

July 31, 1962

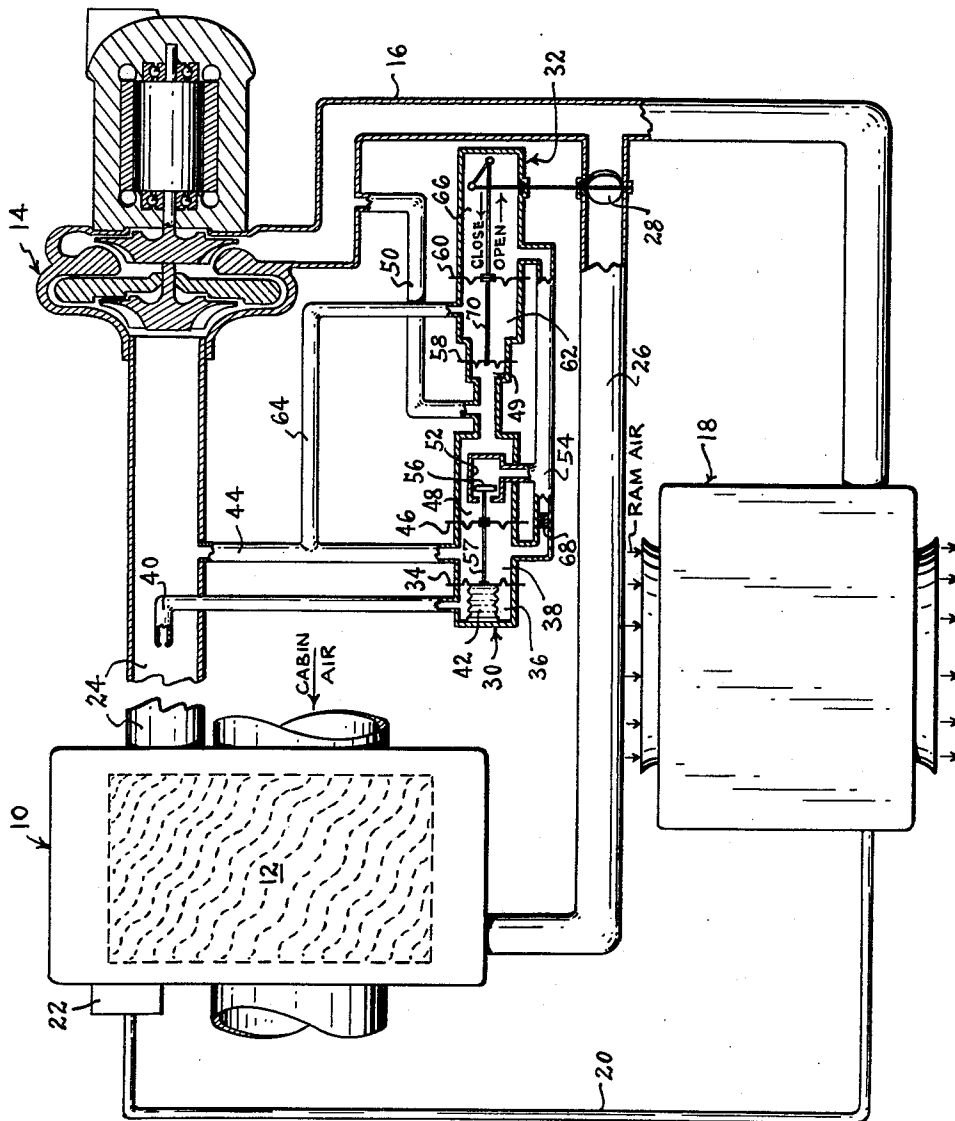
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3,047,210

