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 [21] Appl. No. **825,839**
 [22] Filed **May 19, 1969**
 [45] Patented **June 28, 1971**

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[54] **BALANCED FLUID FLOW CONTROL APPARATUS**
16 Claims, 32 Drawing Figs.

[52] U.S. Cl. 239/76,
 239/75, 239/127, 123/119

[51] Int. Cl. B05b 15/00

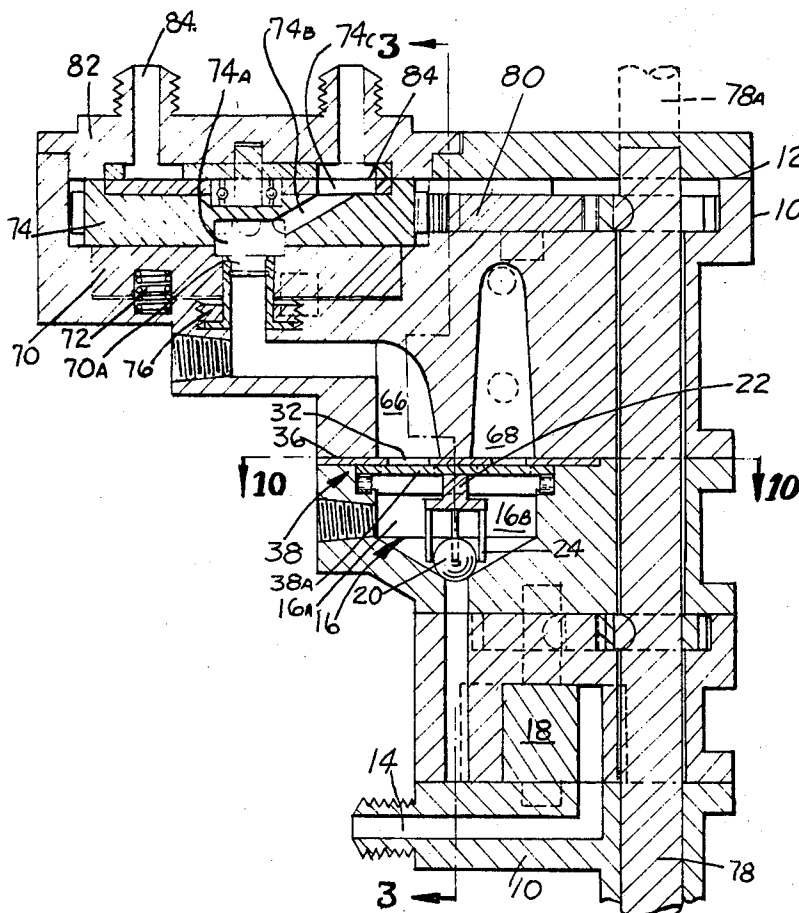
[50] Field of Search 139/75, 76,
 127, 95, 96; 123/119, 139.1; 137/115, 118

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ABSTRACT: A balanced fluid control apparatus for dividing sources of input fluid supply into at least two variable and controlled parts at least one of which is discharged at predetermined rates of flow to utilizing mechanisms, devices or processes, the rates of flow being controlled to range from minima to maxima, the latter including, in the extreme, the total input fluid as the discharge.



