

June 11, 1946.

P. A. CONDIT ET AL

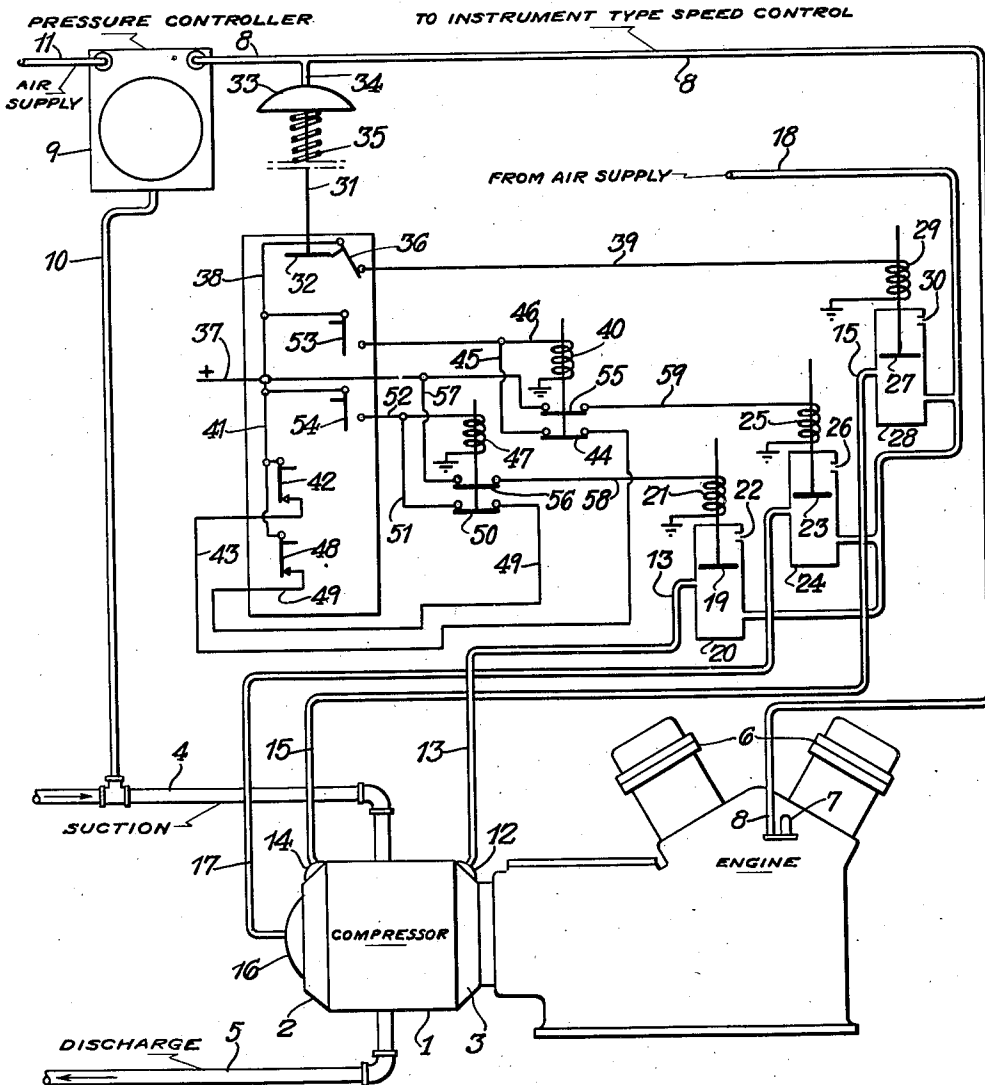
2,401,910

COMPRESSOR REGULATING APPARATUS

Filed March 17, 1943

2 Sheets-Sheet 1

Fig. 1.



INVENTORS.
PAUL A. CONDIT.
KENNETH RUHL.
BY
Hard Crosby & Neal
ATTORNEYS.

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2 Sheets-Sheet 2

Fig. 2.