COMPRESSOR SURGE CONTROL

Filed Dec. 26, 1958

4 Sheets-Sheet 1



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COMPRESSOR SURGE CONTROL

4 Sheets-Sheet 2

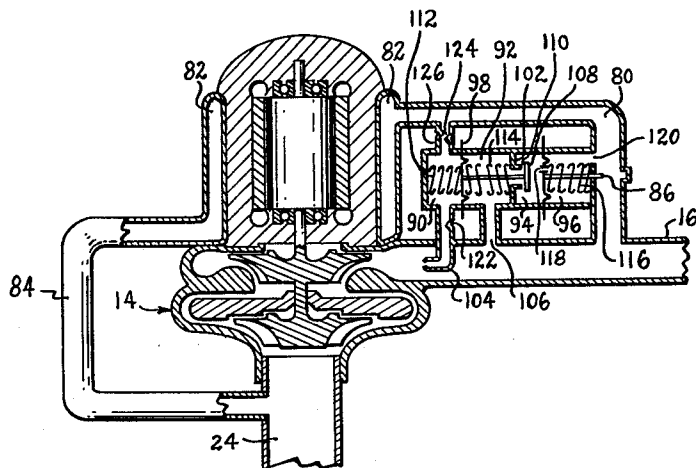


Fig. 2

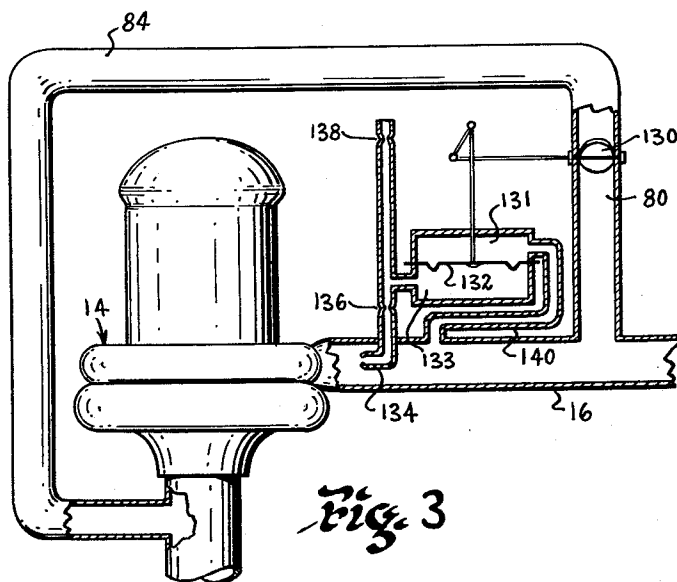


Fig. 3

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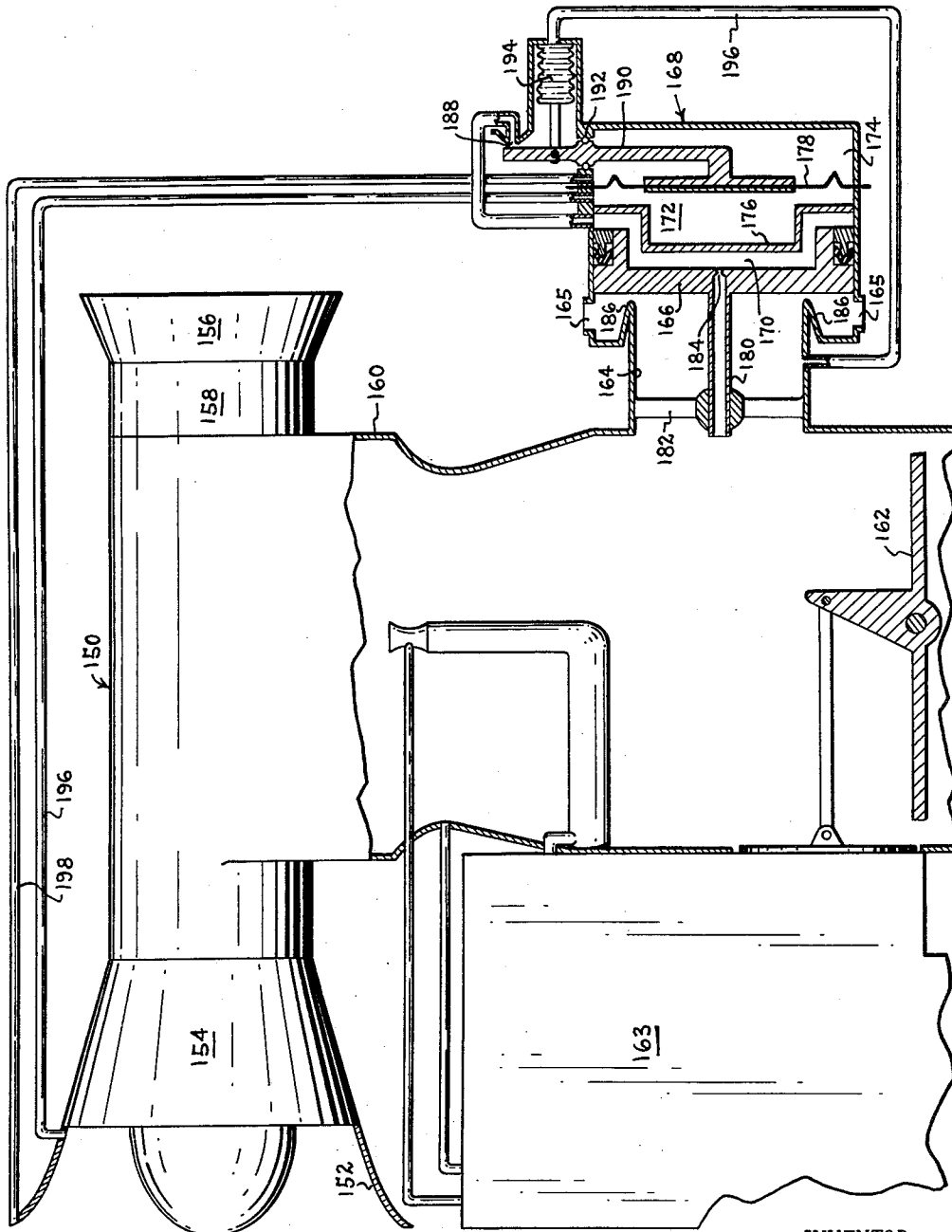
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4 Sheets-Sheet 3