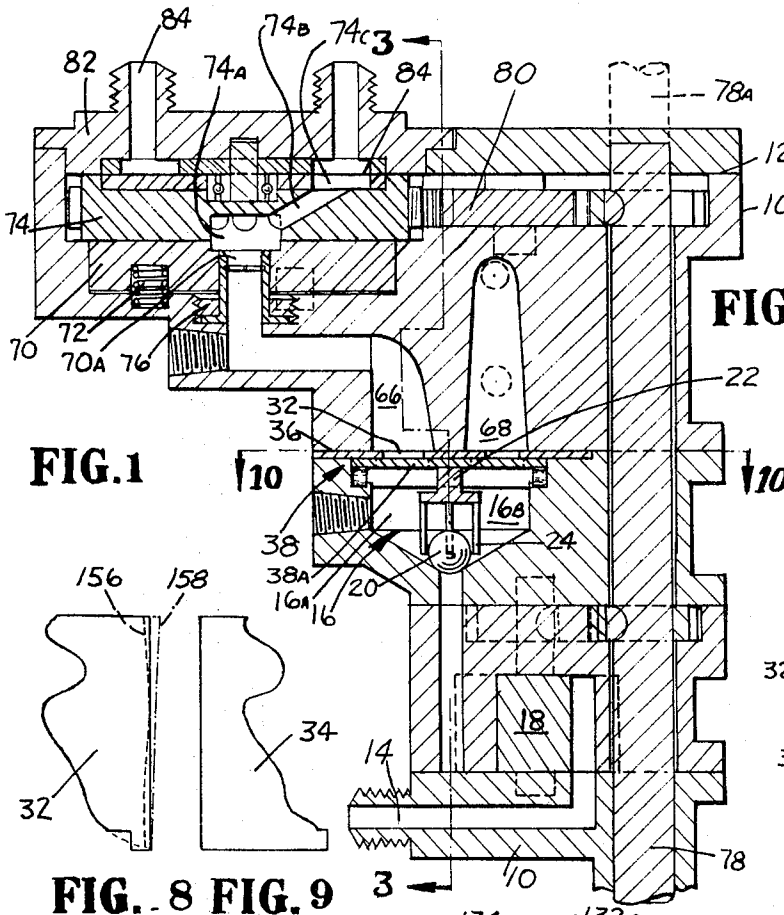


FIG. 1

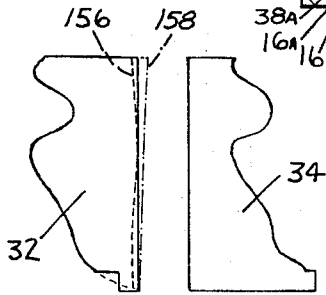


FIG. 8 FIG. 9

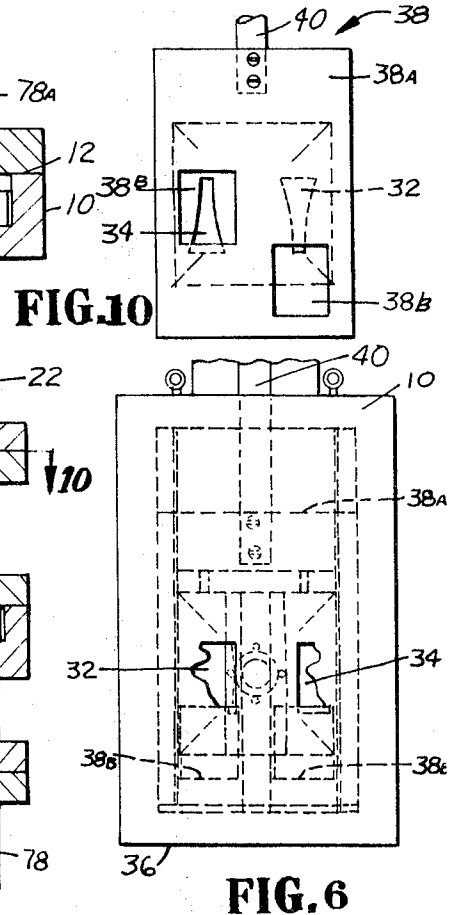


FIG. 10

FIG. 6

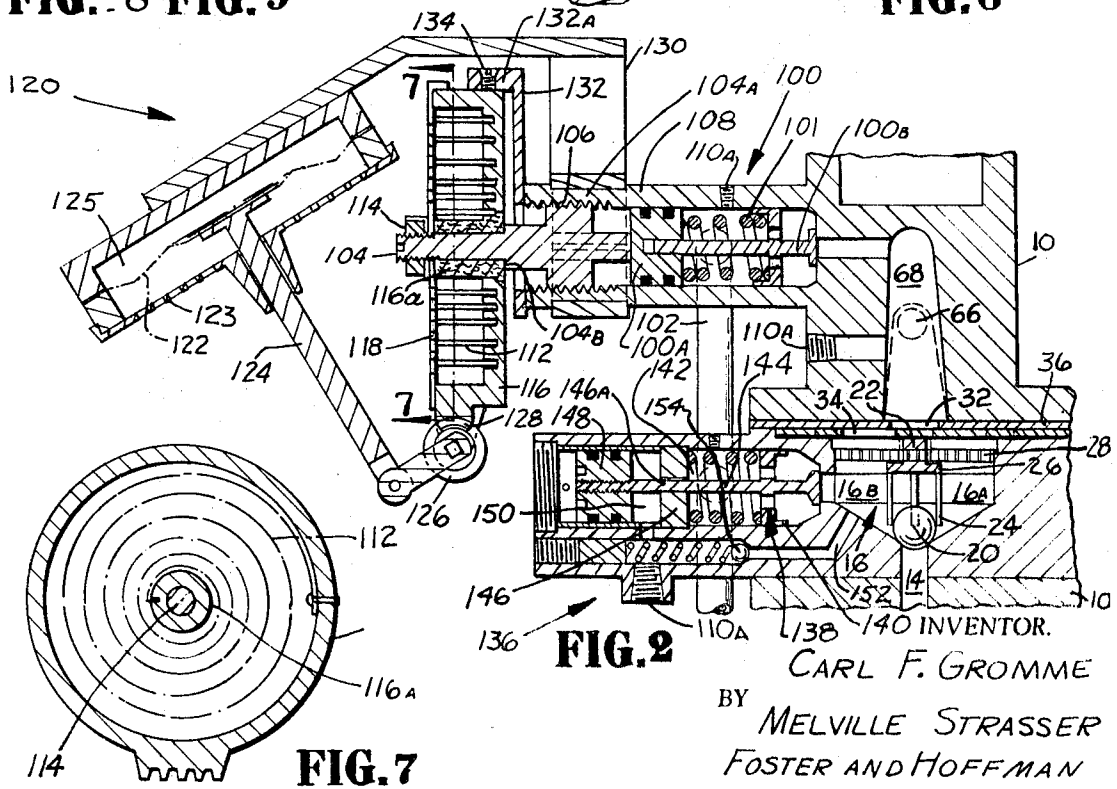


FIG. 2

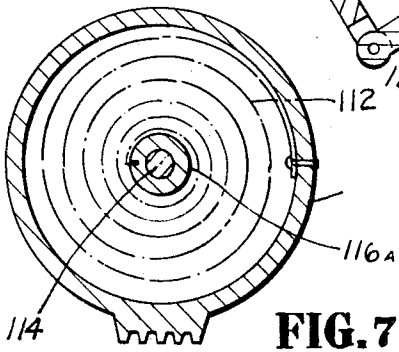
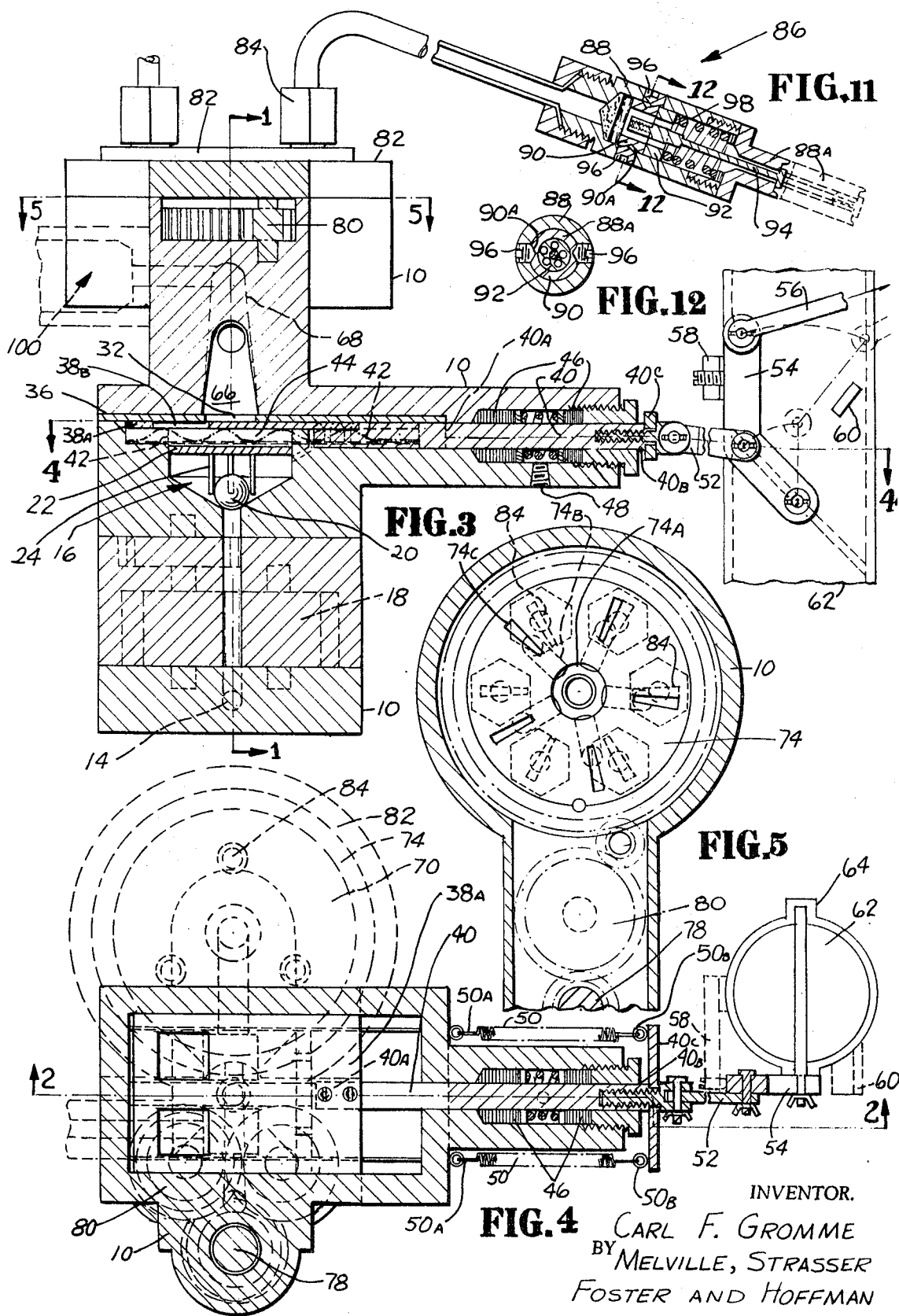
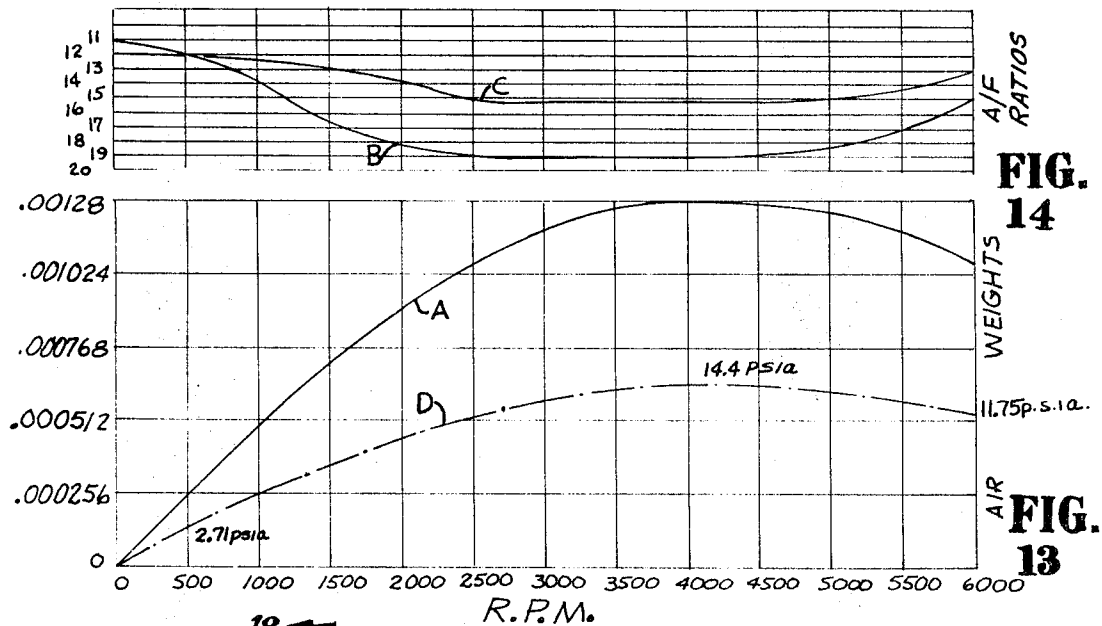
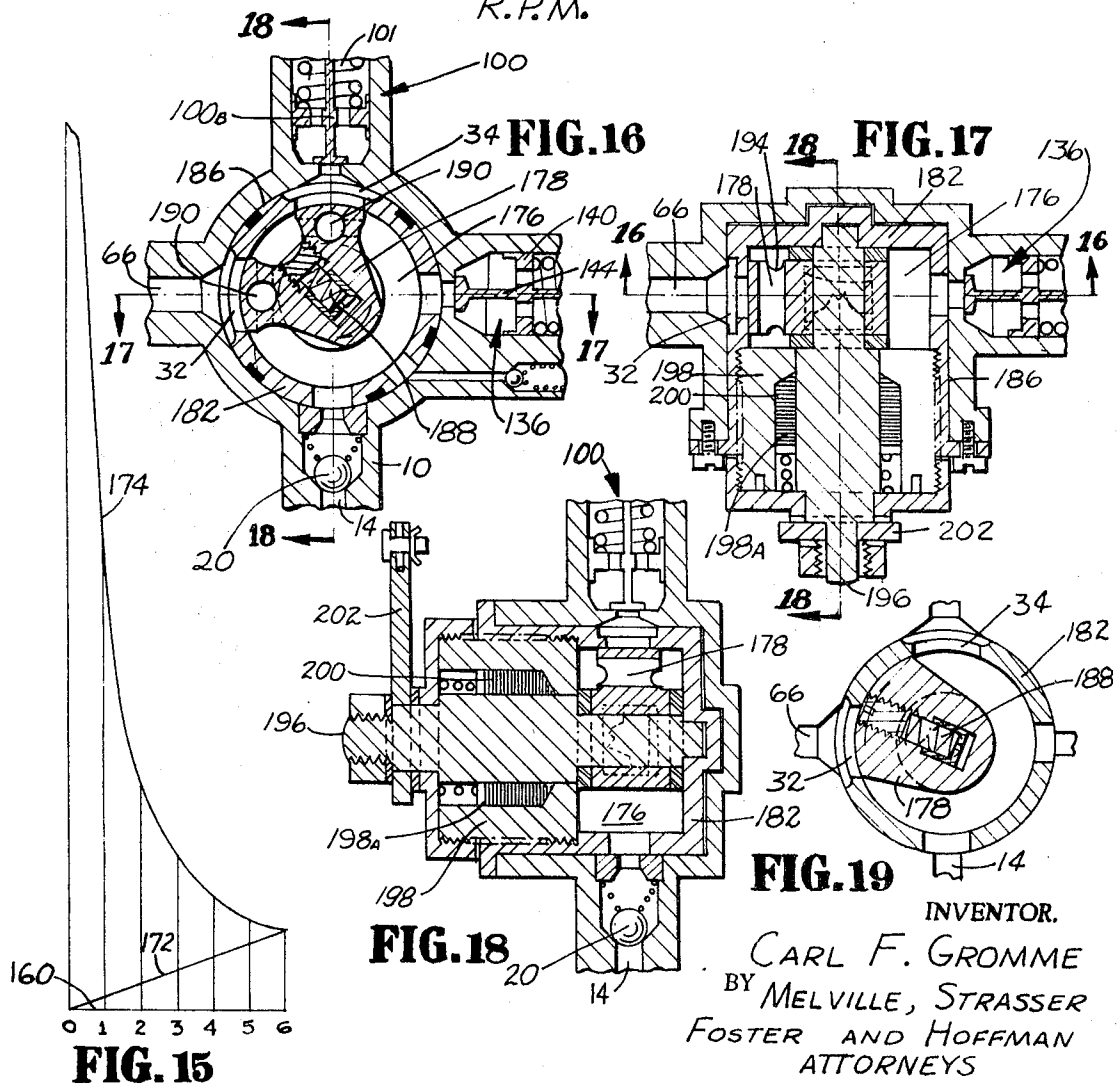
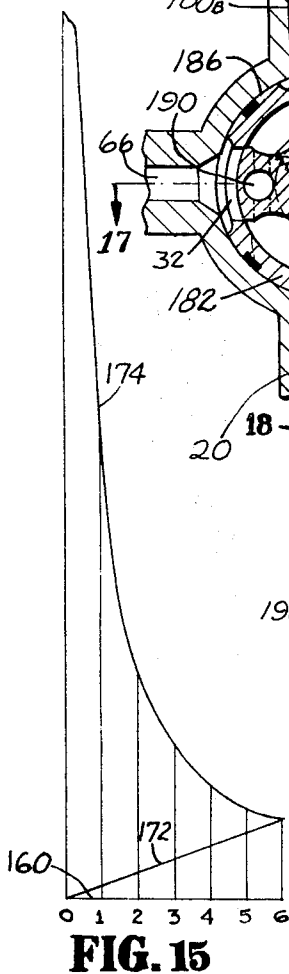


FIG. 7

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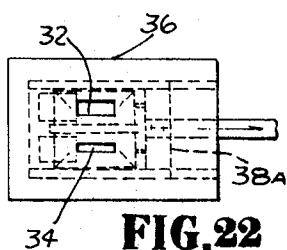