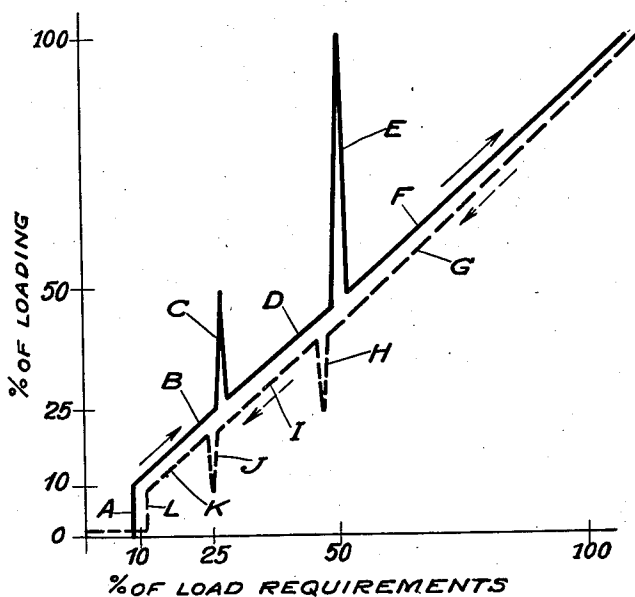
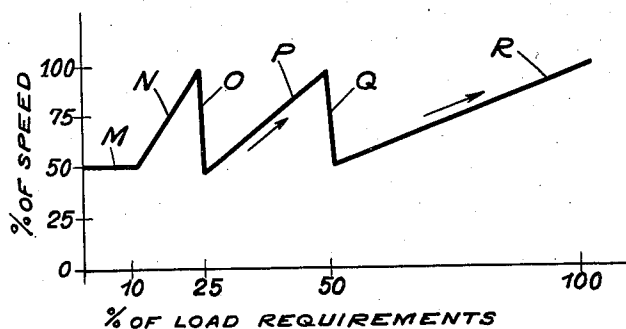


Fig. 3.



INVENTORS.
PAUL A. CONDIT.
BY KENNETH RUHL.
Edward Crosby Kuhl
ATTORNEYS.

UNITED STATES PATENT OFFICE

2,401,910

COMPRESSOR REGULATING APPARATUS

Paul A. Condit and Kenneth Ruhl, Mount Vernon, Ohio, assignors to Cooper-Bessemer Corporation, Mount Vernon, Ohio, a corporation of Ohio

Application March 17, 1943, Serial No. 479,454

5 Claims. (Cl. 230-7)

1

The invention relates to the regulation of the output of gas compressors or similar gas or liquid pumps. In the unloading of compressors the usual practice has been to reduce the speed of the prime mover, in proportion to the reduction in demand, down to as low a speed as possible, regulation below minimum speed being secured by resort to unloading devices such as clearance chambers and suction valve openers, or combinations of such devices, which would further unload the compressor cylinder in step by step fashion. The result of the above prior art practice was to produce straight line regulation through the part of the range in which the output could be controlled by varying the speed of the prime mover, with less accurate control where greater unloading was required because of the abrupt changes in loading produced by cutting the step unloading devices into and out of operation as needed to approximate the desired degree of unloading.

By the present invention we aim primarily to extend the range through which substantially straight line regulation may be obtained. In its broader aspects the invention involves regulation by a speed controlled prime mover driving the compressor and so coordinated with step unloading devices that when one or more of the step unloading devices is cut into or out of operation, the speed of the prime mover is thereupon increased or decreased as needed to reduce or smooth out the abrupt change in output which the step unloaders otherwise would cause, speed control being thus utilized over the range of unloading afforded by the step unloading devices to cause the combined unloading or loading effect to vary in straight line fashion proportional to changes in demand. Thus straight line regulation is achieved over the combined range afforded by the two methods of regulation.

Further objects and advantages of the invention will be in part obvious and in part specifically referred to in the description hereinafter contained which, taken in conjunction with the accompanying drawings, discloses schematically certain preferred forms of apparatus capable of operating in accordance with the invention. The disclosure however should be considered merely as illustrative of the principles of the invention in its broader aspects.

In the drawings—

Fig. 1 is a schematic view showing one form of apparatus constructed to operate in accordance with the invention.

Figs. 2 and 3 are graphs illustrating the oper-

2

ation of the apparatus of Fig. 1 when operated over its range of action.

