

June 11, 1963

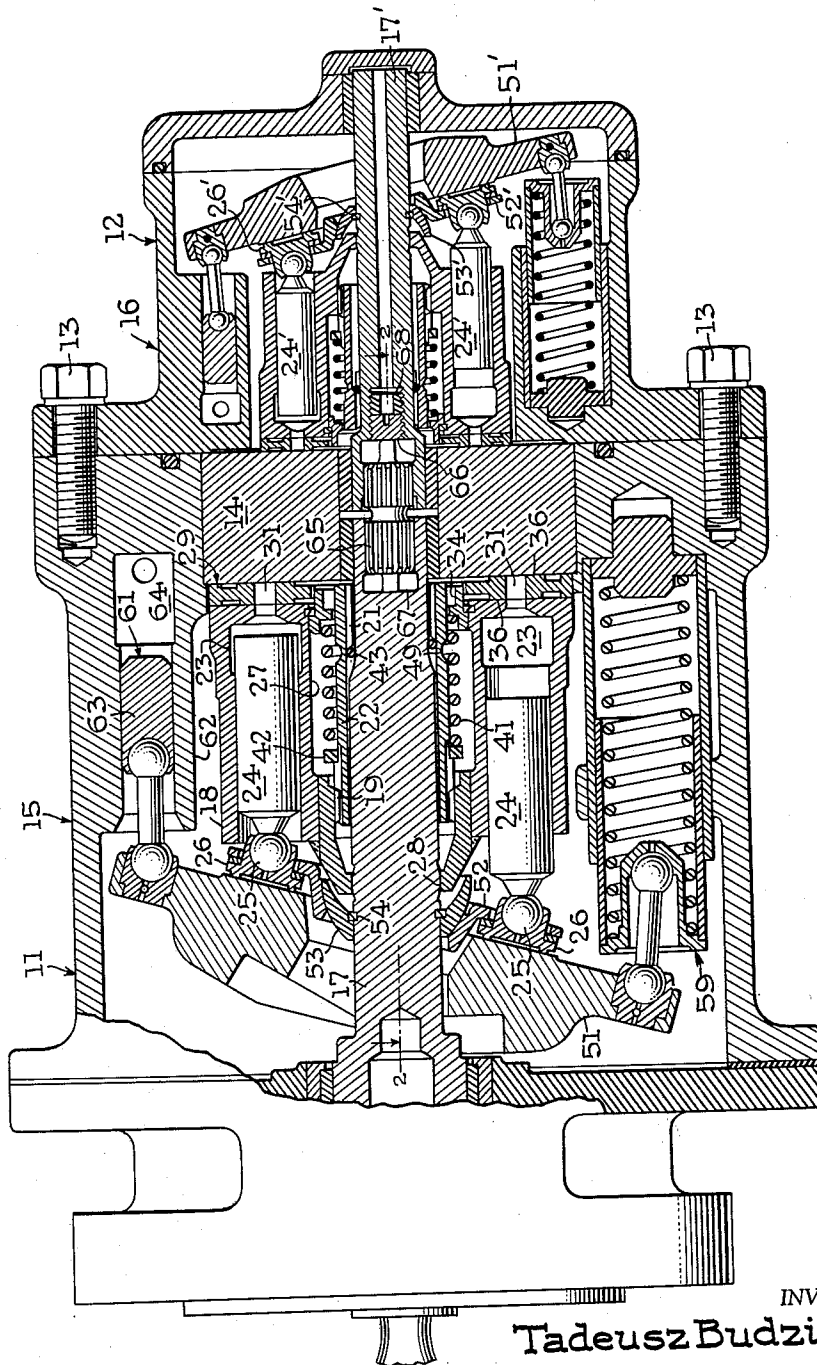
T. BUDZICH
PUMPING DEVICE

3,093,081

Filed Jan. 29, 1959

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FIG. 1



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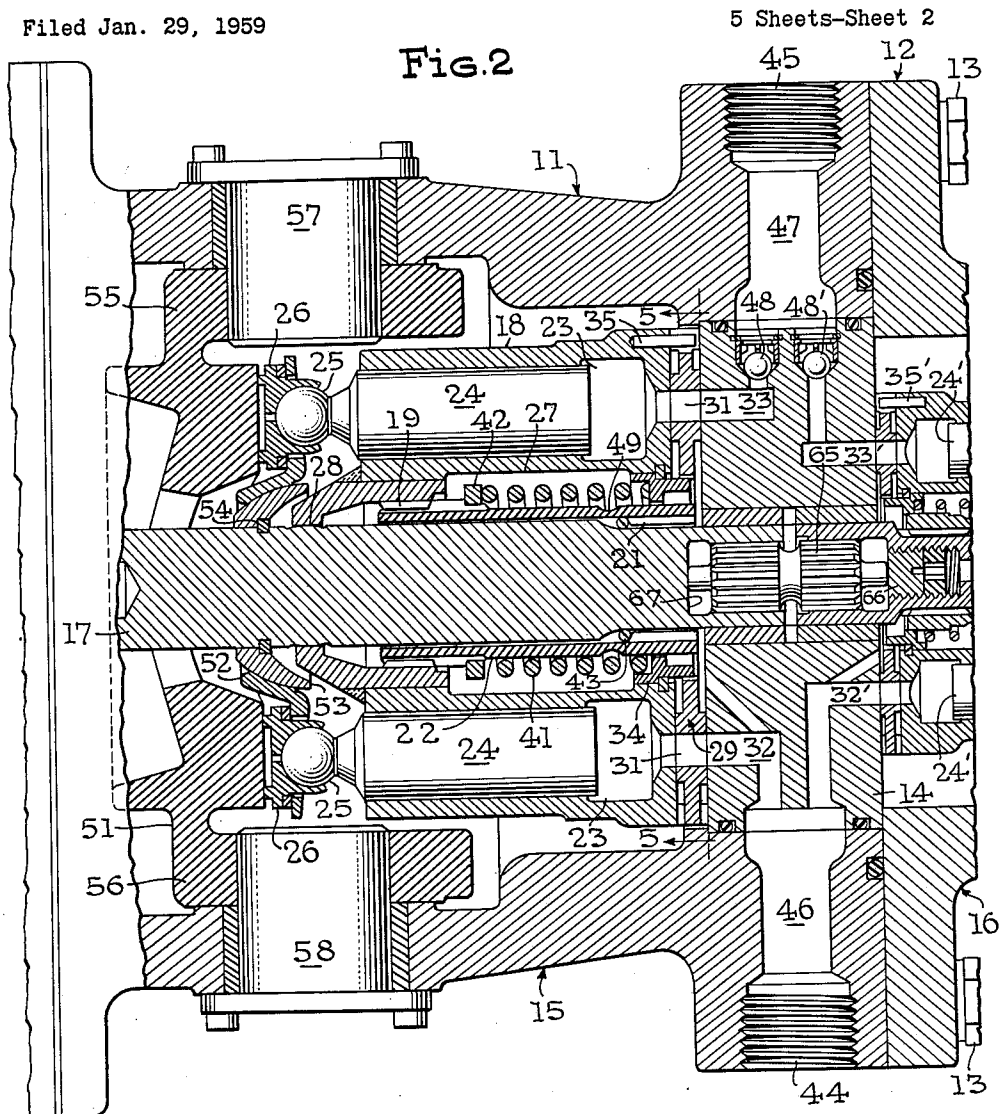