FIG. 22

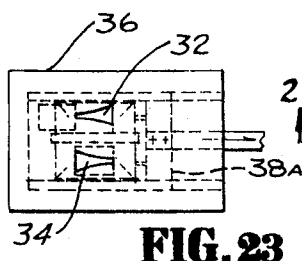


FIG. 23

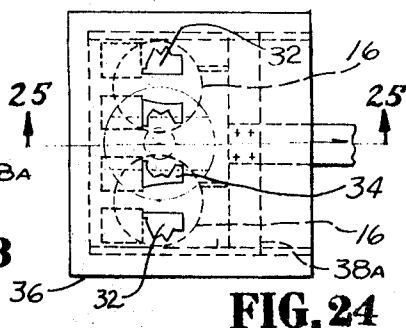


FIG. 24

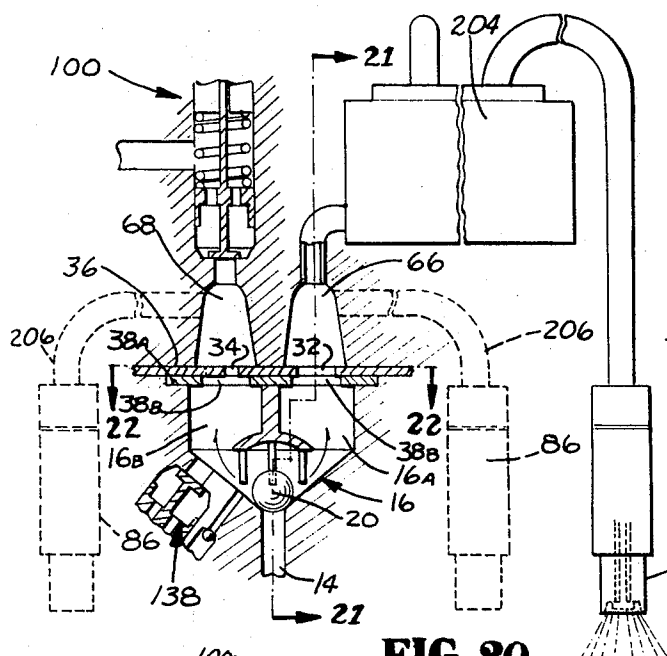


FIG. 20

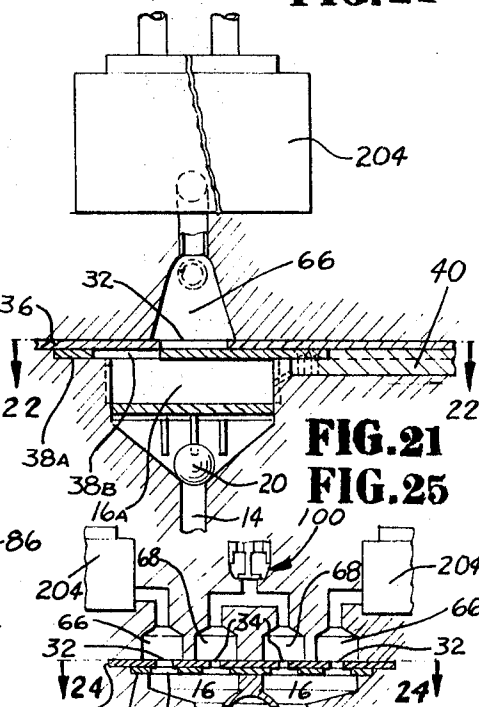


FIG. 21
FIG. 25

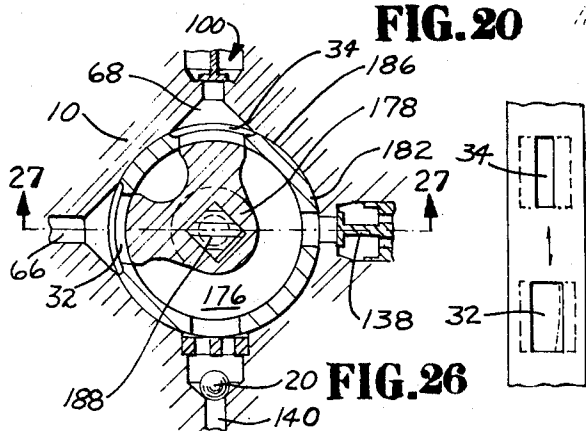


FIG. 26

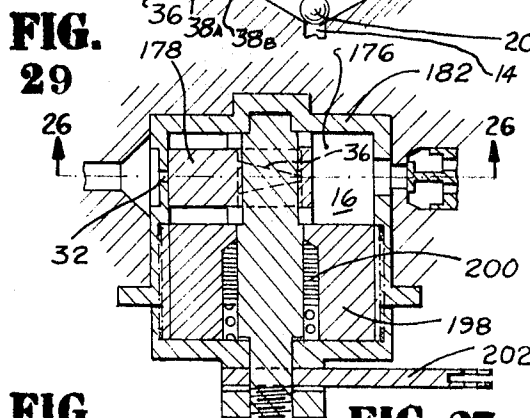


FIG. 27

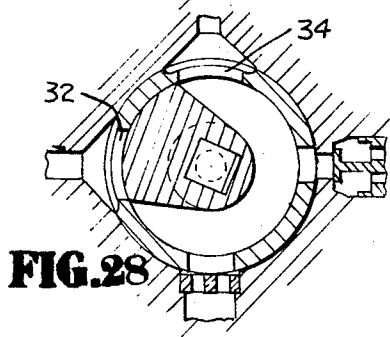


FIG. 28

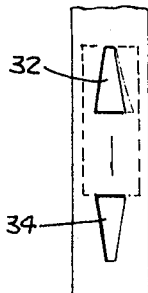


FIG. 29
FIG. 30

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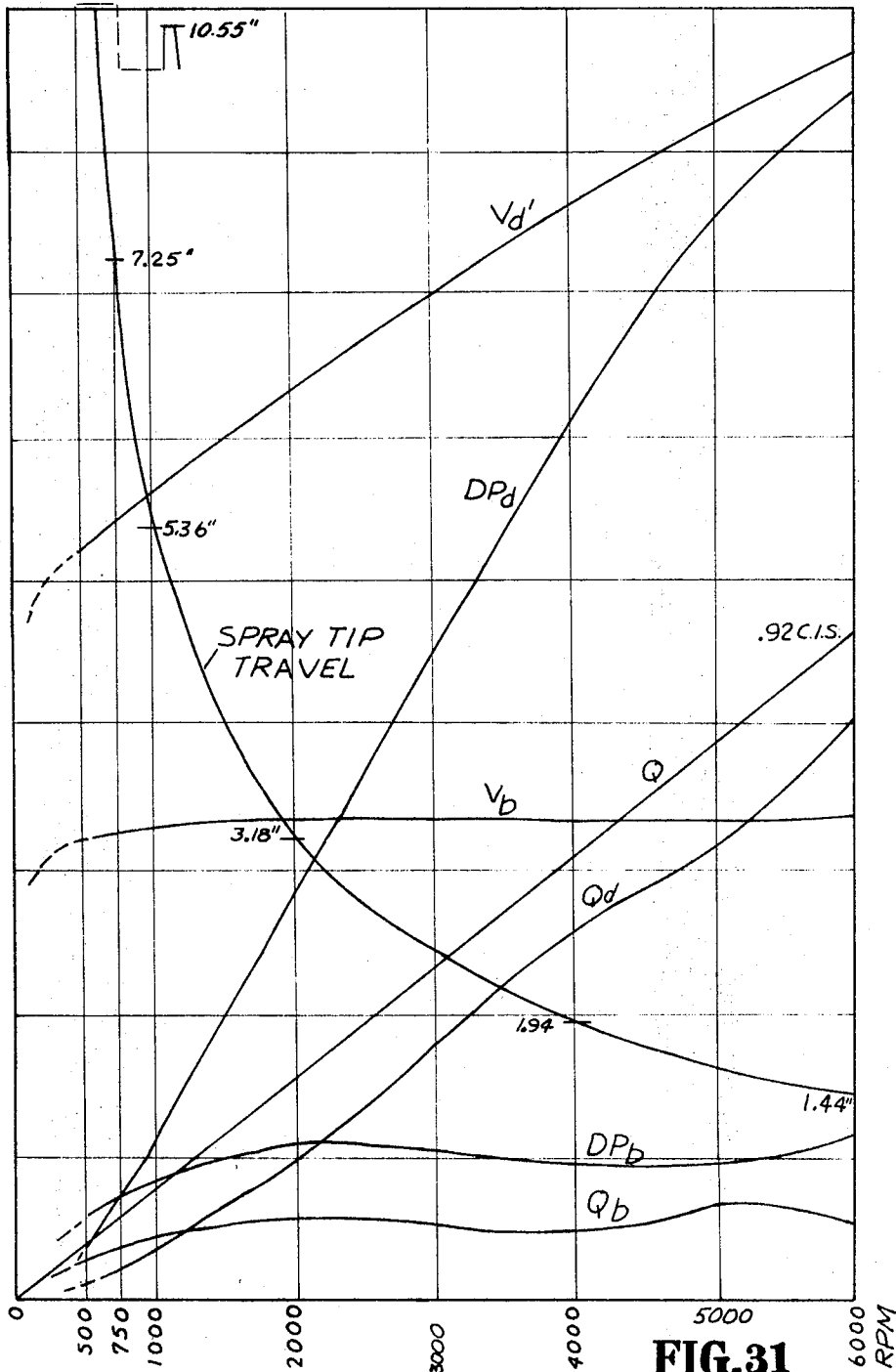


FIG.31

↑
 SCALES
 $Q-Q_d-Q_b$ - 0 - .2 CU. IN. PER SEC.
 V_d-V_b - 0 - 50 IN. PER SEC.
 DP_d-DP_b - 0 - 5 P.S.I.
 SPRAY TIP - 0 - 1 IN. OF TRAVEL

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TABLE I PART ONE

A/F RATIO	RPM	W_a	L	t	A_d	A_b	INCREM- ENTS A_d	INCREM- ENTS A_b	DIVISION FACTOR
11:1	500	.0002406	.0766	.040	.002099	.005567	.05204	.1391	.04
12.15:1	750	.0003635	.115	.0266	.004309	.00719	.1105	.0811	.02
13.2:1	1000	.0004864	.1533	.020	.007076	.008257	.1383	.05325	.02
17.9:1	2000	.0089344	.3066	.010	.019168	.011498	.3022	.0810	.04
19:1	4000	.001280	.6133	.050	.051742	.009591	.4071	.02828	.08
15:1	6000	.001044	.92	.0033	.080264	.011736	.03562	.0268	.08

TABLE I PART TWO

A/F RATIO	RPM	$A'd$	$A'b$	V_d	V_b	DP_d	DP_b	P_j	P	Z
11:1	500	.000079	.000347	263.7	160.4	1.84	2.83	22.1	26.77	10.55
12.15:1	750	.000153	.000438	272.2	163.85	3.66	3.59	22.1	29.35	7.25
13.2:1	1000	.000253	.000503	279.0	164.0	5.84	4.11	22.1	31.05	5.58
17.9:1	2000	.000602	.000683	318.4	168.28	13.93	5.6	22.1	41.63	3.18
19:1	4000	.001332	.0005757	388.45	166.59	30.85	4.71	22.1	57.66	1.94
15:1	6000	.00183	.0006946	433.15	169.1	42.88	5.68	22.1	70.76	1.44

FIG. 32

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BALANCED FLUID FLOW CONTROL APPARATUS

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to a new and improved construction for a fluid flow control system and more particularly to a new and improved balanced fluid flow control apparatus which will controllably and variably divide one or more inflowing supplies of inviscid fluids, viscous fluids, or fluidized solid powders into at least two parts in such manner as to permit at least one series of useful pressurized fluid discharges emitted in any ratios relative to the fluid quantities supplied and to each other.

2. Description of the Prior Art

Prior art fluid flow systems have proven to be unsatisfactory in many respects. In general, these systems have not permitted the emission of dual or multiple series of useful pressurized fluid discharges in any ratios relative to the fluid quantities supplied and to each other. Further, permitted discharge rates from such prior art systems do not range from minima to maxima and still provide systems having internal orifices and passages which are large in area so as to offer minimal resistance to fluid flows and to act as minimal sources of possible cumulative obstructions.

The prior art has also been unable to provide a fluid flow system which successfully delivers controlled, variably pressurized quantities of fluid to or into utilizing devices or processes, whether such deliveries are made to or into receivers which are operating at pressures which are equal to, greater than, or less than local atmospheric pressure. Additionally, the prior art has been unable to develop a fluid flow system which will solve the aforementioned problems and which will be a closed mechanism which remains sealed during periods of nonoperation, after having been operational, and will thus be continuously primed and ready for instant operation.

Prior art fluid flow systems have also not provided a positive and variable means of incremental distribution of fuel to the intake systems of internal combustion engines, whereby each cylinder of multicylinder engines receives precisely identical quantities of fuel required for any given speeds, local ambient temperatures and local atmospheric pressures. Further, the prior art fluid flow systems have not yielded constant in degree stratification of the combustible charge of internal, spark ignition, combustion engines or where such stratification may be varied in relation to the piston position to occur at any portion of the piston stroke and still be constant for any such chosen piston stroke portion for any chosen engine speed range.