Many of the parts used in carrying out the invention as hereinafter described may be individually of construction and mode of operation well known in the art, and accordingly are not illustrated nor described in detail. In Fig. 1, for example, is shown a double action compressor cylinder 1 which may be understood as of well known reciprocating type having a head end 2 and a crank end 3, the suction pipe of the compressor being indicated at 4 and the discharge pipe at 5. The compressor may be understood as driven by appropriate gas engine cylinders 6 or equivalent prime mover unit, the speed of which is controlled by a suitable speed control regulator 7, appropriate forms of which are obtainable on the market. The form of speed control regulator shown in Fig. 1 may be understood as of known type which causes the speed of the prime mover 6 to vary between maximum and minimum limits, as the pressure obtaining in a pipe 8 connected to the element 7 fluctuates between certain limits.

A control mechanism indicated diagrammatically at 9 is used, which is responsive to load requirements of the compressor. The illustrated form of control mechanism may be understood as being of a type known in the art as a controller, which will respond to different load requirements as indicated by suction pressure, or discharge pressure, or flow, the controller operating to proportionately regulate the pressure in a control line connected thereto when the compressor is operating above or below the rate called for by the then existing demand on the engine. As shown in Fig. 1 the pressure controller 9 is connected to the suction pipe 4 of the compressor by a pipe 10, so as to respond to suction pressure. A pipe 11 leading to the pressure controller supplies fluid under constant pressure. A pressure delivery pipe 8 (previously described as connected to the speed regulator 7) leads from the pressure controller and constitutes the control line in which the pressure controller regulates the pressure as above mentioned. For example, if the pipe 11 be connected to a source of supply of air under 20 lbs. pressure, the pressure in line 8 may fluctuate between 0 and 20 lbs. and the speed regulator 7 may be such that minimum operating speed of the engine 6 is reached when the pressure in line 8 drops to 8 lbs. and maximum speed of the engine is reached when the pressure in pipe 8 rises to 15 lbs.

As previously mentioned the regulation effected

by the speed regulator 7 is combined with regulation obtained by appropriate forms of step unloading devices which may be such as are per se known in the art. For example, in the form of the example shown in Fig. 1, the flexible diaphragm 12 may be understood as controlling a suction valve opener located at the crank end of the compressor cylinder, detailed construction being neither illustrated nor described since appropriate forms of such devices are known in the art, it being understood that fluid under pressure is supplied to diaphragm 12 through a pipe 13 when the suction valve at the crank end of the compressor cylinder is to be held open.

Likewise a similar diaphragm 14 and pipe 15 at the head end of the cylinder may be provided to hold open a suction valve at the head end when desired. A further flexible diaphragm 16 operated by fluid pressure in the pipe 17, is also shown at the head end of the cylinder for opening and closing a clearance pocket, the latter being neither illustrated nor described in detail since suitable forms thereof are known in the art. When pressure is present in all of the pipes 13, 15 and 17, the compressor is completely unloaded, the suction valves at both ends of the cylinder being held open and the clearance pocket at the head end being also in action.

As a first step in the loading sequence the pressure in pipe 15 may be released to release the suction valve opener operated by diaphragm 14. This with the engine 6 operating at minimum speed may result in a loading of say 10%. Then as later described more in detail, the engine may be speeded up to a maximum so as to respond proportionately to increased demand up to say 25% capacity. Upon increase in demand slightly above 25% the pressure in pipe 17 may be released to increase the loading to 50% of capacity, but simultaneously the speed of the engine 6 is cut down to minimum so that loading immediately reverts to the demand of slightly more than 25%. Then by increasing the engine speed from minimum to maximum, increased loading requirements may be responded to in straight line fashion up to about 50%. Upon increase in requirements above 50%, pressure is released in the pipe 13 to release the suction valve opener which is held open by diaphragm 12. This increases the loading up to 100% but simultaneously the speed of the engine 6 is cut down sufficiently to produce a loading of only slightly more than 50%. By proportionate increase in engine speed, the loading may be thereafter increased in straight line fashion up to maximum capacity. Thus in the example under discussion regulation in straight line may be obtained throughout the range between about 10% of capacity up to 100%.