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Fig. 2



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Fig.3

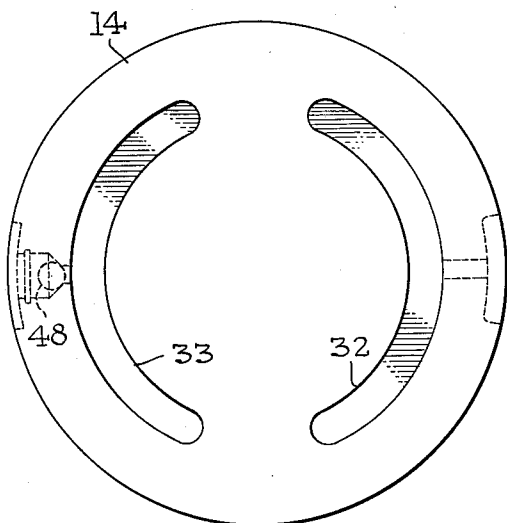


Fig.4

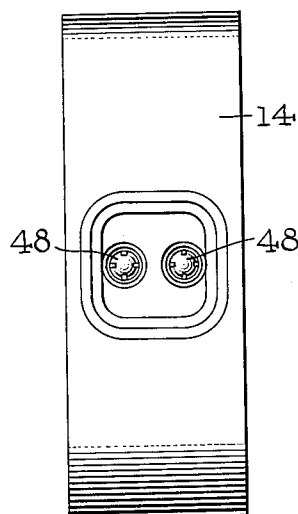


Fig.5

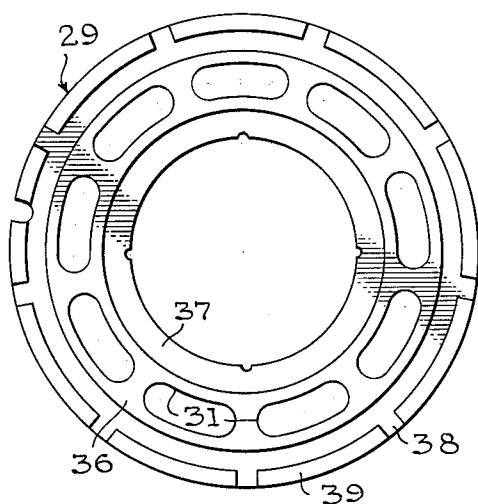
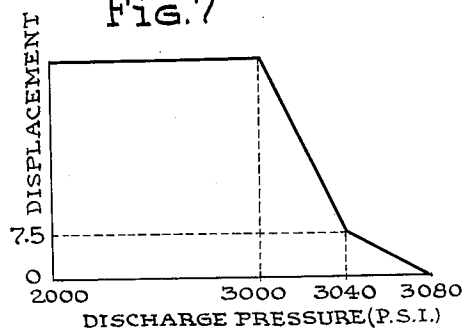