SUMMARY OF THE INVENTION

The present invention provides a balanced fluid flow control apparatus for dividing sources of input fluid supply into at least two variable and controlled parts, at least one of which is discharged at predetermined rates of flow to utilizing mechanisms, devices or processes, the rates of flow being controlled to range from minima to maxima, the latter including, in the extreme, the total input fluid as the discharge, which overcomes all of the aforementioned disadvantages of prior art devices.

Briefly, the balanced fluid flow control apparatus of the present invention comprises a hollow assembled body, the various parts of which are rigidly secured to each other along fluidtight joints, communicating with a primary source of fluid supply. A dispersion chamber provided with a nonreturn valve is located within the assembled body and receives the fluid supplies from the primary source. At least one discharge orifice and at least one bypass orifice are associated with the dispersion chamber, and control means is associated with the discharge and bypass orifices to regulate the flow of fluid therethrough. A surge chamber is located downstream from each of the discharge and bypass orifices. An outlet system is associated with the surge chamber downstream from the discharge orifice and furnishes resistance to the flow of the

fluid. A bypass system is associated with the surge chamber downstream from the bypass orifice and balances the outlet system, the flows therethrough and the pressures created therein. Relief means associated with the dispersion chamber provides pressure relief in the apparatus when the input is greater than the sum of the desired maximum fluid flows through the discharge and bypass orifices and when the discharge and bypass orifices are rapidly closed and fluid is still supplied to the apparatus from the primary source.

In a preferred embodiment, the discharge and bypass orifices are positioned in an orifice plate which is fixed within the assembled body, and the control means associated with the discharge and bypass orifices to regulate the flow of fluid therethrough comprises a slidable control plate upstream of the orifice plate and slidably in contact therewith, and means to impart sliding movement to the plate, the plate being provided with a control port accurately aligned with respect to each of the discharge and bypass orifices so as to precisely control the open areas of the orifices as the control plate is moved.

In another preferred embodiment of the present invention, the dispersion chamber comprises a fixed in place cylinder and the discharge and bypass orifices are formed in the periphery thereof. In this embodiment the control means associated with the discharge and bypass orifices to regulate the flow of fluid therethrough comprises a rotating valve positioned within the cylinder so as to traverse the discharge and bypass orifices and means to impart rotation to the valve.

The present invention provides carefully determined formulas for determining the area of each of the discharge orifices and the area of each of the bypass orifices.

According to the present invention the outlet system of the control apparatus may comprise a collector disc, a rotatable distributor disc juxtaposed to the collector disc, gear train means operatively connecting the distributor disc to a power source, a cap atop the assembled body overlying the distributor disc the cap having a plurality of outlet ports correspond to the delivery locations required by the utilizing mechanism, such as the number of cylinders of an internal engine utilizing the control apparatus, means for aligning the ports in the distributor disc relative to the outlet ports in the cap, and injector means associated with each of the outlet ports on the cap.

The bypass system of the control apparatus may comprise a spring adjustable bypass valve associated with the bypass surge chamber, a bypass duct in communication with the bypass valve to return the bypassed fluid to the primary source of fluid supply, and means to adjust the fluid flow rates through the bypass valve. The means to adjust the fluid rates through the bypass valve includes a spiral metallic strip which expands and contracts in length with sufficient force and movement to adjust the tension in the spring of the bypass valve and to thereby set the bypass valve for the desired flow rate therethrough at various temperatures. The means to adjust the fluid flow rate through the bypass valve may also include an automatic adjustment for the flow rate through the valve with changes in altitude.

The relief means associated with the dispersion chamber to provide pressure relief in the apparatus preferably comprises a spring adjustable relief valve communicating with the primary source of fluid supply and means for delaying the closing of the relief valve so as to eliminate even momentary high internal pressures within the apparatus.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a vertical longitudinal section through the device taken on line 1-1 of FIG. 3.

FIG. 2 is a partial vertical section taken on the line 2-2 of FIG. 4.

FIG. 3 is a vertical section taken on the line 3-3 of FIG. 1.

FIG. 4 is a sectional plan view taken on the line 4-4 of FIG. 3.

FIG. 5 is a horizontal cross-sectional view taken on the line 5-5 of FIG. 3.

FIG. 6 is a horizontal plan view taken on the line 6-6 of FIG. 1.

FIG. 7 is a vertical cross section taken on the line 7-7 of FIG. 2.

FIG. 8 represents a discharge orifice in outline.

FIG. 9 represents a bypass orifice in outline.

FIG. 10 is a plan view of a control plate of this invention, reflected, taken on the line 10-10 of FIG. 1.

FIG. 11 is a longitudinal sectional view through an injector.

FIG. 12 is a cross-sectional view through an injector taken on the line 12-12 of FIG. 11.

FIG. 13 is a graphical representation showing assumed gas intake and vacuum curves where such intakes and vacuums vary in accordance with assumed operational speeds of one type of utilizing mechanism and where the discharges from the apparatus of the present invention are regulated in accordance with specific requirements of the utilizing mechanism at the indicated operational speeds.

FIG. 14 is a graphical representation showing air to fuel ratios.

FIG. 15 is an area and velocity diagram pertaining to radial ports of a distributor.

FIG. 16 is a vertical cross section through a second embodiment of the present invention taken on the line 16-16 of FIG. 17.

FIG. 17 is a horizontal cross section through a second embodiment of this invention taken on line 17-17 of FIG. 16.

FIG. 18 is a vertical cross section through a second embodiment of this invention taken on the line 18-18 of FIG. 17.

FIG. 19 is a cross-sectional view similar to FIG. 16 showing a modification thereof.

FIGS. 20 to 30 are schematic representations of both embodiments of the present invention.

FIG. 20 is a combination detail derived from FIG. 1.

FIG. 21 is a vertical cross section taken on the line 21-21 of FIG. 20.

FIGS. 22 and 23 are plan views of orifice plates provided with differing orifices, taken on the line 22-22 of FIGS. 20 and 21.

FIG. 24 is a plan view of an orifice plate which is provided with two pairs of orifices, taken on the line 24-24 of FIG. 25.

FIG. 25 is a vertical cross section taken on the line 25-25 of FIG. 24.

FIG. 26 is a cross-sectional view of the second embodiment of this invention taken on the line 26-26 of FIG. 27.

FIG. 27 is a cross-sectional view of the second embodiment of this invention taken on the line 27-27 of FIG. 26.

FIG. 28 is a cross-sectional view of the second embodiment of this invention similar to FIG. 26 and showing a modification thereof where this embodiment is arranged for constant fluid flow input.

FIGS. 29 and 30 are planar, developed, representations of discharge and bypass orifices which are utilized in the second embodiment of this invention.

FIG. 31 is a graph of typical fluid flows, fluid flow velocities, fluid pressures and discharge travel for a balanced fluid flow control apparatus of this invention when associated with a variable primary source of fluid supply.

FIG. 32 is a compilation of the numerical data from which the graph of FIG. 31 is plotted.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

As was previously explained, the present invention provides a balanced fluid flow control apparatus for dividing sources of input fluid supply into at least two variable and controlled parts at least one of which is discharged at predetermined rates of flow to utilizing mechanisms, devices, or processes, the rates of flow being controlled to range from minima to maxima, the latter including, in the extreme, the total input fluid as the discharge. It is important to emphasize that the word fluid is meant to include any gas, liquid or fluidized solid powder, capable of flowing under force of pressure, i.e., capable of deformation when subjected to shear stress. It should

also be emphasized that by use of the word balanced is meant the action of divided flows where one or a plurality of delivery flows are counterbalanced by at least one separately controlled flow, sometimes referred to as bypass flow, not necessarily equal in rate to the deliveries, yet contributing importantly to the control of the rates of flow of the deliveries.

It should also be noted that in fluid flow control systems such as hereinafter described, the separate quantities of fluid flow are contingent in rate upon the areas of the orifices through which the fluid is flowing and the velocities of the flows, the specific gravity, viscosity, if any, and the temperature of the fluid. Additionally, the internal or operating pressures of systems such as the instant invention depend upon the resistance to flow established or required by outlet valves, injectors or nozzles, coefficients of discharge of internal orifices, ports, ducts, or tubing connections, bypass valves and other resistance contributing elements.

Referring first to FIGS. 1 and 2, it will be seen that the balanced fluid flow control apparatus of this invention is contained within an assembled body 10, the various parts of which are rigidly secured to each other along fluidtight joints 12, by bolts or screws [not shown]. The assembled body is associated with a primary source of fluid supply. The fluid flow to be divided enters the control apparatus through the inlet duct 14 and flows into the dispersion chamber 16. The incoming fluid flow may be directly from the primary source or through a pump built into the apparatus, such as the pump 18, the only requirement being that the fluid supplied be under a pressure corresponding to the internal pressure or pressures developed within the control apparatus.

The fluid flow entering the dispersion chamber 16 lifts the nonreturn valve 20 and flows into each compartment of the chamber. The dispersion chamber 16 is preferably provided with a slidable partition 22 which divides the chamber into two compartments, one compartment 16a for receiving that portion of the fluid supply which will be directed through a discharge orifice and one compartment 16b for receiving that portion of the fluid supply which will be directed through the bypass orifice. The slidable partition 22 is also preferably provided with downwardly extending prongs 24 thereon, which limit the lateral movement of the nonreturn valve 20, and with a baffle 26, which directs the entering fluid supply into each of the compartments 16a, 16b.

The vertical depth of the conically bottomed dispersion chamber 16 is optional and may be shallow as in FIGS. 1 and 2, or deeper as in FIG. 20. Such dimensions are governed by the ultimate fluid velocities which when high require directional vanes 28, in the dispersion chamber 16 in order to unify and maximize the coefficients of discharge through the discharge and bypass orifices. The dispersion chamber 16 may, of course, be rectangular, circular or oval in cross section, as desired.

At least one discharge orifice 32 and at least one bypass orifice 34 are provided in association with the dispersion chamber 16. Control means 38 are associated with the discharge and bypass orifices 32 and 34 to regulate the flow of fluid therethrough.

The discharge and bypass orifices 32 and 34 may be positioned in an orifice plate 36 which is fixed within the assembled body 10. In this instance the control means 38 comprises a slidable control plate 38a upstream of the orifice plate 36 and slidably in contact therewith, and means 40 to impart a sliding movement to the control plate. The control plate 38a and the orifice plate 36 are in intimate contact along the interface thereof so as to ensure that no leakage occurs therebetween. The control plate 38a is provided with a control port 38b accurately aligned with respect to each of the discharge and bypass orifices 32 and 34 so as to precisely control the open areas thereof as the control plate 38a is moved.

The discharge and bypass orifices 32 and 34 in the orifice plate 36 are a vital part of the present invention and must be accurately formed, aligned and located, as will be more fully explained hereinafter. It will be obvious that the requirement

of precision in alignment of the orifices 32 and 34 and the control ports 38b is vital to the proper functioning of the control apparatus of the present invention, since the necessary delicate control of the open areas exposed to fluid flow and the consequent delicate control of the fluid flows, whether small or large, are dependent upon the precise placement of the leading edges of the ports 38b and the preciseness of the forming of the orifices 32 and 34.

The orifice plate 36 and the control plate 38a must have intimate contact along their interface, preferably by means of lapping, to ensure that no leakage occurs therebetween and to still permit the free movement of the control plate 38a. While the orifice plate 36 and the control plate 38a are held in contact by fluid pressure, it has been found that strip springs 42 located within the channels 44 also aid in holding the plates 36 and 38a contiguous with each other.