When the compressor is to be progressively unloaded from operation at maximum capacity, the sequence is reversed, the regulator 7 being initially in position to operate the engine 5 at highest speed, and all of the step unloading devices being out of action. The unloading sequence in the example under discussion will first involve reducing the speed of the engine 6 down to a minimum, this regulation being in straight line manner down to about 50% of capacity. Thereupon pressure is supplied to pipe 13 to hold open the suction valve on the crank end of the compressor cylinder, which will produce a further drop in output to about 25% of capacity, but the engine is thereupon speeded up simultaneously, the change in pressure affecting speed

regulator 7, so as to bring the output nearly back to 50% of capacity, whereupon regulation of output down to 25% may be obtained in straight line manner by reduction in speed of engine 6 through speed regulator 7. At this point pressure may be supplied in pipe 17 to throw into operation the clearance pocket at the head end of the compressor cylinder, thereby reducing the output to about 10% of capacity, but by the simultaneous speeding of the engine 6 through the change in pressure affecting speed regulator 7, the output will be brought back to 25% of capacity. Regulation down to 10% of capacity in straight line manner may then be obtained by reduction in the speed of engine 6 through regulator 7. When operation at zero capacity is desired, pressure may be supplied through pipe 15 to hold open the suction valve at the head end of the cylinder, thereby totally unloading the compressor and operating the engine 6 at minimum speed.

Various systems involving pilot mechanism for effecting sequences comparable to those above described, and operated electrically, or by pneumatic or liquid pressure, or combinations thereof, are known to the art. The system shown in Fig. 1 may be described as of electro-pneumatic type, with air under pressure being supplied through a pipe 18. The diaphragm 12 is controlled by a plunger 19 operating in a cylinder 20 and actuated by a solenoid 21. When the solenoid is energized, plunger 19 is raised to open communication between pressure pipe 18 and pipe 13. When solenoid 21 is deenergized plunger 19 falls, venting the pressure in pipe 13 through a port 22 in cylinder 20.

The diaphragm 16 is likewise controlled through a similar plunger 23, cylinder 24, solenoid 25 and venting port 26. The diaphragm 14 is likewise controlled by a similar plunger 27, cylinder 28, solenoid 29 and venting port 30.

A pilot mechanism suitable for operating plungers 19, 23 and 27 in proper sequence during the loading and unloading operations, will now be described. In Fig. 1 we have shown for this purpose a reciprocating rod 31 having a cross bar 32, which is operated by a suitable pneumatic pressure responsive device 33 connected by pipe 34 to the variable pressure pipe 8 previously described, and loaded by a spring 35, the arrangement being such that when the pressure in pipe line 8 is at a minimum the cross bar 32 is held in uppermost position by spring 35. This uppermost position corresponds to fully unloaded condition of the compressor, and in this position the cross bar 32 holds closed a tumbler switch 36 which closes circuit to the solenoid 29 from a supply line 37 and branch line 38 through a line 39 leading to solenoid 29. This circuit causes pressure to be supplied through pipe 17 to a diaphragm 16 so that the suction valve at the head end of the compressor cylinder is held open.

Under the same conditions the solenoid 25 is energized through a holding circuit associated with a relay 40 as follows: Leading from supply line 37 through a branch line 41, tumbler 42 and line 43, then through contact bar 44 of relay 40, and branch line 45 to a branch line 46 leading to the coil of relay 40. Thus the clearance pocket at the head end of the cylinder is held open. In similar manner the solenoid 21 is energized under the same conditions to supply pressure to the diaphragm 12 and thus hold open the suction valve at the crank end of the compressor

5

cylinder, through a holding circuit for relay 47 as follows: From supply line 37 to the branch line 41, then through a tumbler 48 and branch line 49, contactor 50, and then through a branch line 51 which connects with the branch line 52 leading to the coil of relay 47.

Now if the load on the compressor be increased (in the specific example above given) up to about 10%, the pressure within the delivery line 8 will rise, and at this point the cross bar 32 will move downwardly far enough to release itself from the tumbler 36, which will thereupon swing downwardly to vertical position, opening circuit to solenoid 29 and thereby releasing the pressure on diaphragm 14 and releasing suction valve opener. Assuming that the demand on the compressor continues to increase, the pressure in delivery line 8 leading from pressure controller 9 will progressively rise and the cross bar 32 will move down past tumblers 53 and 54. The tumbler 53 thereupon temporarily closes circuit through branch lines 38 and 46 to solenoid 40, but this produces no functional change in the apparatus, since the holding circuit through this solenoid was already closed through contactor 44. Likewise when the cross bar 32 moves down past tumbler 54, circuit is closed through branch lines 41 and 52 to the relay 47, but the latter was already in closed circuit condition through its contactor 50.