Fig.7



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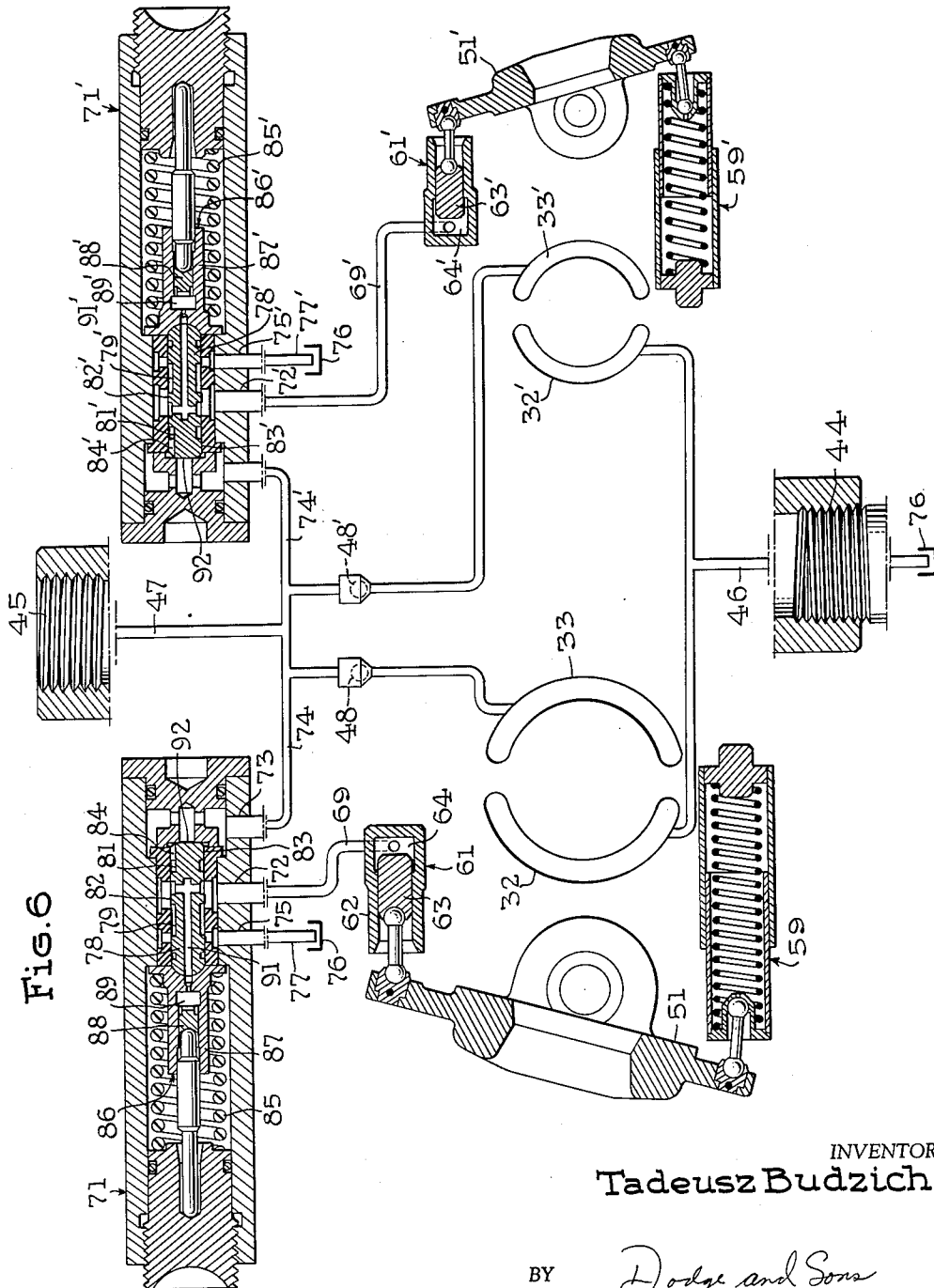
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Fig. 8