A slidable control rod 40, one end 40a of which is received within the assembled body 10 and fixed to the control plate 38a, and the other end 40b of which is provided with a transverse bar portion 40c is exemplary of how sliding movement may be imparted to the control plate 38a. Leakage of fluid around the slidable rod 40 is prevented by the conventional packing 46 shown in two parts with the precautionary leak-off connection 48. Since internal pressures tend to force the rod 40 outwardly to the right with respect to FIGS. 3 and 4, a plurality of return springs 50 are provided to counteract this movement. The return springs 50 are each secured at one end 50a to the assembled body 10 and at the other end 50b to the transverse bar portion 40c of the control rod 40. It should also be noted that the return springs 50 are necessary in case of breakage or disconnection in any operating means.

An application of the present invention to an internal combustion engine which includes the distributor-collector elements disclosed in U.S. Pat. No. 3,327,632, in the name of Carl F. Gromme, is illustrated by FIGS. 3 and 4 where the link 52 and the bellcrank 54 activate the control rod 40 through the movement of the accelerator link 56. The stops 58 and 60 limit the movement of the link 52 and the bellcrank 54, the sliding control rod 40 and the control plate 38a, thereby limiting the fuel output to suit the desired speeds from idle to full throttle. The bellcrank 54 at the same time moves the throttling butterfly valve 62, or its equivalent, from a nearly closed position to a wide open position within the intake stack 64.

Separate surge chambers 66 and 68 are located on the downstream side of each discharge and bypass orifice 32 and 34. The surge chamber 66 receives the fluid supplied through the discharge orifice 32 and forms a part of the discharge system, while the surge chamber 68 receives the fluid supplied through the bypass orifice 34 and forms a part of the bypass system. The surge chambers 66 and 68 are important in that it is in these chambers that the vena contracta is formed, to a greater or lesser degree, depending upon the characteristics of the flow of the fluid through the control apparatus and the local velocity therethrough.

The fundamental requirement of establishing known resistances to fluid flow throughout the system of the control apparatus of this invention, particularly with regard to outlet flow, is met by providing spring-loaded poppet-type valves for the injectors and for the bypass system, as opposed to small drilled holes in nozzles. The ranges of working pressures for the poppet-type valves are less than for the nozzles for comparable ranges of rates of flow.

As will be more fully explained hereinafter, discharge and bypass orifice areas and dimensions, reactions of spring-loaded valves, velocities of flow through the valves, and internal pressures within the control apparatus, are substantially determined when input rates of fluid flow and the lengths of the discharge and bypass orifices 32 and 34 are known or specified and when valve sizes and spring rates are assigned.

The outlet system associated with the surge chamber 66 downstream from the discharge orifice 32 furnishes resistance to the flow of the fluid through the control apparatus. Exemplary of such an outlet system is an application of the control

apparatus of this invention to an internal combustion engine which includes the distributor-collector elements of U.S. Pat. No. 3,327,632, in the name of Carl F. Gromme. Briefly, this distributor-collector outlet system comprises a collector disc 70 overlying and in communication with the surge chamber 66 downstream from the discharge orifice 32. The collector disc is provided with a central recessed area 70a in its upper face and is held against rotation by the springs 72. A rotatable distributor disc 74 is juxtaposed to the collector disc 70. The distributor disc 74 is also provided with a recessed area 74a in its under face which mates with the central recessed area 70a in the upper face of the collector disc 70. A plurality of ducts 74b project radially from the recessed area 74a in the distributor disc 74, each of the ducts 74b terminating in a port 74c in the upper face of the distributor disc 74.

The springs 72 which hold the collector disc 70 against rotation also provide the necessary light thrust of the collector disc 70 against the distributor disc 74. A flexible seal 76 prevents fluid from leaking into any space between the assembled body 10 and the collector disc 70 without restricting movement of the disc.

A gear train means operatively connects the distributor disc 74 to a power source to impart rotational movement to the distributor disc. As shown, the gear train means may include a shaft 78 which communicates through the gear train 80. The shaft 78 may be conventionally geared to turn at any desired speed. For example, if the shaft 78 is made to turn at one-half crank speed, the gear train 80 will turn the distributor disc 74 at one-fifth of that speed. It should also be noted that an ignition spark distributor may also be mounted on the end 78a of the shaft 78.

A cap 82 is positioned atop the assembled body 10 overlying the distributor disc 74. The cap 82 is provided with a plurality of outlet ports 84 which meet the requirements of the utilizing mechanisms, devices or processes. For example, the ports could correspond to the number of cylinders of an internal combustion engine utilizing the control apparatus. Accordingly, if the control apparatus is to be utilized in connection with a 6 cylinder, four stroke engine, the distributor disc 74 is provided with five traversing ports 74c, and the cap 82 is provided with six outlet ports 84. The distributor disc 74 is caused to rotate at one-tenth of the crank speed. The duct 74b in the distributor disc 74 and the ports 74c and 84 in the distributor disc and cap, respectively, are made relatively large to limit the fluid flow velocity therethrough. The total angle subtended by the ports 74c and 84 is determined by the chosen duration of the injections. FIG. 5 indicates that the subtended angle would be substantially 12° when the speed of rotation of the distributor disc 74 is one-tenth of the crank speed. The injection period would therefore be 120° of the crankshaft rotation.

Means are also provided for aligning the ports 74c in the distributor disc 74 relative to the outlet ports 84 in the cap 82.

As can best be seen from FIG. 11, spring-loaded outlet means or injector means 86 may be associated with each of the outlet ports 84 in the cap 82. The injector means 86 are necessarily very sensitive and the discharges therethrough may take the form of a spray into any desired receiver. Preferably, the injector means 86 comprises a spring-loaded poppet valve. The poppet valve is provided with a body portion 88. The extended ends 88a of the body portion 88 are, of course, optional as to length. However, when the ends 88a are to be extended to deliver fluid into either a chilled or heated receiver, such as, for example, for delivery to an internal combustion engine, it is desirable to make the extended portions of the injector body 88 of a material which provides a minimum coefficient of expansion. A preferable material has been found to be invar steel.

When multiple injectors are utilized, it is very important that each injector be matched to ensure equal discharges. Accordingly, manufacturing tolerances of the injectors must require that the injectors be made delicately adjustable. To accomplish this end, it has been found that a preferable injector

body 88 includes a slant-sided groove 90a in the slidable collar 90. The slidable collar 90 bears on the enlarged, perforated end 92 of the stem 94. Slant tip set screws 96 are placed so as to exert or release pressure on opposite sides of the groove 90a in the collar 90. This permits delicate adjustment of the tensions of the springs 98.

It will, of course, be obvious that the aforementioned description of the outlet system is only exemplary of an outlet system which may be utilized in conjunction with the surge chamber 66 downstream from the discharge orifice 32, and that any desired outlet system may be utilized, depending upon the utilizing mechanisms, devices or processes to which discharge will be made, so long as the outlet system furnishes resistance to the flow of the fluid within the control apparatus.

The bypass system associated with the surge chamber 68 downstream from the bypass orifice 34, which balances the outlet system, the flows therethrough and the pressures created therein, may best be seen in FIGS. 2 and 7. Briefly, the bypass system comprises a spring adjustable bypass valve 100 associated with the bypass surge chamber 68, a bypass duct 102 in communication with the bypass valve 100, to return the bypassed fluid to the primary source of fluid supply, and means to adjust the fluid flow rate through the bypass valve.

It will be obvious that when it is desirable to increase or decrease the flow velocities through the bypass valve 100 and the injector means 86, pressure areas and the resistive strength of springs may be varied to suit the desired needs. Correctly proportioned quantities of fluid will pass through the bypass valve 100 and the injector means 86 in concert with the proportioned fluid quantities flowing through the discharge and bypass orifices 32 and 34.

The fluid flow rates through the bypass valve 100 may be adjusted manually, by utilizing expansion and contraction characteristics of metals, and/or by response to changes in operating altitude relative to a given altitude as datum. The initial setting of the tension of the spring 101 may be accomplished by means of the shaft 104, which includes a threaded section 104a which engages the matching threads 106 in the extension 108 of the assembled body 10. The shaft 104 bears against the slidable sealed plunger-type end 100a of the valve stem 100b of the bypass valve 100, compressing or releasing the spring 101 and balancing the pressure set by the injector spring or springs 98. It will be understood that all such pressures will be measured by suitably placed gauges, such as a gauge through the plugged hole 110. The plugged hole 110 may also be used to enable visual verification of flow through the bypass valve 100.

The means to adjust the fluid flow rates through the bypass valve 100 may include a spiral metallic strip 112 associated with the spring adjustable bypass valve 100. The spiral metallic strip 112 is accommodated to a desired temperature datum and remains so until the tension in the spring 101 of the bypass valve 100 is adjusted so that the bypass valve 100 is properly set for the desired flow rate therethrough. Accordingly, subsequent temperature changes deviating from the temperature datum cause the strip 112 to expand and contract in length with sufficient force and movement to adjust the tension in the spring 101 of the bypass valve 100 and to thereby set the bypass valve 100 for the desired flow rate therethrough at such temperatures.

It is important to note that appreciable variations in ambient temperatures with respect to a relative datum affect the relative viscosities of many viscous fluids and also the relative weights of gases. Accordingly, each temperature change requires a modification, however slight, in the tension in the springs of the bypass valve 100 and the injector means 86. Although spring tension adjustments may be easily accomplished with respect to one injector means 86, a plurality of injector means creates matching difficulties which do not easily permit automatic variation. The spiral metallic strip 112 automatically adjusts the tension in the spring 101 of the bypass valve 100. When the lock nut 114 is loosened, the holder 116 is freed so that it will assume a position where the strip 112 is

accommodated to the temperature datum and remains so until the bypass valve 100 is properly set. Without disturbing this setting of the bypass valve 100, the lock nut 114 may then be tightened so that the collar 116a and the perforated cover 118 are securely held against the shoulder 104a on the shaft 104. Temperature changes deviating from the temperature datum cause the strip 112 to expand or contract in length with sufficient force and movement to rotate the shaft 104 by reason of the thread 106 and to adjust the tension in the spring 101 of the bypass valve 100. The amounts of the movements are governed as desired by the stretching and shrinking of the strip 112 and by the pitch of the threads 106.

The means to adjust the fluid rates through the bypass valve 100 may also include an automatic adjustment 120 for controlling the flow rate through said bypass valve with changes in altitude. The automatic adjustment 120 comprises a flexible diaphragm 122 associated with the spring adjustable bypass valve 100. The flexible diaphragm 122 is exposed to atmospheric pressure on one side thereof through the perforated cover 123 and is sealed on the other side thereof in the chamber 125 at a desired pressure and temperature datum. Accordingly, the tension in the spring 101 in the bypass valve 100 is adjusted by changes in the position of the diaphragm 122 as the sealed in air expands and contracts. In operation, the expansion and contraction of the diaphragm 122 as the sealed in air expands and contracts permits secondary rotation of the shaft 104 through the linkage 124, which causes rotation of the worm 126 engaged in the worm gear 128 formed integrally on or applied to the rim of the holder 116.

As can be seen, the rotation of the shaft 104 due to changes in length of the coil 112 with respect to temperature deviations from a temperature datum is separate from the rotation thereof with respect to changes in altitude. This is so because the automatic altitude adjustment 120 is secured in place by the releasable clamping bracket 130, which must remain loosened with the worm 126 and the worm gear 128 nearly in full mesh until the lock nut 114 is tightened after the holder 116 has been established in position. The holder 116 is locked in position relative to the altitude assembly 120 and rotates only through the movement of the flexible diaphragm 122. If the automatic adjustment for altitude assembly 120 is omitted from the control apparatus, the holder 116 must be secured against rotation. This may be accomplished by means of a stationary bracket 132, which is formed with an extended lug 132a within which the locking screw 134 may be placed provided that the collar 116a is always free to turn with the shaft 104.