During the movements last described it has been assumed that the pressure in the delivery line 8 is still below the minimum which will cause speed regulator 7 to speed up the engine 6, but as the cross bar 32 of pilot rod 31 moves down below the position where it engages tumbler 54, the pressure in pipe 8 becomes high enough to cause the engine 6 to speed up proportionately to increase in load. In the specific example given this proportionate increase in speed will occur where the pressure in line 8 rises from 8 lbs. to 15 lbs., at which time the loading of the compressor at maximum speed will be about 25%. At that point the cross bar 32 has traveled downwardly far enough to engage the tumbler 42 and thereby open the holding circuit of relay 40, the relay thereupon dropping to open position and its contact bar 55 opening circuit to the solenoid 25. The plunger 23 thereupon drops to lower position, relieving the pressure in pipe 17 and cutting out of action the clearance pocket controlled by diaphragm 15. With the engine 6 then operating at relatively high speed, the compressor thereupon tends to assume a condition of 50% loading, but this tendency produces a drop in the pressure in delivery line 8 which thereupon causes speed regulator 7 to decrease the speed of engine 6 until with this decreased speed and the clearance pocket out of operation, the loading effect is the same as it was previously when the clearance pocket was active and the engine operating at higher speed.

With the clearance pocket out of operation as above described, the cross bar 32 as the load is increased from 25% to 50% will again pass through that part of its path just beneath the point where it engages with tumbler 54, which is the part of the path where the speed of the engine is increased, i. e. that part of its path corresponding to a pressure rise in delivery line 8 between 8 and 15 lbs. per square inch. Upon further increase in load, regulator 7 will speed up the engine 6 until a loading of about 50% is attained.

Upon increase in load beyond that point, in the specific example under discussion, the pressure in delivery line 8 will rise above 15 lbs. and

6

the cross bar 32 will thereupon move downwardly until it contacts with the tumbler 48, thereby opening the holding circuit of relay 47 so that the contactor 55 thereof opens circuit between the branch line 57 extending from supply line 37 and the branch line 58 which leads to the solenoid 21. Thereupon the plunger 19 will drop to lower position venting the pressure in pipe 13 and causing the diaphragm 13 to release the suction valve opener which is located at the crank end of the cylinder. At this stage all of the step unloading devices accordingly will have been cut out of operation, and the speed of engine 6 being at or near maximum, the compressor will tend to operate at 100% of capacity, but if we assume that the load requirements are still only 50% or slightly above, the pressure in delivery line 8 will decrease instantly causing the speed regulator 7 to slow down the engine to minimum speed, where the output is about 50%. In other words, with the engine operating at minimum speed and both suction valves operating normally, the loading is about the same as with the suction valve at one end of the cylinder and the engine operating at maximum speed. As the load increases from 50% the speed of the engine will increase proportionately up to maximum load. As the load on the engine decreases from maximum the cross bar 32 of pilot plunger 31 will first rise to a position just under the point where it engages tumbler 54, decreasing the speed of the engine 6 proportionately to the decrease in load down to about 50% loading, at which time the engine will be operating at minimum speed. Thereupon tumbler 54 will be actuated to energize the relay 47 through branch line 52, the holding circuit of such relay will be closed through its contactor 50, and its contactor 56 will close circuit to the solenoid 21. This will cause plunger 19 to be lifted, supplying pressure through pipe 13 to the diaphragm 12 which holds open the suction valve at the crank end of the compressor cylinder. This adjustment, with the engine operating at low speed, would tend to reduce the compressor to an output of 25% of capacity, but if it be assumed that the load has dropped only to slightly less than 50%, the pressure in delivery line 8 will immediately rise and regulator 7 will immediately speed up the engine until the output is slightly under 50%, the cross bar 32 in this stage operating in the range just under the point of contact with tumbler 54, in which adjustment of the speed of the engine takes place.