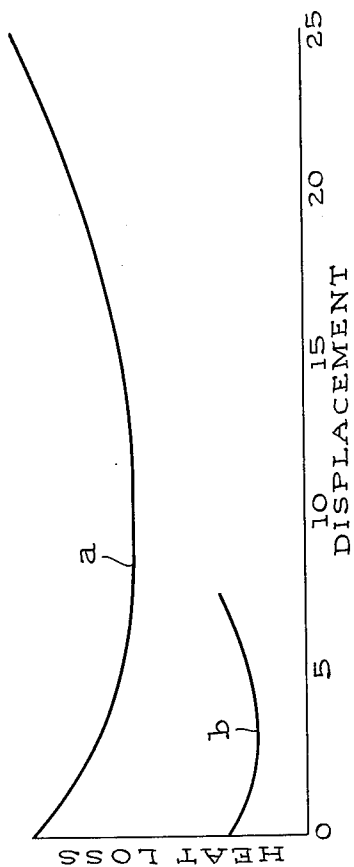
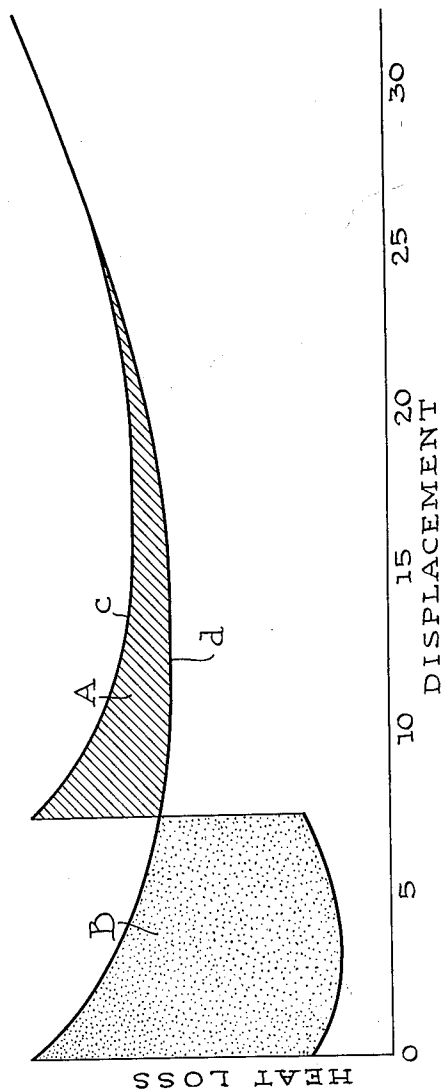


Fig. 9



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3,093,081

PUMPING DEVICE

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Filed Jan. 29, 1959, Ser. No. 789,995
10 Claims. (Cl. 103-11)

This invention relates to hydraulic pumping devices and particularly to those pumping devices which are suitable for use in aircraft.

In an aircraft hydraulic system, the demand for hydraulic fluid varies between wide limits and, during the major part of a typical flight, the demand is but a small fraction of the maximum demand. There is a continuing need for small, light-weight, reliable pumping devices which can operate efficiently under these conditions, and it is the object of this invention to provide such a device.

In its preferred form, the pumping device of this invention comprises two variable displacement pumps of the rotary cylinder barrel longitudinally reciprocating piston type which are driven from a common drive shaft and whose inlet and discharge ports are connected with common supply and delivery ports. The maximum displacement of each pump is so selected that their combined output is required to meet peak demands but one pump is capable of supplying the small demand which exists for the greater part of the operating cycle. Each pump is equipped with a discharge pressure compensator which functions to vary its displacement in inverse relation to discharge pressure (and thus in direct relation to demand), and these compensators are arranged to operate in sequence so that one pump continues to operate at full displacement until the displacement of the other pump has been reduced to zero. The pumping device also includes means for unloading the pistons of each pump after its displacement has been reduced to zero, and means for isolating each pump in the event of failure.

This type of pumping device is superior to the conventional device incorporating only a single pump in the following respects:

(1) Since the maximum speed of a pump is, in general, limited by its size, and since this device includes two pumping units, each of which is smaller than a single pump whose displacement is equal to their combined displacement, the maximum speed of this device is higher than that of the conventional single pump device. This results in a saving in space as well as weight.

(2) The smaller size of the individual pumping units permits operation at higher temperatures.

(3) The sequential operation of the discharge pressure compensators and the provision of the piston unloading means reduce the amount of heat transferred to the hydraulic oil during flight. This feature results in a reduction in the size and weight of the oil cooling equipment.

(4) The reliability of the present device is superior to the conventional pump because two independent pumping units are provided and failure of one does not affect the ability of the other to supply high pressure oil.

The preferred embodiment of the invention will now be described in detail with reference to the accompanying drawings, in which:

FIG. 1 is a partial axial sectional view of the pumping device.

FIG. 2 is an enlarged sectional view taken on line 2-2 of FIG. 1.

FIG. 3 is a view of one face of the valve member.

FIG. 4 is an elevation view of the valve member.

FIG. 5 is a view taken on line 5-5 of FIG. 2 showing the rear face of one of the valve plates.

FIG. 6 is a schematic diagram of the discharge pressure compensator circuits for the two pumping units.

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FIG. 7 is a graph showing the relationship between the displacement of the pumping unit and the discharge pressure.

FIG. 8 is a graph showing the relationship between heat loss and displacement for each of the two pumping units acting separately.

FIG. 9 is a graph showing the relationships between heat loss and displacement for the preferred pumping device and for a single pump having the same maximum displacement.