Relief means 136 is associated with the dispersion chamber 16 so as to provide pressure relief in the control apparatus when the input is greater than the sum of the desired maximum fluid flows through the discharge and bypass orifices 32 and 34 and when the discharge and bypass orifices 32 and 34 are rapidly closed and fluid is still supplied to the apparatus from the primary source. The relief means 136 preferably comprises a spring adjustable relief valve 138 communicating with the primary source of fluid supply and means for delaying the closing of the relief valve so as to eliminate even momentary high internal pressures within the control apparatus. The relief valve 138 is provided with a perforated guide 140, which acts against the spring 142, and the stem 144 of the relief valve extends through a bore 146a in the stationary collar 146, terminating in a plunger 148 within a bore or chamber 150. The means for delaying the closing of the relief valve 138 comprises a duct 152 leading from the dispersion chamber 16 to the bore or chamber 150. The duct 152 is provided with a nonreturn valve 154 which is operable at a lower pressure than the relief valve 138. Accordingly, as internal pressures buildup within the control apparatus, fluid first opens the nonreturn valve 154, flowing into the bore or chamber 150 at substantially the same time at which the internal pressures open the relief valve 138, causing pressure to be exerted against the plunger 148 and thus multiplying the movement of the relief valve 138. The speed of the closing of the relief valve 138 may

be regulated because the fluid trapped within the bore or chamber 150 may only be gradually released therefrom through the clearances between the valve stem 144 and the bore 146a in the stationary collar 146.

An enlarged outline of the discharge and bypass orifices 32 and 34 is disclosed in FIGS. 8 and 9 when the outline of these orifices is determined from the curves A and B of FIGS. 13 and 14. Curve A of FIG. 13 is an assumed illustrative curve of the air inducted by one cylinder of a four stroke internal combustion engine under normal atmospheric pressure and temperature, operating at speeds of from 500 to 6,000 revolutions per minute. Curve B of FIG. 14 is the curve of the air-to-fuel ratios assigned to the aforementioned four stroke internal combustion engine. Curve C of FIG. 14 is a typical representative curve of air-to-fuel ratios differing from the air-to-fuel ratios indicated by curve B, and is equally applicable in calculating the areas of the discharge and bypass orifices 32 and 34. It will be obvious that an unlimited number of similar curves may be produced and usefully adopted.

Curve D of FIG. 13 is a representative curve of the partial vacuums which inevitably exist to varying degree in the cylinders and intake systems of internal combustion engines, reciprocating or other types. These partial vacuums reach their maximum at low speeds, decreasing as the throttle is opened to the speed range of maximum inflow, then increasing as friction, inertia and dynamic effects redelimit the air intake. The effects of delivering fluid into a receiver under any appreciable vacuum are to increase the fluid flows from the injectors or nozzles as compared with the fluid flows designed or required to be delivered by identical means and under identical internal pressures into receivers with the same atmospheric pressures as in the receivers of bypassed fluid, normally the atmospheric pressure of the location of use. This imbalance, if undesirable, is compensated for by decreasing the calculated area of the discharge orifice 32 while retaining the calculated area of the bypass orifice 34, as shown by the curved, dashed line 156 of FIG. 8. The converse of this statement, however, is true when the injectors or nozzles are delivering fluid into a receiver which is under greater pressure than atmospheric pressure. In this situation the bypass orifice 34 remains as originally calculated and the discharge orifice 32 is enlarged from the calculated area so as to counter the imbalance created by the increases in receiver pressure, as shown by the curved, dashed line 158 in FIG. 8. This enlargement may be uniform along the length of the discharge orifice, or it may be of a form shown by the line 158, which is applicable to receivers (in this case a supercharged internal combustion engine), with varying increasing pressures.

FIG. 15 shows the mean velocities of fluid flow and attendant pressure drops through and across the ports 74c and 84 located in the distributor disc 74 and the cap 82 of the outlet system associated with the surge chamber 66 downstream from the discharge orifice 32, as the ports 74c traverse the ports 84 when the distributor disc 74 is rotated. During the operation of the supply pump 18, the ports 74c and 84 are filled with fluid under varying pressures. Accordingly, unless the injection periods overlap, the fluid cannot flow until a port 74c begins to traverse a port 84. It will be obvious, therefore, that fluid flow through the traversing ports 74c and 84 begins from zero, reaches a maximum, and then decreases to zero. Line 160 of FIG. 15 represents one-half of the travel of one 6° port 74c past another 6° port 84 and has six divisions, each division representing 1° of travel. A complete symmetrical graph may be formed by joining-on a mirror image of FIG. 15. Line 172 of FIG. 15 is a curve representing the areas of the ports 74c and 84 which are open to fluid flow during a period of one-half traverse of the distributor disc 74. Ordinates erected on the line 170 at 1° intervals, laid off at a convenient scale, represent the known fluid velocities at the 1° intervals. The area under the curve produced by joining the ends of the ordinates divided by the line 170 multiplied by the scale of ordinates, in the consistent units, is the mean velocity of the fluid flow during one traverse of the ports 74c in the distributor disc 74.

It should be emphasized that FIG. 15 applies to a known constant rate of flow from the discharge orifice 32, and also to known, varying rates of flow through the discharge orifice 32, in which cases the curve 174 will assume differing paths. FIG. 15 illustrates the result of one quantity of fluid flowing through the discharge orifice 32 at a constant rate, for example, at the maximum rate of speed of an internal combustion engine.

When the mean velocities of flow through the ports 74c and 84 are known, the mean coefficients of discharge through the ports 74c and 84 may be estimated for preliminary purposes until they are precisely determined by calibration. Such estimates are necessary in order to gauge the corresponding mean pressure drops across the ports 74c and 84 during periods of traverse as the distributor disc 74 is rotated. These pressure drops, however slight, add to the fluid pressures in the dispersion chamber 16. It should again be pointed out that the internal pressures in the control apparatus originate at the injector means 86, such as the injectors 88 or their equivalent, and determine the resistances to fluid flow required by the bypass valve assembly 100. With low fluid velocities through the discharge and bypass orifices 32 and 34 and relatively large orifices and ports, these added pressures are minimized.

When the rate of flow from the primary source of supply is constant, such as, for example, from a pump 18 driven at constant speed, the discharge orifice 32 and the bypass orifice 34 require that the control ports 38b in the slidable control plate 38a be placed so as to expose one orifice to flow while simultaneously reducing the area of the other orifice to flow. An exemplary disclosure of the orifices 32 and 34 is shown in FIG. 10 where it can be seen that the discharge orifice 32 is open from zero to full open area as the bypass orifice 34 goes from full open area to zero. The rates of flow of the fluid through the discharge orifice 32 may follow any prescribed law or requirement which, as hereinbefore explained, determines the configurations of the orifices. It should be noted that when the discharge orifice 32 and the bypass orifice 34 are formed as identical rectangles, the sum of the areas open to flow is always equal to the area of one orifice. Accordingly, the curve of internal pressure within the control apparatus is linear since the quantity of fluid flowing therethrough increases or decreases in direct proportion to the orifice areas open to flow, assuming, of course, constant flow velocities through the ports 74c and 84 during the traversing process as the distributor disc 74 is rotated.

In a second embodiment of the control apparatus of this invention, as seen in FIGS. 16 through 19, the dispersion chamber 176 comprises a fixed in place cylinder and the discharge and bypass orifices 32 and 34 are formed in the periphery thereof. The control means associated with the discharge and bypass orifices 32 and 34 to regulate the flow of fluid therethrough comprises a rotating valve 178 positioned within the cylindrical chamber 176. Means are provided to impart rotation to the valve 178 so that it traverses the discharge and bypass orifices 32 and 34. The means to impart rotation to the rotating valve may generally correspond to the exemplary means which has heretofore been described in connection with imparting sliding movement to the slidable control plate 38a of the control means 38.

As can be seen from FIG. 16, the fluid supply from the primary source enters the cylindrical dispersion chamber 176 formed in the sleeve 182 within the assembled body 10 through the inlet duct 180, which is provided with a nonreturn valve 184. The sleeve 182 is provided with a snug fit in the bore 186 of the assembled body 10. The discharge orifice 32 and the bypass orifice 34 are formed in the periphery of the sleeve 182. The rotating valve 178 controls the areas of the discharge and bypass orifices 32 and 34 which are open or closed to fluid flow as it is rotated through an arc equal in length to the arcuate length of the orifices. When the rotary valve 178 is in the position shown in FIG. 16, the discharge and bypass orifices 32 and 34 are closed to flow. The rotary valve 178 is made so that it conforms precisely to the inner cylindrical surface of the sleeve 182 of the cylindrical disper-

sion chamber 176, and it is held in positive contact thereagainst by means of fluid pressure within the chamber and by means of the springs 188. The rotary valve 178 is formed so as to provide the free flow of the variable fluid input from the primary source to the discharge and bypass orifices 32 and 34 by means of the apertures 190, the cutout portion 192 and the grooves 194. Rotation of the valve 178 may be either clockwise or counterclockwise. The outlet system, bypass system and release means are partially shown on FIGS. 16 through 18. However, it should be emphasized that these systems are identical in principle with the system hereinbefore described.

The rotary valve 178 is mounted within the cylindrical dispersion chamber 176 on the squared end of the shaft 196. The shaft 196 is sealed against leakage along the bore 198a in the fixed member 198 by means of the conventional packing 200. Rotation may be imparted to the rotary valve 178 and the shaft 196 by means of the lever 202 which may be actuated by the linkage means hereinbefore described in connection with FIGS. 3 and 4.

When the rate of fluid flow into the cylindrical dispersion chamber 176 is constant, the discharge and bypass orifices 32 and 34 are traversed in a manner analogous to that previously discussed in connection with the first embodiment. The rotary valve 178 is therefore formed with a single-arcuate contact face having a length sufficient to close one of the orifices 32 and 34 to flow while leaving the other of the orifices fully open and exposed to flow, regulating the orifices open to flow as it is rotated in the proper direction (clockwise when arranged as shown in FIG. 19).

FIGS. 20 through 30 are schematic representations of the two embodiments of the control apparatus of this invention and will more fully aid to explain the operation thereof.

Turning first to FIG. 20, it will be seen that fluid from the primary source of supply enters the dispersion chamber 16 through the inlet duct 14. The dispersion chamber 16 may be a single chamber or it may be divided into multiple compartments 16a and 16b. The fixed orifice plate 36 is provided with a discharge orifice 32 and a bypass orifice 34. The slidable control plate 38a is provided with the control ports 38b. The bypass valve 100 and the relief valve 138 are also indicated. The collector-distributor system in connection with the outlet system is indicated as a separate collective system 204 having one tubing connected injector 86.

An outlet system not hereinbefore described is shown by a plurality of tubing connections 206 leading directly from the discharge surge chamber 66 and the bypass surge chamber 68. Such an arrangement makes it possible to eliminate the bypass valve assembly 100 and the collector-distributor 204. In this way the entire input either flows at uniform rates through the injectors 86 of the tubing connections 206 or at differing rates, depending upon whether the injectors 86 are matched and whether the discharge and bypass orifices 32 and 34 are identical or have differing forms. It will, of course, be obvious that the bypass assembly 100 may be retained as an overall flow regulating or modifying device.

When multiple outlets are required, as for example, in an internal combustion engine of sufficient number of cylinders, and two distributors are used, twin dispersion chambers 16 may be formed in the assembled body as indicated in FIG. 25. The bypass and discharge orifices 32 and 34 may, of course, be identical, and only one bypass assembly 100 is required.