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4 Sheets-Sheet 4

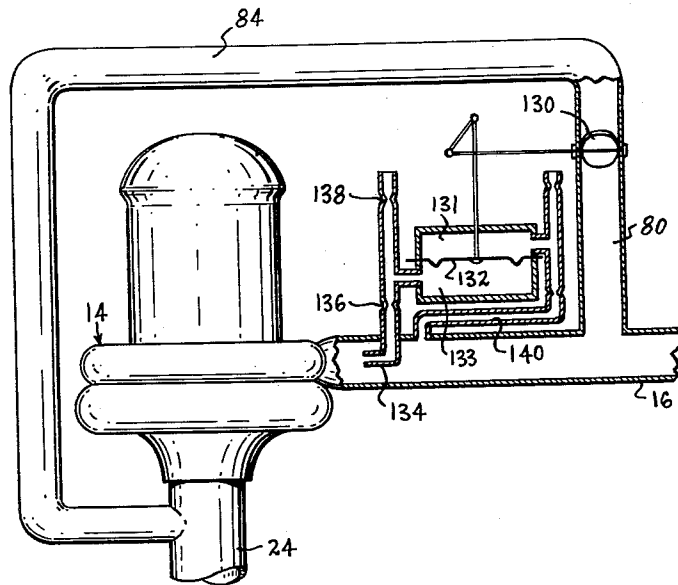


Fig. 4

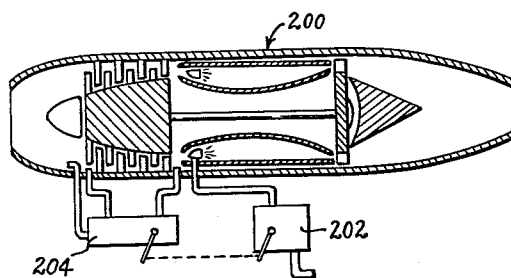


Fig. 6

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3,047,210

COMPRESSOR SURGE CONTROL

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5 Claims. (Cl. 230-115)

This invention pertains to non-positive displacement compressors for elastic fluids, and more particularly to means for controlling such compressors to prevent unstable operation under certain flow conditions of the fluids being pumped thereby.

Non-positive displacement compressors, that is, centrifugal, axial and similar types of compressors, have many advantages which make them especially desirable for use in aircraft systems. These advantages include freedom from reciprocating motion, low vibration, general mechanical simplicity, substantially little frictional wear, and very high pumping capacity for light weight units due to the high speeds at which they may be operated.

At the same time, it is a well known fact that the foregoing types of compressors are subject to surge difficulties under certain operating conditions when pumping elastic fluids, which can become so severe as to make the compressor incapable of serving its intended function, and seriously disturb the system of which the compressor is a part. The surge condition may even result in damage to or destruction of the compressor or the associated system if the condition is allowed to go unchecked. This becomes apparent from inspection of the characteristic operating curves of one of these compressors when employed in pumping an elastic fluid, i.e. a gas. In these curves, the corrected flow of the gas is plotted against the ratio of output to input pressures at constant compressor speeds, and there is obtained a definite boundary line defining areas of stable and unstable operation. This boundary line is generally referred to as the characteristic surge line of the compressor. As the flow through a compressor decreases under conditions of constant pressure ratio or constant speed, the surge line is approached. Operation at mass flow and pressure ratio conditions below that represented by such line results in unstable air flow through the compressor, and surging or pulsating pressures are produced. These may become extremely severe through sympathetic vibrational stresses and actually damage the compressor mechanically; at the very least they produce unstable or undesirable conditions in the system of which the compressor is a part.

Thus, while it is desirable to use non-positive displacement rotary compressors in many cases, it is necessary to provide means for controlling their operating conditions to avoid operation in the unstable or surge area. As will be apparent from the flow curves discussed above, control may be effected by changing one or more variables in the operation of a compressor. Three variables which may readily be controlled in the operation of a compressor of given design are its speed or r.p.m., the ratio of the outlet to inlet pressures and the corrected flow through the compressor. Various means for controlling one or more of these variables have been tried, but it is the over-all or broad purpose of this invention to accomplish this objective by means which is both mechanically and functionally simpler than that of previously known systems.