As shown in FIG. 1, the pumping device comprises a housing having separable sections 11 and 12 which are connected together by bolts 13 and which, when assembled, locate and rigidly hold a stationary valve member 14. Located on opposite sides of valve member 14 are two independent pumping units 15 and 16 which, except for size, are identical. Because of this, only the unit 15 will be described in detail.

The drive shaft 17, supported in housing section 11 and valve member 14, is connected in driving relation with the cylinder barrel 18 of pumping unit 15 by splines 19 and 21 and torque tube 22. The cylinder barrel 18 contains a circular series of nine longitudinal cylinder bores 23 which extend through the barrel and receive pistons 24. Each piston carries a spherical head 25 at one end for universally supporting a piston shoe 26. An axial bore 27 extends through the cylinder barrel and, at its left end, rests on the spherical enlargement 28 carried by drive shaft 17. The center of the surface of spherical enlargement 28 is located at the point of intersection of the drive shaft and the plane of the centers of spherical piston heads 25. Enlargement 28 is in line contact with the surface of bore 27 and thus permits the cylinder barrel to tilt and to move longitudinally relatively to the shaft. The method of driving and supporting the cylinder barrel is more fully described and claimed in applicant's copending application Serial No. 656,574, filed May 2, 1957, now Patent No. 2,925,046 issued February 16, 1960.

Located between cylinder barrel 18 and stationary valve member 14 is a valve plate 29 containing nine small arcuate passages 31 which are arranged to connect the cylinder bores 23 with the arcuate inlet and discharge ports 32 and 33 formed in valve member 14 as the cylinder barrel rotates. The valve plate 29 is located radially by a sleeve 34 and is connected in driven relation with the cylinder barrel 18 by pin 35. The front and rear faces of valve plate 29 are provided with a land 36, leakage grooves 37 and 38, and dynamic pads 39, as shown in FIG. 5. This type of valve plate is more fully described and claimed in applicant's copending application Serial No. 775,437, filed November 21, 1958, now abandoned. A spring 41, reacting between snap ring 42 carried by torque tube 22 and sleeve 34, maintains the mating faces of cylinder barrel 18, valve plate 29, and valve member 14 in sealing engagement. This spring load imposed on the torque tube is transmitted to the shaft by splines 21 and snap ring 43.

The arcuate ports 32 and 33, in valve member 14, communicate with common supply and delivery ports 44 and 45 via passages 46 and 47, respectively. A check valve 48 is located between passage 47 and arcuate port 33 for preventing reverse flow from delivery port 45. Check valve 48 cooperates with the leakage path provided along the front and rear faces of valve plate 29 to unload pistons 24 when the displacement of pumping unit 15 is zero. This valve also cooperates with shear section 49, formed in torque tube 22, to isolate unit 15 in the event of failure.

Pistons 24 are reciprocated in a known manner by cam plate 51 and nutating plate 52. Nutating plate 52 is seated on a collar 53 having a spherical outer surface

which engages a similarly shaped recess formed in the nutating plate. The center of this spherical surface is coincident with the center of spherical enlargement 28. Snap ring 54, seated in a groove formed in drive shaft 17, prevents longitudinal movement of collar 53 under the action of the piston inertia loads, and thus serves to transmit these loads into shaft 17.

Cam plate 51 is supported in housing section 11 by yokes 55 and 56 and trunnions 57 and 58 for angular movement about an axis extending in a direction normal to the axis of drive shaft 17 and intersecting that axis at the center of spherical enlargement 28. The angular position of the cam plate determines the length of the strokes of pistons 24, and the cam plate is biased toward its maximum stroke-establishing position by a spring plunger 59. The cam plate 51 is moved in the opposite direction against the bias of spring plunger 59 by control motor 61. This motor comprises a cylinder 62, a piston 63 connected with the cam plate, and a working chamber 64.

The two shafts 17 and 17' are connected in driving relationship by a splined coupling 65 whose opposite ends bear against a plug 66 threaded in a bore formed in shaft 17' and a wall 67 formed in shaft 17. When the pump is assembled, the plug 66 is rotated and thus advanced to thereby force the shafts 17 and 17' to the left and right, respectively, and cause snap rings 54 and 54', collars 53 and 53', and nutating plates 52 and 52' to move the piston shoes 26 and 26' into operative engagement with cam plates 51 and 51'. The adjusted position of plug 66 is maintained by a threaded locking plug 68. During operation, the inertia loads of pistons 24 and 24' oppose each other in coupling 65, and if these loads are equal, no force will be transmitted to the housing. On the other hand, if the inertia forces from the two pumping units are unequal, a net force (equal to the difference between the two) will be transmitted to the housing through the cam plate trunnions of that pumping unit having the lower inertia force. This method of bringing the piston shoes into engagement with the cam plate and of handling the piston inertia loads is more fully described and claimed in applicant's copending application Serial No. 665,387, filed June 13, 1957, now Patent No. 2,953,099 issued September 20, 1960.