If the load be further decreased down to about 25%, the speed of the engine will be proportionately reduced, and at about 25% of capacity the pressure in the delivery line 8 becomes so low that the cross bar 32 rises into engagement with the tumbler 53, which thereupon closes circuit through relay 40. The latter thereupon closes its holding circuit through contactor bar 44 and tumbler 42, and the contactor bar 55 of relay 40 closes circuit from the supply line 37 through a branch line 59 extending to solenoid 25. Thereupon plunger 23 is lifted to supply pressure through pipe 17 to diaphragm 16 and throw the clearance pocket into action at the head end of the cylinder. This tends to decrease the compressor down to about 10% of capacity, the engine 6 at the time running at low speed. But assuming that the load has only been decreased to about 24%, the pressure in line 8 will rise so that the engine speeds up until the output is 24% of capacity. Thereupon as the loading is decreased from 24% to 10% the pressure in line 8 will fall so that speed regulator 7 proportionately reduces the

speed of the engine down to 10% of capacity. If the load drops under 10%, the pressure in line 8 will fall to a point where cross bar 32 reaches its uppermost position as shown in Fig. 1 whereupon solenoid 29 is energized to supply pressure through pipe 15 and cause the diaphragm 14 to hold the suction valve open at the head end of the compressor, thus completely unloading the latter.

The response of the above-described apparatus to variations in load requirements is illustrated by the graphs of Fig. 2 wherein the graph in solid lines shows the percentages of loadings (i. e. the percentage of full output) to which the compressor is adjusted in response to load requirements changing progressively from zero to maximum capacity. The dotted line graph of Fig. 2 shows the percentages of loading obtained as the demand shifts from 100% down to zero. The section A of the full line graph indicates the increase of loading from 0 to slightly under 10%, which takes place in step fashion when the load requirement reaches slightly less than 10%, precise regulation at such low loads being usually unnecessary. The full line section B indicates straight line regulation as obtained when the load requirements rise up to about 25%. The peak C of the solid graph indicates the change in loading which takes place when the clearance pocket at the head end of the cylinder is cut out of action, whereupon straight line regulation is resumed throughout the section D of the solid line graph up to a load requirement of about 50%. The peak E of the solid line graph indicates the release of the suction valve opener at the crank end of the compressor cylinder, whereupon straight line regulation continues on up to 100% of capacity as indicated by section F of the solid line graph.

As the demand on the engine decreases from 100%, straight line regulation is obtained by speed regulator 7 down to about 50% or slightly under as indicated by the section G of the dotted line graph in Fig. 2. The valley H of such graph indicates the cutting into operation of the suction valve opener at the crank end of the compressor cylinder. As the demand decreases from this point, straight line regulation is obtained down to a load requirement of about 25% as indicated by the section I of the dotted line graph in Fig. 2. At this point the clearance pocket at the head of the cylinder is cut into service as indicated by the valley J of such graph. Then as load requirements continue to decrease straight line regulation is continued down to about 10% as indicated by the section K of such graph. Further decrease of load requirements will cause the suction valve opener at the head end of the compressor cylinder to be thrown into operation as indicated by the section L of such graph.

In Fig. 3 the speed of operation of the compressor is plotted against variations in load requirements between 0 and 100% capacity, and back to 0 again. When the load requirements range under about 10%, the compressor operates at the minimum speed assumed to be about 50%, as indicated by the section M of the full line graph in Fig. 3. In response to demands varying from about 10% up to about 25% the speed of the engine increases proportionately up to maximum as indicated by the section N of such graph. As the demand rises further the clearance pocket at the head end of the compressor cylinder is cut out of operation, and the speed of the compressor drops rapidly to minimum as

indicated by the section O of such graph. As the demand rises to about 50% the compressor speeds up proportionately to maximum speed as indicated by the section P of such graph. At a demand of about 50%, and at the time when the valve opener at the crank end of the compressor is cut out of operation, the compressor speed drops rapidly to a minimum as indicated by the section Q of such graph. On demand increasing up to 100% the compressor again speeds up proportionately to maximum, as indicated by the section R of the graph of Fig. 3. The above described specific operating conditions are to be considered merely as an example illustrative of the principle of operation.