In accordance with the present invention, prevention of operation of the compressor in the unstable area is accomplished primarily by means governing the flow of gas through the compressor whereby the corrected flow is increased, or at least maintained at a proper level, to avoid surge. Since the flow in a compressor is usually

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determined by the requirement at any given time of the system of which it is a part, changing the flow in the system as a whole may not be possible or practical in some instances. However, this can be overcome in one of several ways whereby the flow through the compressor itself may be caused to be different from that of the rest of the system. That is, control of the flow in the compressor may be made more or less independent of the flow in the system as a whole, so long as it is never less than the minimum requirement of the system. This can be accomplished by bypassing some of the fluid flow from the compressor discharge directly back to the inlet, without causing such flow to go through the entire system in which the compressor is located. It can also be accomplished by simply dumping overboard or exhausting to ambient pressure or atmosphere, fluid flow in excess of that required in the system. Both these methods are utilizable in practicing the invention disclosed herein, and examples of specific systems for this are described in detail hereinafter. In another application of the invention, air flow in a turbojet is controlled indirectly by changing the fuel flow, as will be brought out more fully hereinafter.

In the present invention, the flow through a compressor is modified as required to prevent surge by control means which is adapted to produce a signal proportional to the flow through the compressor and to compare this to a reference pressure. This reference pressure is one existing in the compressor, and may be absolute inlet or discharge pressure, or combinations thereof which include pressure difference (not due to flow) across the compressor or across or between one or more stages thereof. At subsonic flow, the velocity head (ρV^2 of a moving fluid varies roughly with the corrected volume flow, therefore the above relationship may be represented mathematically by the expression

$$\frac{\rho V^2}{P}$$

where ρ is the density and V the velocity of the moving fluid, while P is the reference pressure as above defined. An indication of ρV^2 in a system may be obtained by measuring pressure difference between two points in the flow stream, as by measuring the difference between Pitot total and static pressure, or the difference between throat and either inlet or outlet pressure of a venturi, or the difference between entrance and exit pressures of a diffuser section, for example. This pressure difference (ΔP), which is due to the flow, is approximately proportional to the velocity head ρV^2 . Therefore if the ratio of corrected flow, as indicated by ΔP , to the reference pressure P is too low, as will be the case if there is too little flow through the machine, then the control means of the invention produces a signal or mechanical movement to operate a flow modifying member in the compressor system to bring about increased flow in the compressor. In order to reduce error, the control means preferably is of the integrating type rather, than the proportioning, as the former provides a more accurate and stable system.

Since ΔP is approximately proportional to the velocity head, the general equation for the whole family of surge controls disclosed herein may be stated as follows:

$$\Delta P = K_2 P_2 - K_1 P_1$$

where ΔP is pressure difference due to flow, as defined above; P_2 and P_1 are the compressor discharge and inlet pressure, respectively; and K_2 and K_1 are design constants. By selecting the proper K_1 and K_2 , the surge limits of a wide variety of compressors can be matched. If these constants are equal, the expression becomes:

$$\Delta P = K_2 P_2 - K_1 P_1$$

and a surge control device can be employed which does not require an evacuated bellows. On the other hand, if K_1 and K_2 are not equal (including the case where K_1 is zero), an evacuated bellows is required, or a pair of series orifices must be employed with the downstream orifice choked to produce a differential pressure proportional to absolute pressure. One or both of these orifices may take the form of a venturi, as will appear hereinafter.

It is a further object of the invention to employ discharge pressure from a compressor to supply the moving force for controlling the setting of a flow control member located either in the by-pass from the compressor discharge to its intake, or in a port in the compressor discharge line which allows the compressor to discharge to ambient atmospheric pressure, or in some other location, as in the fuel control of a jet engine. Accordingly surge control devices are provided which act to admit compressor discharge or some other available source of pressure to a servo mechanism to actuate the aforesaid flow control member.

The foregoing concept is capable of practical application in many different types of centrifugal, axial and similar non-positive displacement compressor systems. A number of typical systems are shown in the accompanying drawings and are described in some detail hereinafter.

In the drawings,

FIG. 1 is a schematic representation of a vapor cycle refrigerating system, as used in an aircraft air conditioning system, employing a surge control incorporating the invention;

FIG. 2 is a fragmentary schematic view, illustrating a modification of the surge control means of FIG. 1, the control in this instance being located in the compressor discharge;

FIGS. 3 and 4 are, similarly, partial schematic views of other modified forms of surge controls useful in compressor systems;

FIG. 5 is a partial schematic view of a ground unit supplying compressed air for starting turbojet engines in aircraft; and

FIG. 6 is a simple block diagram of a control connected into the fuel system of a jet engine.