As shown in FIG. 6, the working chamber 64 of the control motor 61 is connected by passage 69 with a control valve 71. This valve comprises a housing containing an outlet port 72 which is connected with passage 69, an inlet port 73 which is connected with delivery port 45 by passages 47 and 74, and an exhaust port 75 which is connected with a sump 76 by passage 77. A valve plunger 78, including annular grooves 79 and 81 and lands 82 and 83, controls communication between the outlet port 72 and the inlet and exhaust ports 73 and 75. A longitudinal slot 84, formed in land 83, provides continuous communication between inlet port 73 and groove 81. The valve plunger 78 has three operative positions, namely, a first position (shown in FIG. 6) in which groove 79 interconnects ports 72 and 75, a second position in which groove 81 and slot 84 interconnect ports 72 and 73, and an intermediate lap position in which land 82 isolates port 72 from the other two ports. The plunger 78 is biased toward its first position by a spring 85 and by a fluid pressure motor 86 which includes a cylinder 87, a piston 88, and a working chamber 89. The working chamber 89 is in constant communication with outlet port 72 through a passage 91 formed in the plunger. The plunger is moved toward its second position against this bias by the pressure fluid in inlet port 73 which acts upon the end face 92 of plunger land 83. This control valve is more fully described and claimed in applicant's copending application Serial No. 685,530, filed September 23, 1957, now Patent No. 2,921,560, issued January 19, 1960.

Valve 71, spring plunger 59, and control motor 61

form the discharge pressure compensator for pumping unit 15. When the pump is at rest, valve plunger 78 of control valve 71 will be in the position shown in FIG. 6 and working chamber 64 of control motor 61 will be vented to sump 76 via passage 69, outlet port 72, plunger groove 79, exhaust port 75, and passage 77. As a result, spring plunger 59 will move cam plate 51 to its maximum stroke-establishing position (shown in FIGS. 1 and 6). When the pump is running, the discharge pressure in port 45 (which is transmitted to control valve 71 by passages 47 and 74 and inlet port 73) acts upon the end face 92 of plunger 78 and urges this plunger to the left against the bias of spring 85. When discharge pressure rises to a certain value, hereinafter termed the "reference pressure," plunger 78 will have been moved to its lap position in which land 82 interrupts communication between ports 72 and 75 and will be held in that position against the bias of spring 85 by the pressure force developed at end face 92. When discharge pressure exceeds the "reference pressure," valve plunger 78 moves to the left from the lap position thereby causing groove 81 and slot 84 to interconnect ports 72 and 73. Pressure fluid is now transmitted to the working chamber 64 of control motor 61, and through passage 91 to the working chamber 89 of biasing motor 86. When the pressure in these two chambers rises to a value at which the sum of the force of spring 85 and of biasing motor 86 exceeds the force developed at end face 92, the valve plunger will move to the right toward its lap position. When it has again reached the lap position, the pressures established in working chambers 64 and 89 will be proportional to the difference between the discharge pressure in port 45 and the "reference pressure." Further increase in discharge pressure will produce proportional increase in pressure in working chambers 64 and 89. As explained in application Serial No. 685,530 (mentioned above), the factor of proportionality is the ratio of the area of end face 92 to the cross-sectional area of piston 88.

The pressure in working chamber 64, acting on control piston 63, develops a force which urges the cam plate 51 toward its neutral or zero stroke-establishing position (a vertical position as viewed in FIGS. 1 and 6) against the bias of spring plunger 59. The parts are so dimensioned that when the discharge pressure in delivery port 45 reaches the desired maximum, the cam plate 51 will be in its zero stroke-establishing position.

The discharge pressure compensators of the two pumping units are designed to operate in sequence; the compensator of pumping unit 15 moving cam plate 51 to its zero stroke-establishing position before the compensator of pumping unit 16 begins to shift cam plate 51' toward its corresponding neutral position. This sequential operation can be realized either by making the relationship between the cross-sectional area of control motor piston 63' and spring plunger 59' different from the relationship between the cross-sectional area of motor piston 63 and spring plunger 59 so that the pressure required by motor 61' to move cam plate 51' away from its maximum displacement-establishing position is greater than the pressure required by motor 61 to hold cam plate 51 in its neutral position, or by making the springs 85 and 85' different so that valve 71' establishes a reference pressure higher than that established by valve 71, or by a combination of these two methods. For convenience, the second method has been adopted in the following description.

In a typical aircraft hydraulic system, the demand for hydraulic fluid during a major portion of the flight is but a small fraction of the maximum demand. The present pumping device is particularly useful in a system of this type, as will be apparent from the following numerical example.

Let it be assumed that:

(1) The hydraulic system in which the pumping device is to be used creates a maximum demand of 32.5

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g.p.m. and that the demand does not exceed 7.5 g.p.m. during 80% of the time.

(2) The maximum desirable discharge pressure is 3,080 p.s.i.

(3) A pressure of 1,500 p.s.i. in working chambers 64 and 64' will enable motors 61 and 61' to hold cam plates 51 and 51' in their neutral positions.