From the above described method of operation it will be noted that speed control is utilized to compensate and smooth out the sudden changes in loading which otherwise would occur when the step unloading devices are cut into and out of operation. Whenever a step unloading device is cut out of operation, thus tending to produce a sudden increase in output, this is done at a stage when the compressor is operating relatively rapidly and the speed is thereupon simultaneously reduced to avoid the abrupt increase in output which otherwise would result. And whenever a step unloading device is cut into operation this is done at a stage when the compressor is operating relatively slowly, and the speed of the compressor is thereupon increased to avoid the abrupt drop in capacity which otherwise would occur. Between the settings where the step unloading devices are cut into and out of operation, speed control is utilized to produce straight line regulation and accordingly the two types of control used in combination enable straight line regulation, or substantially so, to be obtained over the combined range which the sequence affords.

While the invention has been disclosed as carried out by apparatus of the above described specific construction and mode of operation, it should be understood that many changes may be made therein without departing from the invention in its broader aspects, within the scope of the appended claims.

We claim:

1. Regulating apparatus of the class described for compressors and the like, including a regulator of the speed control type having means constructed to afford substantially straight line regulation through part of the load range of a compressor, a second regulator of the step unloading type also controlling said compressor, and control mechanism responsive to load conditions on the compressor, said mechanism including means for throwing said step regulator into and out of action as the load demand falls below and rises above a definite point and for actuating said first mentioned regulator to suppress the sudden changes in loading which a step unloader tends to produce when thrown into and out of action and produce substantially straight line regulation of said compressor in response to load changes while said second mentioned regulator is in action.

2. Regulating apparatus of the class described for compressors and the like, including a regulator of the speed control type having means constructed to afford substantially straight line regulation through part of the load range of a compressor, a second regulator of the step unloading type also controlling said compressor, and control mechanism responsive to load conditions on

the compressor for throwing said step regulator into and out of action, said control mechanism including means operating upon the occurrence of an increase in load requirements while said step regulator is in action which is greater than will be met by throwing said step regulator out of action, first to adjust said first mentioned regulator to speed increasing position while said second mentioned regulator is in action and then to throw said second mentioned regulator out of action and adjust said first mentioned regulator to speed decreasing position.

3. Regulating apparatus of the class described for compressors and the like, including a regulator of the speed control type having means constructed to afford substantially straight line regulation through part of the load range of a compressor, a second regulator of the step unloading type also controlling said compressor, and control mechanism responsive to load conditions on the compressor for throwing said step regulator into and out of action, said control mechanism including means operating upon the occurrence of a decrease in load requirements while said step regulator is out of action which causes said step regulator to be thrown into action, first to adjust said first mentioned regulator to speed decreasing position while said second mentioned regulator is out of action, then to throw said second mentioned regulator into action and adjust said first mentioned regulator to speed increasing position, and then adjust said first mentioned regulator to secure substantially straight line regulation while said second mentioned regulator is in action.

4. Regulating apparatus of the class described for compressors and the like, including a regulator of the speed control type having means constructed to afford substantially straight line regu-

lation through part of the load range of a compressor, a plurality of regulators of the step unloading type also controlling said compressor, a control mechanism responsive to load conditions on the compressor, and pilot devices actuated by said control mechanism including means for throwing said step regulators into and out of action in definite sequence upon increase and decrease in load requirements above and below definite points, said control mechanism also including means for actuating said first mentioned regulator to increase and decrease speed in coordination with the aforesaid sequence to produce substantially straight line regulation of said compressor over the load range afforded by said step unloading regulators in combination with said speed control regulator.

5. Regulating apparatus of the class described for compressors and the like, including a regulator of the speed control type having means constructed to afford straight line regulation through part of the load range of a compressor, a second regulator of the step unloading type, a pressure controller having means responsive to load conditions on the compressor and having a delivery line means controlling said speed regulator to operate between maximum and minimum limits upon the occurrence of pressure fluctuations in such delivery line between predetermined limits, and a pilot mechanism for said second mentioned regulator having means which is also responsive to pressure fluctuations in such delivery line and is arranged to coordinate the action of said regulators to produce substantially straight line regulation of said compressor through the load range when said step regulator is in action.

P. A. CONDIT.
KENNETH RUHL.