Referring to FIG. 1 of the drawings, a closed vapor cycle refrigerating system of a type useful in air conditioning aircraft cabins is shown schematically. The system comprises an evaporator 10 having a core 12 over which cabin air is passed in order to cool it. The cooling is supplied by the evaporation of condensed Freon, for example, within the core. The Freon is compressed by a centrifugal compressor indicated generally at 14, and this is delivered by a duct 16 to a condenser 18. Ram air passes over the core of the condenser and the Freon is condensed to a liquid within the core. The Freon is delivered to evaporator 10 through a duct 20, the admission of the Freon to the evaporator being controlled by a thermostatic expansion valve 22 in well known manner. The evaporated Freon, upon leaving the evaporator, passes through intake duct 24 to the compressor, where the cycle is repeated.

As here shown, compressor 14 is driven by a constant speed motor or other suitable device, so that, regardless of the demand for Freon at the evaporator, the compressor tries to maintain a constant output whenever it is running. It is undesirable for several reasons to operate the compressor for frequent periods of short duration, but on the other hand, if the Freon demand at the evaporator is at or near a minimum, and consequently the expansion valve allows little Freon to pass through the system, a condition of unstable operation will result if the flow drops to a point where the compressor is operating at or below conditions represented by its characteristic surge line curve.

In order to correct this condition, the system illustrated in FIG. 1 incorporates a by-pass duct 26 leading from

the compressor discharge to the evaporator core, without going through condenser 18 and expansion valve 22, thus returning the Freon substantially directly to the compressor intake. This by-pass allows uncondensed Freon to be circulated back through the compressor, independently of the demand determined by expansion valve 22, and the flow of vapor through the compressor can thus be maintained at a level sufficient to avoid surging. The recirculated Freon flow is controlled by a flow control member 28, represented here as a butterfly valve for purposes of illustration. Valve 28 is normally closed, as will be explained presently, and is opened to correct for too low a Freon flow through the compressor by surge control means 30. Servo actuator 32 is connected to and operated by control 30.

Surge control 30 comprises a diaphragm chamber or housing having a number of diaphragms therein dividing the interior into a series of compartments. A first diaphragm 34 divides the left portion (as viewed in FIG. 1) of the housing into compartments 36 and 38. Compartment 36 is connected by suitable duct means to a total pressure head 40 facing upstream in the compressor inlet duct 24. An evacuated bellows 42 in compartment 36 biases diaphragm 34 to the left. Compartment 38, in turn, is subjected to static pressure at the compressor intake by means of a duct 44, so that diaphragm 34 is acted upon by the difference between the total and static pressures at the compressor intake. This difference is modified by the bias introduced by evacuated bellows 42.

A second diaphragm 46 separates compartment 38 from compartment 48 in control 30. This latter compartment is connected to compressor discharge pressure by a duct 50. Within compartment 48 there is located a valve housing 52 which communicates with a passage 54, the purpose of which will be explained presently. A valve 56, located within housing 52, seats internally thereof and is connected by a stem 57 to diaphragms 46 and 34, respectively. Under no-flow conditions, valve 56 is open slightly and is opened further as the pressure differential across diaphragm 34 increases. Diaphragm 34 accordingly moves as the pressure difference across it changes, which movement constitutes a signal proportional to the flow in duct 24. Diaphragm 46, it will be seen, is subject in turn to compressor inlet and discharge pressures, and thus provides an indication of pressure rise across the compressor. Diaphragm 46 normally opposes the action of diaphragm 34, tending to close valve 56.

Control 30 therefore provides a comparison of flow through the compressor to pressure rise (as modified by bellows 42) across the compressor. If the forces acting on the two diaphragms 34 and 46 are not equal, valve 56 is moved appropriately, and this produces a correction signal. In the illustration in FIG. 1, the correction signal is applied through servo actuator 32 associated with control 30, as will now be described. The bellows 42, in addition to modifying the pressure rise across the compressor, also provides a bias to valve stem 57 which is proportional to absolute pressure.

Servo 32 has a pair of diaphragms 58, 60, which divide the housing into compartments. At the left as seen in FIG. 1, a first compartment 49 is in direct communication with compartment 48 of control 30, and with duct 50 carrying discharge pressure. Diaphragms 58 and 60 define between them a second compartment 62, which is vented to compressor intake static pressure by duct 64 tapping into duct 44. As here shown, diaphragm 58 has an effective area one-half that of diaphragm 60. The third compartment 66 communicates with valve housing 52 of control 30 by means of duct 54. The latter also connects into compartment 38 through a restricting orifice 68. A stem 70 interconnects diaphragms 58 and 60 and actuates by-pass flow valve 28 through suitable linkage.