(4) The control pressure differential (i.e., the pressure change in the working chamber of the control motor required to move the cam plate between its limiting displacement-establishing positions) of each compensator is 40 p.s.i.

(5) The proportionality factor of each control valve is 1.

Under these conditions, the two pumping units would be so designed that the maximum displacement of unit 15 is 25 g.p.m. and that of unit 16 is 7.5 g.p.m. The reason for this arrangement will be apparent after considering the discussion presented below.

The springs 85 and 85' of the two control valves 71 and 71' are designed to establish "reference pressures" of 1,540 p.s.i. and 1,580 p.s.i., respectively. This results in sequential operation of the two compensators and produces the flow rate-discharge pressure curve of FIG. 7. Thus, when discharge pressure is below 1,540 p.s.i., both working chambers 64 and 64' will be vented, both cam plates 51 and 51' will be in their maximum stroke-establishing position, and the displacement of the pumping device will be 32.5 g.p.m. When discharge pressure exceeds 1,540 p.s.i., control valve 71 establishes a pressure in working chamber 64 equal to the difference between discharge pressure and 1,540 p.s.i., and when discharge pressure exceeds 1,580 p.s.i., control valve 71' establishes a pressure in working chamber 64' equal to the difference between discharge pressure and 1,580 p.s.i. At a discharge pressure of 3,000 p.s.i., the pressure in working chamber 64 will be 1,460 p.s.i. (3,000-1,540), and under the assumed conditions, this pressure will cause control motor 61 to equalize the turning moments acting on cam plate 51. If demand for hydraulic fluid should now decrease so that discharge pressure rises to 3,020 p.s.i., control valve 71 will produce a 20 p.s.i. increase in pressure in working chamber 64 and control motor 61 will move cam plate 51 to a position in which the length of the strokes of pistons 24 is one-half of maximum and the displacement of unit 15 is 12.5 g.p.m. If demand should continue to decrease and discharge pressure rises to 3,040 p.s.i., control motor 61 will move cam plate 51 to its neutral position thereby reducing the displacement of unit 15 to zero. At this point, control valve 71' will have established a pressure in working chamber 64' of 1,460 p.s.i. (3,040-1,580) and the turning moments acting on cam plate 51' will be balanced. Any further increases in discharge pressure (decreases in demand) will produce proportional decreases in the displacement of unit 16 until when discharge pressure equals 3,080 the displacement of this unit will also be zero. As a practical matter, system leakage would prevent cut-off of unit 16 (i.e., prevent cam plate 51' from moving to its zero stroke position) unless the system included an accumulator.

When demand increases and discharge pressure drops, the displacement of unit 16 increases progressively, and when the cam plate of this unit is in its maximum stroke-establishing position, further increases in demand will automatically effect a progressive increase in the displacement of unit 15. When discharge pressure has again reached 3,000 p.s.i., the cam plates of both pumping units will be in their maximum displacement-establishing positions.

It should be observed that when the cam plate 51 of pumping unit 15 is in its neutral position and the displacement of this unit is zero, the pistons 24 will be unloaded. This unloading is attributable to the fact that the pressure fluid in cylinder bores 23 can leak across

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the front and rear faces of valve plate 29 and, since check valve 48 prevents reverse flow from port 45 to the cylinder bores, the pressure in those bores will decrease to housing pressure (sump pressure). This unloading action is an important feature because it means that when demand has dropped below 7.5 g.p.m., the only energy lost in the larger pumping unit is that due to windage; leakage and friction losses being eliminated. Check valve 48' will function to unload the pistons 24' of pumping unit 16 in those cases where accumulators are employed because then it is possible that system demand will be reduced to zero.

The advantages of the present invention will be apparent from a consideration of FIGS. 8 and 9. Curves *a* and *b* of FIG. 8 show the relationships between heat loss (expressed in terms of horsepower) and displacement for the pumping units 15 and 16, respectively. In FIG. 9, the individual heat loss curves *a* and *b* of FIG. 8 have been combined into curve *c* which illustrates the heat loss-displacement relationship for the present pumping device. It will be noted that below 7.5 g.p.m., the heat losses in the present device are those attributable to the unit 16; the losses in unit 15 being negligible since they are due only to windage. The curve *d* in FIG. 9 shows the heat loss-displacement relationship for a pumping device having the same maximum displacement (i.e., 32.5 g.p.m.) as the present device but having only one pumping unit.

A comparison of curves *c* and *d* of FIG. 9 shows that in the region A between 7.5 g.p.m. and 28 g.p.m., the heat losses in the present pumping device are greater than those occurring in the single unit pumping device, but that in the region B below 7.5 g.p.m., the present device is vastly superior. Since the pumping device will operate most of the time (80% in the example) below 7.5 g.p.m., it can be seen that this invention affords a substantial reduction in the amount of heat generated during a typical flight. This reduction in heat loss is of prime importance in present-day aircraft because of the extreme difficulty of dissipating heat in supersonic flight. Furthermore, since most of the heat generated in the pump is transferred to the hydraulic oil, the invention makes it possible to use smaller and lighter oil cooling devices (heat exchangers, etc.).

