure in accordance with said receiver pressure, conduit means connecting said control line to said unloading means for delivering said controlling pressure thereto, means for delivering said controlling pressure to said engine speed varying means, said control means being responsive to increases in receiver pressure to vary said controlling pressure in a direction which actuates said unloading means to unload said compressor and which actuates said throttle operating means to decrease the engine speed, said control means including a pilot valve means for regulating the rate of flow through said control line to vary the fluid pressure in said control line, said pilot valve having a control chamber in pressure responsive communication with said receiver, and a selector valve controlling fluid flow from said control line to said pilot valve and said engine throttle operating means, said selector valve having one selective position in which said control line is connected to both said pilot valve and said throttle operating means.

2. The combination according to claim **1** wherein said selector valve has a second selective position in which said control line is shut off, and a third selective position in which said control line is connected only to said pilot regulator.

3. In combination, a receiver for compressed gas, a compressor connected to said receiver to deliver gas thereto, said compressor being of the reciprocating type including a piston reciprocating in a cylinder, an inlet valve for the cylinder and an exhaust valve for the cylinder, an engine driving said compressor, pressure responsive means for unloading said compressor proportionately in response to a controlling pressure, pressure responsive means for operating the engine throttle to vary the engine speed in response to a controlling pressure, said throttle operating means including a linkage connected to the engine throttle rod and an expandable chamber for actuating said linkage, means for supplying said controlling pressure including means for delivering fluid under pressure to a control line, said fluid supplying said controlling pressure, control means connected in pressure responsive communication with said receiver for varying said controlling pressure in accordance with said receiver pressure, conduit means connecting said control line to said unloading means for delivering said controlling pressure thereto, and means for delivering said controlling pressure to said engine speed varying means, said control means being responsive to increases in receiver pressure to vary said controlling pressure in a direction which actuates said unloading means to unload said compressor and which actuates said throttle operating means to decrease the engine speed, said control means including a pilot valve means for regulating the rate of flow through said control line to vary the fluid pressure in said control line, said pilot valve having a control chamber in pressure responsive communication with said receiver, and a selector valve controlling fluid flow from said control line to said pilot valve and said engine throttle operating means, said selector valve having one selective position in which said control line is connected to both said pilot valve and said throttle operating means, a second selective position in which said control line is shut off, and a third selective position in which said control line is connected only to said pilot regulator, said linkage being manually operable when said selector valve is in said second and third positions shutting off communication between said control line and said throttle operating means.

4. In combination, a source of compressed gas, a compressor having its discharge connected to said source, said compressor being of the type including a piston reciprocating within a cylinder, an intake valve for the cylinder and an exhaust valve for the cylinder, a pressure source supplying a controlling pressure to a control chamber, pressure responsive unloader means associated with said intake valve for partially unloading said compressor, conduit means connecting said control chamber to said unloader means, said conduit means and said control chamber providing a closed system, and pilot regulator means connected in pressure responsive communication with said source of compressed gas for venting said closed system various amounts for sustained periods in proportional relation with variations of the pressure of said source of compressed gas, the pressure in said closed system being varied in proportion to the variations of the pressure of the source of compressed gas so that as the pressure of the source of compressed gas varies throughout a range, the unloader partially unloads the compressor in proportional relation.

5. The combination according to claim **4** wherein said unloader means comprises cam operated means operatively connected to the compressor intake valve to open the same in timed relation with the compressor piston movement and including an unloader in the operative connection to the intake valve.

6. The combination according to claim **5** wherein said unloader includes a pair of pistons defining an unloader chamber therebetween and arranged for relative movement to prevent closing of said intake valve in accordance with said timed relation, said conduit means providing communication between said unloader chamber and said control chamber.

7. The combination according to claim **6** wherein said pressure source supplying said controlling pressure to said control chamber comprises a pump delivering liquid under pressure to said control chamber, said conduit means delivering liquid from said control chamber to said unloader chamber.