As the compressor starts up, there is initially little fluid flow in duct 24 and consequently little differential pressure across diaphragm 34. Also the discharge to intake pres-

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sure ratio is low so that there is little differential pressure across diaphragm 46. As the flow of Freon through the compressor builds up, a differential pressure is produced across diaphragm 34, opposing the bias of bellows 42 and tending to cause valve 56 to open. At the same time, however, there is a differential pressure developed across diaphragm 46, the effect of which tends to close valve 56. An equilibrium condition is established which determines the position of valve 56. If the ΔP across diaphragm 34 is too high or too low in comparison to the differential across diaphragm 46, valve 56 will be repositioned accordingly. This will change the pressure in compartment 66 of servo actuator 32 and cause rotation of valve 28 until the system returns to equilibrium.

With increase in the discharge pressure of the compressor, the differential pressure across diaphragm 34 required to open valve 56 becomes greater, which corresponds with the fact that at higher discharge pressures, the surge limited flow is higher.

If the normal flow through compressor 14 is reduced, owing to a minimum cooling requirement at the evaporator, by-pass flow control valve 28 opens to maintain the flow through the compressor. This is accomplished as follows. As the flow through the compressor decreases, the pressure differential across diaphragm 34 likewise decreases. Under the assumed condition, however, the differential across diaphragm 46 remains relatively high and this causes valve 56 to move toward its closed position. Pressure in compartment 66 bleeds off through restriction 68 faster than it is replaced by discharge pressure entering through valve 56. The result is a net force acting on diaphragms 58 and 60 to move stem 70 to the right in FIG. 1, which corresponds to opening of valve 28. Thus Freon vapor is allowed to by-pass from the compressor discharge back to its intake, thereby maintaining a sufficient flow to prevent surge.

In order to provide greater accuracy and stability, an integrating control as shown, rather than a proportioning type, is preferred. In the illustrated system, large correcting movement of valve 28 is made initially, therefore, and the valve is continuously reset to provide progressively smaller flow corrections as the system comes back to equilibrium.

A modification of the control described above is illustrated in FIG. 2 which differs from the former in that ΔP is sensed in the compressor discharge instead of the intake. In this case, a by-pass from compressor discharge to intake is provided by duct 80, motor housing 82 and duct 84, the by-passed fluid being there mixed with other refrigerant fluid which was passed through the motor for cooling purposes. A flow control in the form of a slide valve 86 is located in duct 80, and in normal operation of the system this remains closed. The surge control in this case comprises a housing 88 which is divided into compartments 90, 92, 94 and 96, respectively, by diaphragms 98, 100, and an intermediate fixed partition 102.

Total discharge pressure is picked up by total pressure head 104 and static pressure by duct 106. These pressures are supplied to compartments 90 and 92 and act upon diaphragm 98. The latter is connected to a poppet valve 108 cooperating with a seat 110 provided in partition 102. Valve 108 serves to control communication between compartments 92 and 94. As here shown, diaphragm 98 is biased by opposing springs 112, 114, the net effect of which is to tend to hold valve 108 on its seat, thus shutting off communication between compartments 92 and 94.

The use of spring biasing on diaphragm 98, as described above, represents a compromise since, generally, springs will introduce errors into the system. However, in some instances a spring may be used for practical purposes where the compressor is working over a limited operational range. In such case, the spring may be used in lieu of extra bellows, diaphragms or orifices. A spring

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may also be used in cases where the pressure level is such that design requirements dictate an inconveniently small size diaphragm. A large diaphragm may then be used in conjunction with a compensating spring to provide the equivalent arrangement.

Diaphragm 100, which acts as a servo positioning slide valve 86 is normally biased by a spring 116 toward open position. A bleed 118 is provided in diaphragm 100, and port 120 in end wall 122 of housing 88 at the compressor inlet side of slide valve 86 vents compartment 96 to compressor intake pressure.

It will be seen that if poppet valve 108 is opened by diaphragm 98, pressure from the compressor discharge duct 16 can flow through duct 106, compartment 92, valve seat 110 into compartment 94 where it acts on diaphragm 100, moving the latter in a direction to close the by-pass valve 86. This condition is maintained so long as the flow through the compressor is sufficient to avoid surge conditions. If the normal flow is reduced for any reason, as in the preceding discussion, the differential pressure acting across diaphragm 98 is reduced, and poppet valve 108 assumes a proportionately more closed position. This reduces the force on diaphragm 100, and the latter moves upward (in the illustration in FIG. 2), allowing fluid to flow through the by-pass to the compressor inlet.

The differential pressure acting on diaphragm 98 is compared to absolute discharge pressure which is produced by means of series restrictions 122 and 124 in total pressure duct 104 and a downstream duct 126, respectively. Duct 126 is connected between compartment 90 and by-pass 80. Restriction 124 is choked sufficiently such that for all except very low pressure ratios, the velocity through it is in the sonic range, in which event the pressure drop across orifice 122 becomes proportional to absolute pressure.