The check valves 48 and 48' serve not only to unload the pistons 24 and 24' when the cam plates 51 and 51' are in neutral, but also (in combination with shear sections 49 and 49') to isolate the pumping units 15 and 16 in the event of failure. Thus, for example, if the pistons 24 should seize in bores 23, shear section 49 in torque tube 22 would fail thereby permitting continued pumping action by unit 16. Check valve 48 in this case would prevent flow to arcuate port 33 and thus preclude leakage of hydraulic fluid along the front and rear faces of wear plate 29. Check valve 48' and shear section 49' will perform a similar function in the event of failure of pumping unit 16.

As stated previously, the drawings and description relate only to a preferred embodiment of the invention. Since many changes can be made in this embodiment without departing from the inventive concept, the following claims should provide the sole measure of the scope of the invention.

What is claimed is:

1. In combination, a plurality of variable displacement pumps connected to be driven by a common shaft, each pump having an inlet port and a discharge port; common supply and delivery passages connected with the inlet and discharge ports, respectively; a displacement-controlling element for each pump, each element being shiftable between minimum and maximum displacement-establishing positions; and control means responsive to the pressure in the delivery passage and connected with the displacement-controlling elements for varying the displacement of the pumps in sequence and in inverse relation to

the pressure in the delivery passage when that pressure is above a predetermined value.

2. In combination, two variable displacement pumps, each pump having an inlet port and a discharge port; a common shaft connected in driving relation with the two pumps; common supply and delivery passages connected with the two inlet and discharge ports, respectively; a displacement-controlling element for each pump, each element being shiftable between minimum and maximum displacement-establishing positions; first control means responsive to the pressure in the delivery passage and connected with one of the displacement-controlling elements for varying the displacement of the associated pump in inverse relation to the pressure in the delivery passage as that pressure varies between a low pressure limit and a high pressure limit; and second control means responsive to the pressure in the delivery passage and connected with the other of said displacement-controlling elements for varying the displacement of the pump associated with that element in inverse relation to the pressure in the delivery passage as that pressure varies between a low pressure limit and a high pressure limit, the low and high pressure limits of the second control means being higher than the corresponding limits of the first control means.

3. In combination, two rotary cylinder barrel longitudinally reciprocating piston pumps, each pump having a housing containing an inlet port, a discharge port, and an angularly adjustable cam plate for moving the pistons on their discharge strokes and for varying the lengths of these strokes; a common shaft connected in driving relation with the two pumps; common supply and delivery passages connected with the two inlet and discharge ports, respectively; first resilient means biasing one cam plate toward its maximum stroke-establishing position; second resilient means biasing the other cam plate toward its maximum stroke-establishing position; a first control motor for moving one of the cam plates toward a zero stroke-establishing position against the bias of the first resilient means; a second control motor for moving the other cam plate toward its minimum stroke-establishing position against the bias of the second resilient means; a first motor-actuating device responsive to the pressure in the delivery passage for energizing the first control motor progressively in accordance with the pressure in the delivery passage so that the associated cam plate is moved from its maximum to its zero stroke-establishing position as that pressure varies between low and high pressure limits; and a second motor-actuating device responsive to the pressure in the delivery passage for energizing the second control motor progressively in accordance with the pressure in the delivery passage so that the associated cam plate is moved from its maximum to its minimum stroke-establishing position as that pressure varies between low and high pressure limits, these low and high pressure limits being higher than the corresponding limits of the first motor-actuating means, whereby as delivery pressure rises the displacement of the two pumps is reduced in sequence.

4. The combination defined in claim 3 in which both the low and high pressure limits of the second motor-

actuating means are higher than the high pressure limit of the first motor-actuating means; and which includes means for unloading the pistons of the first pump when its cam plate is in zero stroke-establishing position.

5. The combination defined in claim 4 in which the means for unloading the pistons of the first pump comprise a check valve interposed between the discharge port of this pump and the common delivery passage for preventing reverse flow from the delivery passage to the port; and means defining leakage paths connecting the working chambers of the pistons of the first pump with the interior of the pump housing.

6. The combination defined in claim 5 including a second check valve interposed between the discharge port of the second pump and the common delivery passage for preventing reverse flow from the delivery passage to the port; and means defining leakage paths connecting the working chambers of the pistons of this pump with the interior of the pump housing.

7. The combination defined in claim 6 including shear sections located in the driving connections between the two pumps and the common shaft, whereby upon failure of either pump that pump will be isolated without impairing the pressure fluid supplying capability of the other pump.

8. The combination defined in claim 2 in which the maximum displacement of one pump is greater than the maximum displacement of the other.

9. The combination defined in claim 3 in which the maximum displacement of one pump is greater than the maximum displacement of the other.

10. The combination defined in claim 5 in which the maximum displacement of one pump is greater than the maximum displacement of the other.

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