FIG. 22 shows the orifice plate 36 overlying the control plate 38a at a substantially reduced scale. It will be seen that the discharge and bypass orifices 32 and 34 are of unequal width and will thus yield unequal graduated flows when the input fluid from the primary source is variable. However, when the input fluid from the primary source is at a constant rate, high pressures, with consequent large pressure drops, will develop within the compartments 16a and 16b of the dispersion chamber 16. Any high pressure drops across the discharge orifice 32 result in reduced pressures at the outlet ports 84, with consequent pronounced reduction in outlet velocities.

FIG. 23 is substantially identical to FIG. 22 except that it is arranged for input fluid from the primary source of supply having constant rates of flow.

FIG. 24, also at a reduced scale, indicates the condition when duplicate discharge orifices 32 and duplicate bypass orifices 34, along with twin dispersion chambers 16, are supplied from a common fluid supply.

FIGS. 26 and 27 illustrate in schematic form a second embodiment of the present invention wherein the rotating valve 178 controls the division of a variable input fluid which enters the cylindrical dispersion chamber 176 through the inlet duct 140. FIG. 29 is a planar development of the discharge and bypass orifices 32 and 34 (shown in rectangular form) which are located in the periphery of the sleeve 182 of the cylindrical dispersion chamber 176. It will be seen that when rotation is imparted to the rotary valve 178, that the discharge orifice 32 opens to flow as the bypass orifice 34 opens to flow.

FIG. 28 shows the required rotating valve 178 when the input flow from the primary source is at a constant rate. FIG. 30 is also a planar development of the discharge and bypass orifices 32 and 34. It will be seen that the discharge orifice 32 is shown as fully open, thus passing the entire input flow, and that the bypass orifice 34 is fully closed to flow. Accordingly, when the bypass orifice 34 is fully opened, the entire input fluid from the primary source will be bypassed.

It is important to again note that when the control apparatus of this invention is provided with multiple dispersion chamber 16, there may be an equal number of inlet ducts, each with pressurized fluid supply identical or not as the cases may be. Accordingly, differing fluids under variously chosen velocities may be used.

It should also be noted that fluids cannot flow through the control apparatus of this invention until all air is expelled therefrom and the system or systems are filled. Once filled, the nonreturn valve and the spring-loaded discharge, bypass and relief valves retain the fluid during the periods of no supply or nonoperation.

It should also again be emphasized that the discharge and bypass orifices 32 and 34 may be provided as one pair, multiple pairs or pairs combined with a single orifice. Usually a pair of orifices will consist of one discharge orifice and one bypass orifice. Multiple orifices of more than two may be combined in other ways as long as there is provided one bypass orifice for each fluid inlet into the dispersion chamber. This, of course, means that a plurality of dispersion chambers, each with a separate fluid supply, may be provided within one assembled body 10.

In all cases the fluid flow passing through a discharge orifice 32 is discharged downstream either through a distributor-collector outlet system and then through a series of injector means, or directly through injector means or similar spring-loaded outlets. At the same time, bypassed fluid flows downstream through a spring-loaded bypass valve to the source of supply.

As was hereinbefore explained, the discharge orifice 32 and the bypass orifices 34 are sized so that each pass the varying quantities of fluid, at selected velocities, which are required to be discharged in accordance with the particular demands and to bypass all excess fluid. Accordingly, the discharge and bypass orifices 32 and 34 form the means of dividing the fluid input or inputs. By choosing the proper configuration for the orifices 32 and 34, so that their areas are simultaneously exposed to flow, any conceivable relative divisions of inflowing fluids may be accomplished, all within the limits of the chosen dimension and capacity of this control apparatus.

When the rates of flow of the input fluid are greater than the balanced maximum flows required for the discharge and bypass orifices for given maximum pressures in the dispersion chambers, and under conditions of rapidly reducing the flow areas of the orifices, relief valves act to reduce the internal pressures to the required level or levels. This is particularly a requirement of internal combustion engines during deceleration.

Internal pressures within the control apparatus are governed by resistances to flow established by the springs closing the discharge injectors, or means equivalent thereto, and by the bypass valves. The bypass valves may be made identical to or similar to the discharge injectors. The valve springs are selected to deflect, under selected pressures, precise amounts so as to yield open valve areas through which fluids will flow with particular velocities. Conversely, the deflections, spring wire gauge, and spring mean diameter may be first selected and the load which produces such deflection may then be determined. The resulting internal pressures are then the quotients of the load divided by the valve head areas. The resulting velocities of flow then are the quotients of the rates of flow of discharged fluid divided by the open areas of the lifted valves.

Internal pressures determined in the aforementioned manner substantially establish the pressures in the discharge system, including all tubing, distributor ducts and passages connected thereto, and in the dispersion chambers. Such pressures are increased in varying degrees, usually small, due to the coefficients of discharge of the ports and orifices through which the fluid must flow. The total pressures in the dispersion chambers then give the heads against which the supplies of fluid from the primary source must be pumped.

The discharge and bypass orifices may be of any size. However, by designing both the discharge and bypass orifices to pass fluids at low velocities identical at any given speed of operation, such as for example, at any given revolution per minute, preferably constant throughout flow periods, the discharge and bypass orifices may be made very large relative to the discharge or bypass valve areas. Such procedure will minimize the energy which is lost in the system, i.e., will cause negligible changes in the internal pressures of the control apparatus.

The following are the basic relationships and equations of fluid mechanics relating to and concerning fluids in motion and other physical formulas applicable to fluid flows, orifice areas and spring actions within the control apparatus of this invention. Included therewith is an illustrative example of related calculations and procedures.

Symbols and abbreviations used, with subscripts *d* and *b* denoting the discharge and bypass orifices 32 and 34, respectively, are:

Q =Maximum rate of fluid flow from a variable primary source (in.³/sec.).

L =Fraction of Q flowing into the chamber 16 at any given speed of operation, such as, for example, at any given revolution per minute of an internal combustion engine (in.³/sec.).

Q_d =Fraction of L diverted through the discharge orifice 32 (in.³/sec.).

Q_b =Fraction of L diverted through the bypass orifice 34 (in.³/sec.).

A_d =Area of the discharge orifice 32 when open to flow (in.²).

A_b =Area of the bypass orifice 34 when open to flow (in.²).

A'_d =Free flow area through the discharge valve when lifted (in.²).

A'_b =Free flow area through the bypass valve when lifted (in.²).

V =Flow velocity through the orifices 32 and 34 (in./sec.).

V_d & V_b Flow velocities through the discharge and bypass valves 32 and 34 respectively (in./sec.).

t =Time duration in seconds of discharge flows when intermittent, for example, fuel injections in an internal combustion engine.

n =Crank revolutions per minute of a four stroke internal combustion engine.

O =The angle the crank of an internal combustion engine turns in time t (degrees).

W_a =Weight of fluid or gas of known rate or rates of flow into the utilizing mechanism into which the controlled steady or intermittent flows of discharged fluid are

discharged in accordance with the requirements thereof, such as, for example, weight of air inducted during one intake stroke of one cylinder of an internal combustion engine (lbs.).

A/F =Ratios of the weight of flow discharged to the weight of fluid supplied to the apparatus.

S =Specific gravity of fluid.

w =Specific weight of fluid (lbs./in.³).

P =Nominal working pressure in the chamber 16 (p.s.i.).

DP_d =Pressure drops across the orifices 32 and 34 (p.s.i.).

DP_v =Pressure drops across the discharge valve or valves, (p.s.i.).

DP_b =Pressure drops across the bypass valve (p.s.i.).

P_f =The internal pressure required to precisely balance the effect of precompression loading of valve springs (p.s.i.).

f_d =The lift of discharge valves (in.).

f_b =The lift of bypass valve (in.).

R_d & R_b The spring rate or rates (i.e., load or loads per inch of spring deflection).

F_d & F_b The outward force on valve heads when they are under fluid pressure, due to deflection of flow (lbs.).

$Q'=Q_d$ (gal./min.).

$Q''=Q_b$ (gal./min.).

a =The discharge valve seat area (in.²).

a' =The bypass valve seat area (in.²).

D_d =The inside diameter of the discharge valve seat (in.).

D_b =The inside diameter of bypass valve seat (in.).

J_d =The precompression load on the discharge valve (lbs.).

J_b =The precompression load on the bypass valve (lbs.).

W_d =The weight of the discharge valve (lbs.).

W_b =The weight of the bypass valve (lbs.).

X =The position of the control plate 38a expressed as the fraction of the design maximum revolution per minute corresponding to any selected lesser revolution per minute of an internal combustion engine when the apparatus of the present invention is used therewith.

Z =The discharge spray tip travel into still air (in.).

The equations used are:

(1) $A_d=Q_d V$

(2) $A_b=Q_b V$

(3) $V_d=Q_d A'_d$

(4) $V_b=Q_b A'_b$

(5) $t=0.6n$

(6) $A_d=W_d[A/FwtV]$

(7) $A_b=[L-Q_d]aV$ (8) $F=[Q's][{(26.9)^2a}]$

(9) $M=RQV\sqrt{58.5(J+W-1.3F)3/2}$

(10) $f=[(J+W-1.3F)R][\sqrt{M(1+2M/100)+1-1}]$

(11) $A'_d=3.1416f_d D_d$

(12) $A'_b=3.1416f_b D_b$

(13) $DP_d=[R_d f_d][0.7854(D_d)^2]$

(14) $DP_b=[R_b f_b][0.7854(D_b)^2]$

(15) $Z=V_d t$

(16) $L=QX$

The following is an illustrative example of the application of the foregoing equations when a variable fuel control as postulated for the apparatus of the present invention is calculated for a four stroke internal combustion engine with the hereinafter listed dimensional and speed characteristics, operating under the given ambient atmospheric conditions:

1. A single cylinder of 30 in 3 displacement.

2. Speed range from 500 to 6,000 r.p.m.

3. Maximum input of fuel of 0.92 in.³.

4. Duration period of one discharge or injection = time t , corresponds to 120° of crank revolution.

5. Ambient air temperature at primary adjustment is 75°F.

6. Local barometric pressure and relative humidity are 29 inches of mercury and 40 percent, respectively.

7. Weight of the air under the foregoing conditions is 0.07383 lb./ft.³.

8. Specific gravity of the liquid fuel is 0.72 and the fuel weighs 0.02604 lb./in.³.

9. The effective lengths of the orifices 32 and 34 are assumed at 0.48 in. each.

10. The velocity of flow through the orifices 32 and 34 is, for this example, constant and made 10 in./sec.
11. Spring rates for the discharge and bypass valves are assumed equal and at 98.668 lbs./in. of deflection.
12. Preload $J_p = 0.25$ lb. and weight $W_p = 0.025$ lb.
13. Preload $J_b = 0.5$ lb. and weight $W_b = 0.038$ lb.
14. Inside diameter of the discharge valve seat = 0.12 in.
15. Outside diameter of the discharge valve seat = 0.16 in.
16. Inside diameter of the bypass valve seat = 0.169 in.
17. Outside diameter of the bypass valve seat = 0.25 in.
18. Weight of air inducted at 4,000 r.p.m. = 0.00128 lb.
19. Time $t = 0.005$ seconds.
20. A/F ratios as shown on Table I of FIG. 32.

A general method of determining the varying flow areas of the orifices 32 and 34 at all speeds of an internal combustion engine with variable input is as follows:

For a given Q with known or assumed air weights inducted and chosen speed intervals and A/F ratios desired or assigned for each such interval, A_d is found from equation (6). Then corresponding areas A_b are found from equations (16) and (7) provided the velocities of flow through the orifices 32 and 34 have been predetermined. The velocities of flow through the orifices 32 and 34 are selected so as to be identical to each other at each speed in order to satisfy the basic requirement of specific divisions of L, i.e., they may vary from speed to speed or they may be selected so as to be one velocity at all speeds. At each speed Q_d is found by multiplying the corresponding area A_d by the velocity of flow selected for that speed. In the given example a uniform speed of 10 inches per second is used. Q_b may be found by subtracting Q_d from L.