Similar arrangements are shown in FIGS. 3 and 4. The form of surge controls here illustrated are simplified in that they do not employ servo actuators for positioning the respective by-pass flow control valves 130 in each instance. In FIG. 3, the control employs a diaphragm 132 which is subjected to total and static pressures in the discharge line 16, modified by a pressure increment proportional to absolute discharge pressure. This is produced by connecting compartment 133 at one side of the diaphragm to total pressure duct 134, one end of which picks up total pressure at the compressor discharge, while the other end is open to ambient atmospheric pressure. Upstream and downstream restrictions 136, 138, respectively, are provided in this duct, and compartment 133 is connected to the duct intermediate the restrictions. The downstream restriction is designed to be choked for most ratios of compressor discharge to inlet pressures to produce a reference or absolute pressure between restrictions 136, 138. At the opposite side of diaphragm 132, compartment 131 is connected by line 140 to the static pressure pick off in duct 16. The differential pressure or diaphragm 132 in this case is ΔP (due to flow) minus an increment proportional to absolute discharge pressure.

The arrangement in FIG. 4 is substantially the same, except that a closer match to the actual surge line of the particular compressor concerned may be obtained for the surge control device by incorporating series restrictions in the static pressure duct 140 as well as in the total pressure duct 134.

Various other combinations may be employed for matching the surge line of particular compressors, particularly at conditions of low pressure ratios. For example, an orifice which becomes unchoked at low pressure ratios may produce a closer conformation of surge control operation to the surge line of a given compressor. In another case, a closer match may be obtained by employing two lines incorporating series orifices or venturis which become unchoked at different points or which discharge to different pressure levels.

In FIG. 5, the system illustrated is that of an air com-

pressor used as a ground starting unit for turbojet engines. The device employed for compressing air is itself a small turbojet engine 150, having an intake duct 152 in which is located a compressor 154. The compressor is driven in conventional manner through a shaft, not shown, connected to a turbine 156 located in the rear part of the engine. Fuel metered into a combustion chamber 158 is mixed with air from the compressor section and is ignited so that the expansion of the hot gases of combustion passing over the turbine blades cause the turbine to rotate. The compressor has excess pumping capacity beyond that needed to compress air for combustion in the combustor, and this excess capacity is tapped as a source of high pressure air for external use, as noted above. The excess air is taken off through duct 160, and the flow of air therethrough is adjusted by a throttle or butterfly valve 162 by control member 163. Under stand-by conditions where engine 150 is running but throttle 162 is substantially closed, the capacity of compressor 154 greatly exceeds the demand of combustor 158 and turbine 156, with the result that unstable operation of the compressor can arise. To compensate for this, a dump port 164 having outlets 165 is provided upstream from throttle 162 in duct 160. A valve member, in this instance a piston 166, co-operates with port 164 to allow more or less air to pass out through the port. If the mass air flow through engine 150 drops off, owing to a closed throttle condition in duct 160, piston 166 moves away from port 164 to allow more air to be dumped, and the flow through compressor 154 is thus maintained at a stable level.

Control of piston 166 is effected in the following manner. The piston is located in a surge control housing 168. The interior of this housing is divided into three compartments 170, 172 and 174. A permanent partition 176 in the housing forms compartment 170 with the piston; compartments 172 and 174 are formed by a diaphragm 178. Piston 166 has a tubular sleeve 180 secured to one face and projecting axially thereof through port 164. A bearing for the inner end of this sleeve is provided at the center of a spider 182. Pressure in duct 160 is transmitted through sleeve 180 and through a restricted opening 184 in piston 166 to compartment 170. Owing to the fact that seat 186 of port 164 defines a smaller effective area on the left face of the piston (as viewed in FIG. 5) than the effective area on the piston within compartment 170, the same pressure on each side of piston 166 will result in a net force acting to move the piston to the left, thus closing port 164.

A variable bleed 188 is connected by suitable duct means to chamber 170, whereby the pressure in the chamber is controlled in accordance with the restriction at the orifice of bleed 188. Such restriction is imposed by a lever 190 which is secured to diaphragm 178 and pivots about a sealed fulcrum 192 in housing 168. Pivotal movement of the lever is also controlled by a pressure operated bellows 194 which is connected by a duct 196 leading to port 164. Diaphragm 178 is subjected to a differential pressure through connection of its associated compartments 172, 174, to static and total pressures, respectively, at the compressor intake 152. Suitable ducts 196, 198, are provided for this purpose.

At a condition of normal air flow through engine 150, corresponding to an open setting of throttle 162, substantial differential exists between the total pressure in duct 198 and static pressure in duct 196, whereby diaphragm 178 tends to force lever 190 to close bleed 188. Since this prevents escape of pressure from chamber 170, piston 166 is caused to move toward its seat 186, thus shutting off the air dumped overboard. Pressure in bellows 194 opposes the action of diaphragm 178 produced by the existence of a differential across the diaphragm, so that as the differential decreases (which occurs when throttle 162 is closed), lever 190 is moved away from bleed 188, decreasing the flow restriction. The pressure within chamber 170 thus bleeds off faster than it is replaced by

flow through opening 184 in piston 166, and the latter is therefore moved away from seat 186, allowing air to dump overboard.