8. The combination according to claim **6** comprising an engine for driving said compressor, said engine having a crankcase containing oil, said pressure supply source comprising a pump for delivering oil from said crankcase to said control chamber at a pressure high enough to cause unloading of said unloader.

9. The combination according to claim **4** including means driving said compressor, pressure responsive means connected to said compressor driving means to vary the speed thereof in proportional relation with the variations of the pressure of the source of compressed gas, conduit means connecting said pressure responsive speed varying means with said control chamber, said pilot regulator

means being arranged to vary the pressure in said closed system so that as the pressure of the source of compressed gas increases over a particular range, said unloader means increases the unloading of said compressor and said speed varying means decreases the speed of said compressor driving means.

10. The combination according to claim 4 wherein said pressure source supplying said controlling pressure to said control chamber comprises a pump for delivering liquid under pressure to said control chamber, said conduit means directing liquid from said control chamber to said unloader means.

11. The combination according to claim 4 comprising means defining a restricted orifice between said controlling pressure supply and said control chamber so that said controlling pressure supply is delivered through said restricted orifice to said control chamber, said pilot regulator means including a valve means for blocking flow from said control chamber various amounts in response to variations of the pressure of said compressed gas.

12. The combination according to claim 4 comprising means for driving said compressor at a constant speed, and wherein said unloader means includes cam operated means operatively connected to the intake valve to open the same in timed relation with the piston movement including an unloader in the operative connection to the intake valve, said unloader including a pair of pistons defining the unloading chamber therebetween and arranged for relative movement to prevent closing of said intake valve in accordance with said timed sequence, said pressure source supply including means for pumping oil at a high pressure to said control chamber, said control chamber comprising a control conduit, pressure operated valve means connected in said control conduit for controlling flow therethrough, said unloader conduit means being connected to said unloader chamber and to said control conduit at a location upstream of said last-named valve means, said pilot regulator being connected to said control chamber downstream of said last-named valve means, said last-named valve means being biased to an open position, and pressure responsive means in communication with said source of compressed gas for actuating said valve means to a closed position blocking flow through said control conduit to said pilot regulator in response to the occurrence of a predetermined high pressure of said source of compressed gas whereby the high pressure of said pump is delivered to said unloader chamber.

13. The combination according to claim 4 including an engine driving said compressor, a throttle for controlling the engine speed, pressure responsive means for operating the engine throttle to vary the engine speed in response to variations in a controlling pressure, said throttle operating means including linkage connected to the engine throttle and an expandable chamber for actuating said linkage, and means for delivering the pressure in said control chamber to said expandable chamber.

14. The combination according to claim 4 wherein said connection between said source of compressed gas and said pilot regulator comprises a conduit having a normally closed pilot valve therein for controlling flow to said pilot regulator, said pilot valve being responsive to the occurrence of a predetermined pressure in said last-named conduit for movement to an open position to allow flow to said pilot regulator.

15. The combination according to claim 14 including a supercharger blower, means for unloading said supercharger blower including pressure responsive piston means, and means connecting said last-named pressure responsive piston means to said conduit connected between said source of compressed gas and said pilot regulator at a location upstream of said pilot valve, and check valve means in said connection to said piston means restricting flow in the reverse direction.

16. The combination according to claim 14 comprising an engine driving said compressor, pressure responsive means for varying the engine speed in response to variations in the pressure of said compressed gas, conduit means connecting said source of compressed gas to said engine speed varying means, and a second pilot regulator connected to said last-named conduit means and in pressure responsive communication with said source of compressed gas for venting the said last-named conduit means various amounts in proportional relation with the variations of the pressure of said source of compressed gas.

17. The combination according to claim 16 including a restricted orifice located in said last-named conduit means connecting said source of compressed gas to said speed varying means, said second pilot regulator being connected to said last-named conduit means at a location downstream of said restricted orifice.

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