When the desired or chosen number of area determinations have been made, for example, at 250 revolution per minute intervals, that number of divisions may be laid off on the design length chosen for the orifices 32 and 34 drawn as an ordinate. Each speed interval so laid off corresponds to one position of the control plate 38a. When, for example, the speed range of an internal combustion engine is selected to be from 500 to 6,000 revolutions per minute and the common effective lengths of the orifices 32 and 34 is selected to be 0.48 inches, each speed interval will require a movement of the plate 38a of 0.02 inches or one twenty-fourth of 0.48 inches, 250 being one twenty-fourth of 6000. This unit of movement is identified on Table I of FIG. 32 as division factor.

The differences in the areas A_d of the orifice 32, identified on Table I of FIG. 32 as "Increments— A_d ", represents the increase or decrease in A_d due to a movement of the plate 38a. When such increment is divided by the proper division factor the mean width of the change in area is found. Such mean width laid off as abscissae at right angles to the ordinate of length at exact midpoints between the revolution per minute divisions for which the area increment has been found, establish a series of points through which the outline or configuration of the orifice 32 must pass. The shape and contour of the orifice 34 is determined in a similar manner.

It will be obvious that other methods of determining the shapes of the orifices 32 and 34 may be used. For example, the orifice 32 may be in the form of a simple rectangle with all area increments being equal, in which case the orifice 34 alone will be contoured corresponding to the requirements of the variable input.

A partial graphical representation summarizing typical values for orifice areas, related fluid flows, valve lifts, fluid velocities through the discharge and bypass valves and pressures, as calculated using the aforementioned equations, under the given conditions, may be found in Table I of FIG. 32 for varying speeds of the internal combustion engine. However, it should be noted that in making these calculations, the pressure drops across the distributor port 74c were found to be negligible and were thus omitted. Likewise, pressure losses due to friction in tubing connections and other passages were not included as they were felt to be design factors which could be compensated for by change in the precompression loads J on the valve springs.

It will be understood that modifications may be made without departing from the spirit of this invention and therefore no limitations other than those specifically set forth in the claims are intended to should be implied.

I claim:

1. A balanced fluid flow control apparatus for dividing sources of input fluid supply into at least two variable and controlled parts at least one of which is discharged at predetermined rates of flow to utilizing mechanisms, devices or processes, said rates of flow being controlled to range from minima to maxima, the latter including, in the extreme, the total input fluid as the discharge, which comprises:

- a. a hollow assembled body, the various parts of which are rigidly secured to each other along fluid tight joints;
- b. a primary source of fluid supply in communication with said body;
- c. a dispersion chamber within said body which receives said fluid supply, said dispersion chamber being provided with a nonreturn valve;
- d. at least one discharge orifice in association with said dispersion chamber, said discharge orifice being positioned in an orifice plate which is fixed within said assembled body;
- e. at least one bypass orifice in association with said dispersion chamber, said bypass orifice being positioned in said orifice plate;
- f. control means associated with said discharge and bypass orifices to regulate the flow of fluid therethrough comprising a slidable control plate upstream of said orifice plate and slidably in contact therewith and means to impart sliding movement to said plate, said control plate and said orifice plate being in intimate contact along the interface thereof so as to ensure that no leakage occurs along said interface, said plate being provided with a control port accurately aligned with respect to each said discharge and bypass orifices so as to precisely control the open areas of said orifices as said plate is moved;
- g. a surge chamber downstream from each said orifice;
- h. an outlet system associated with said surge chamber downstream from said discharge orifice to furnish resistance to the flow of said fluid;
- i. a bypass system associated with said surge chamber downstream from said bypass orifice to balance said outlet system, the flows therethrough and the pressures created therein; and
- j. relief means associated with said dispersion chamber to provide pressure relief in said apparatus when the input is greater than the sum of the desired maximum fluid flows through said discharge and bypass orifices and when said discharge and bypass orifices are rapidly closed and fluid is still supplied to said apparatus from said primary source.

2. The control apparatus according to claim 1, wherein said means to impart sliding movement to said plate comprises a slidable rod, one end of which is received within said assembled body and fixed to said control plate and the other end of which is provided with a transverse bar portion, spring means, one end of which is attached to said body and the other end of which is attached to said bar portion, said spring means counteracting the internal pressures within said apparatus tending to force said rod outwardly, and means to actuate said rod.

3. The control apparatus according to claim 1, wherein said orifice plate and said control plate are held in contact by at least one strip spring.

4. The control apparatus according to claim 2, wherein said dispersion chamber is provided with a slidable partition which divides said chamber into two compartments, one compartment for receiving that portion of said fluid supply which will be directed through said discharge orifice and one compartment for receiving that portion of said fluid supply which will be directed through said bypass orifice, said slidable partition being provided with downwardly extending prongs thereon, which limit the lateral movement of said nonreturn valve, and

with a baffle which directs said entering fluid supply into each of said compartments.

5 The control apparatus according to claim 1, wherein the area of each said bypass orifice is substantially determined by a solution of the following equation:

$$A_b = [L - Q_d] V$$

where

A_b = area of said bypass orifice;

L = fraction of maximum rate of input flows from said primary source (exclusive of any excess in said fluid supply released through said relief means) which flow is the proportionately divided quantity simultaneously flowing through said discharge and bypass orifices;

Q_d = rate of flow through said discharge orifice;

V = local velocity of fluid flow through said bypass orifice.

6. The control apparatus according to claim 1, wherein said bypass system comprises:

- a. a spring adjustable bypass valve associated with said bypass surge chamber;
- b. a bypass duct in communication with said bypass valve to return said bypassed fluid to said primary source of fluid supply; and
- c. means to adjust fluid flow rates through said bypass valve.

7. The control apparatus according to claim 6, wherein said means to adjust fluid flow rates through said bypass valve includes a spiral metallic strip associated with said spring adjustable bypass valve, said strip being accommodated to a desired temperature datum and remaining so until the tension in the spring in said bypass valve is adjusted so that said bypass valve is properly set for the desired flow rate therethrough, whereby subsequent temperature changes deviating from said temperature datum cause said strip to expand and contract in length with sufficient force and movement to adjust the tension in the spring of said bypass valve and to thereby set said bypass valve for the desired flow rate therethrough at such temperatures.

8. The control apparatus according to claim 7, wherein one end of a rotatable shaft bears against the plunger of said bypass valve compressing the spring thereof, and wherein the expansion and contraction in length of said strip rotates said shaft so as to adjust the tension in said spring.

9. The control apparatus according to claim 6, wherein said means to adjust fluid rates through said bypass valve includes an automatic adjustment for the flow rate through said bypass valve with changes in altitude comprising a flexible diaphragm associated with said spring adjustable bypass valve, said flexible diaphragm being exposed to atmospheric pressure on one side thereof and being sealed on the other side thereof at a desired pressure datum, whereby the tension in the spring in said bypass valve is adjusted by changes in the position of said diaphragm as the sealed in air expands and contracts.

10. The control apparatus according to claim 9, wherein one end of a rotatable shaft bears against the plunger of said bypass valve compressing the spring thereof, and wherein linkage means responsive to the changes in the position of said diaphragm as the sealed in air expands and contracts rotates said shaft so as to adjust the tension in said spring.

11. The control apparatus according to claim 1, wherein said relief means comprises a spring adjustable relief valve communicating with said primary source of fluid supply and means for delaying the closing of said relief valve so as to eliminate even momentary high internal pressures within said apparatus.

12. The control apparatus according to claim 11, wherein said relief valve is provided with a perforated guide which acts against the spring thereof and the stem of said relief valve extends through a stationary collar, terminating in a plunger located within a bore, and wherein said means for delaying the closing of said valve comprises a duct leading from said dispersion chamber to said bore, said duct being provided with a nonreturn valve which is operable at a lower pressure than said relief valve, whereby as internal pressures build up within said apparatus, fluid first opens said nonreturn valve in said

duct, flowing into said bore at substantially the same time at which said internal pressures open said relief valve, causing pressure to be exerted against said plunger and thus multiplying the movement of said relief valve, and regulating the speed of the closing of said relief valve because fluid trapped within said bore may only be gradually released from said bore through clearances between said stem and said stationary collar.

13. A balanced fluid flow control apparatus for dividing sources of input fluid supply into at least two variable and controlled parts at least one of which is discharged at predetermined rates of flow to utilizing mechanisms, devices or processes, said rates of flow being controlled to range from minima to maxima, the latter including, in the extreme, the total input fluid as the discharge, which comprises:

- a. a hollow assembled body, the various parts of which are rigidly secured to each other along fluidtight joints;
- b. a primary source of fluid supply in communication with said body;
- c. a dispersion chamber within said body comprising a fixed-in-place cylinder having at least one discharge orifice and at least one bypass orifice formed in the periphery thereof and being provided with a nonreturn valve;
- d. control means associated with said discharge and bypass orifices to regulate the flow of fluid therethrough, said control means comprising a rotating valve positioned within said cylinder, said valve traversing said discharge and bypass orifices, and means to impart rotation to said valve;
- e. a surge chamber downstream from each said orifice;
- f. an outlet system associated with said surge chamber downstream from said discharge orifice to furnish resistance to the flow of said fluid;
- g. a bypass system associated with said surge chamber downstream from said bypass orifice to balance said outlet system, the flows therethrough and the pressures created therein; and
- h. relief means associated with said dispersion chamber to provide pressure relief in said apparatus when the input is greater than the sum of the desired maximum fluid flows through said discharge and bypass orifices and when said discharge and bypass orifices are rapidly closed and fluid is still supplied to said apparatus from said primary source.

14. The control apparatus according to claim 1, wherein the area of each said discharge orifice is substantially determined by a solution of the equation:

$$A_d = W_a [A/F \cdot w \cdot t \cdot V]$$

where

A_d = area of discharge orifice;

W_a = weight of fluid of known rate of flow discharged into the utilizing mechanism in accordance with the requirements of said mechanism;

A/F = ratios of the weight of the discharged fluid to the weight of fluid supplied to said apparatus;

w = specific weight of the fluid supplied to said apparatus;

t = time period when the discharged fluid flows are intermittent;

v = flow velocity of fluid flow through said discharge orifice.

15. The control apparatus according to claim 1, wherein said outlet system comprises:

- a. a collector disc overlying and in communication with said surge chamber downstream from said discharge orifice, said collector disc having a central recessed area in its upper face;
- b. a rotatable distributor disc juxtaposed to said collector disc, said distributor disc having a recessed area in its underface which mates with the central recessed area in the upper face of said collector disc and a plurality of ducts projecting radially from the recessed area in said distributor disc, each of which terminates in a port in the upper face of said distributor disc;

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- c. gear train means operatively connecting said distributor disc to a power source to impart rotational movement to said distributor disc;
- d. a cap atop said assembled body overlying said distributor disc, said cap having a plurality of outlet ports corresponding to the requirements of the utilizing mechanism;
- e. means for aligning said ports in said distributor disc relative to said outlet ports in said cap; and

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- f. spring loaded outlet means associated with each of said outlet ports on said cap.

16. The control apparatus according to claim 13, wherein said means to impart rotation to said rotating valve comprises a rotatable shaft, one end of which is received within said cylinder and fixed to said rotatable valve allowing free movement and the other end of which has affixed thereto lever means to actuate the rotation of said valve.

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