In place of the separate total and static pressure taps in the compressor intake, a Pitot tube could be used; or Pitot total and static pressures in the compressor discharge can be employed. Alternatively, two static taps located at spaced points in the diffuser section of the compressor could be used to feed ducts 196, 198. A venturi placed in the flow can likewise be used, and taps taken at the inlet or outlet and at the throat. The foregoing holds true also for the several other systems which have been described above.

The surge control system herein described is applicable also to controlling the fuel flow to the combustor section of a jet engine. In this case, the control senses air flow through the engine and compares this to a reference pressure so that should the comparison indicate incipient surge conditions, the fuel flow will be decreased. Decreasing fuel flow results in reduced heat input to the engine, therefore air flow will increase to prevent surge.

Such a system is shown in block diagram in FIG. 6. A turbojet engine 200 is provided with a fuel control device 202 which regulates the admission of fuel to the combustor section of the engine according to throttle setting, altitude, etc., in known manner. In order to prevent a surge condition arising in the compressor section of the engine, surge control 204 is provided. This control may be like that employed in the specifically illustrated forms shown in FIGS. 1 and 2, for example, whereby the control measures corrected flow through the compressor, compares this to a reference pressure as defined hereinabove, and applies a correcting action to the fuel flow regulator 202 if the air flow through the compressor section is approaching the surge condition.

The invention is illustrated but not limited by the foregoing specific examples. Modifications will be apparent to those skilled in the art and those coming within the scope or equivalency range of the following claims are therefore intended to be covered herein.

What is claimed is:

1. In combination,
 - (a) a nonpositive displacement compressor for elastic fluids,
 - (b) a movable control member,
 - (c) means for adjusting the flow of fluid through the compressor independently of demand in response to movements of said control member,
 - (d) a first pressure sensing means for sensing a pressure differential proportional to fluid flow through the compressor,
 - (e) pressure responsive means secured to said movable control member, said means being in communication with said first pressure sensing means and responsive to said pressure differential for applying the resultant force of said pressure differential to the control member,
 - (f) a second pressure sensing means for sensing a static pressure differential in the compressor,
 - (g) said second pressure sensing means having means for sensing a higher static pressure at one point in the compressor and means for sensing a lower static pressure at another point in the compressor,
 - (h) a second pressure responsive means secured to said movable control member, said means being in communication with said second pressure sensing means and responsive to said static pressure differential for applying the resultant force of said static pressure differential to the control member in opposition to the resultant force of said first mentioned pressure differential,
 - (i) said second pressure responsive means being modified in accord with predetermined design constants based on the surge limit curve of the compressor to

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vary the resultant force of each of said static pressures sensed in unequal degree, and

(j) means for exerting a bias on said movable control member in proportion to an absolute reference pressure.

2. The combination as set forth in claim 1 having in addition, a by-pass from the compressor discharge to its intake and a flow control member in said by-pass for blocking the flow of fluid therethrough, and wherein said movable control member positions said flow control member to vary the flow of fluid through said by-pass in response to the resultant effect of said first and second pressure responsive means, said control member acting to open said by-pass control member as the ratio between the pressures sensed by said first and second pressure sensing means approach a value corresponding to the surge limit of the compressor.

3. The combination as set forth in claim 1 wherein said second pressure sensing means senses the difference between the inlet and outlet static pressures of the compressor.

4. The combination as set forth in claim 3 wherein said first pressure sensing means senses the differential between compressor inlet total and static pressures.

5. The combination as set forth in claim 2 wherein

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said first and second pressure responsive means comprise a pair of interconnected diaphragms, one of said diaphragms being subjected to a pressure differential proportional to fluid flow through the compressor, the other of said pair of diaphragms being subjected to a difference in static pressures across the compressor, and a fluid actuated servo mechanism for moving the control member in the by-pass, said movable control member being actuated by the resultant effect of the pressure differences sensed by said diaphragms to control the admission of fluid to said servo mechanism.

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UNITED STATES PATENT OFFICE
CERTIFICATE OF CORRECTION

Patent No. 3,047,210

July 31, 1962

Stanley G. Best

It is hereby certified that error appears in the above numbered patent requiring correction and that the said Letters Patent should read as corrected below.

Column 3, line 1, the equation should appear as shown below instead of as in the patent:

$$\Delta P = K(P_2 - P_1)$$

column 6, line 3, for "large" read -- larger --.

Signed and sealed this 11th day of December 1962.

(SEAL)

Attest:

ERNEST W. SWIDER
Attesting Officer

DAVID L. LADD
Commissioner of Patents