

July 3, 1934.

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PUMP CONTROL MEANS

Filed Oct. 30, 1933

5 Sheets-Sheet 1

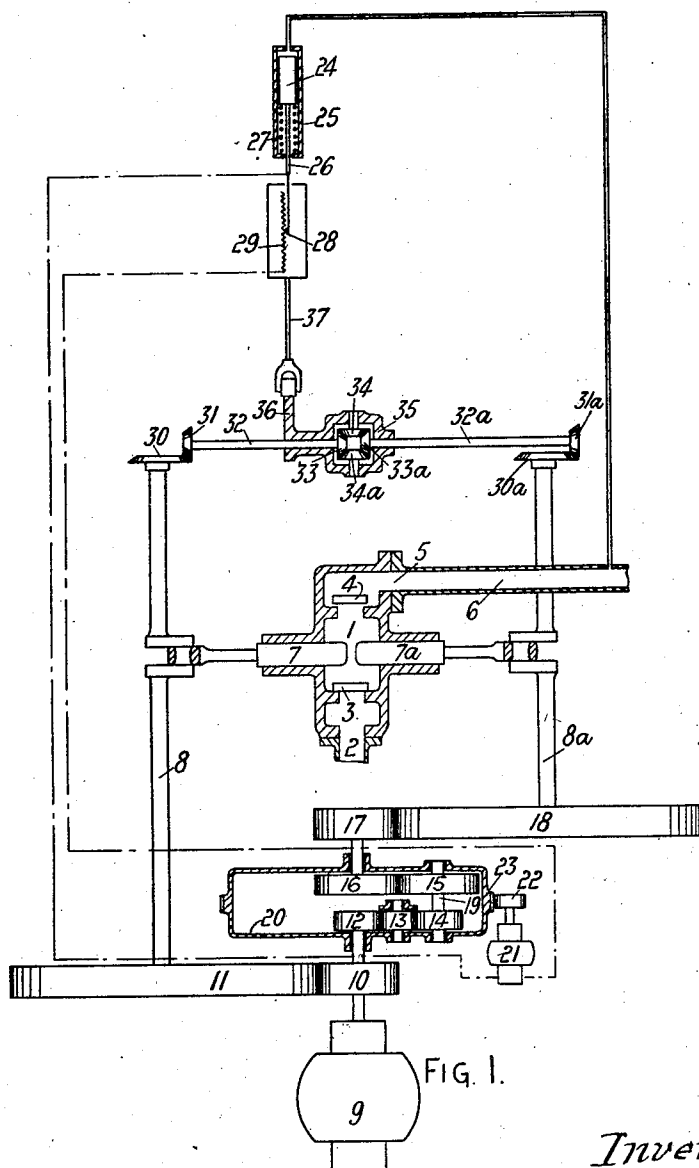


FIG. 1.

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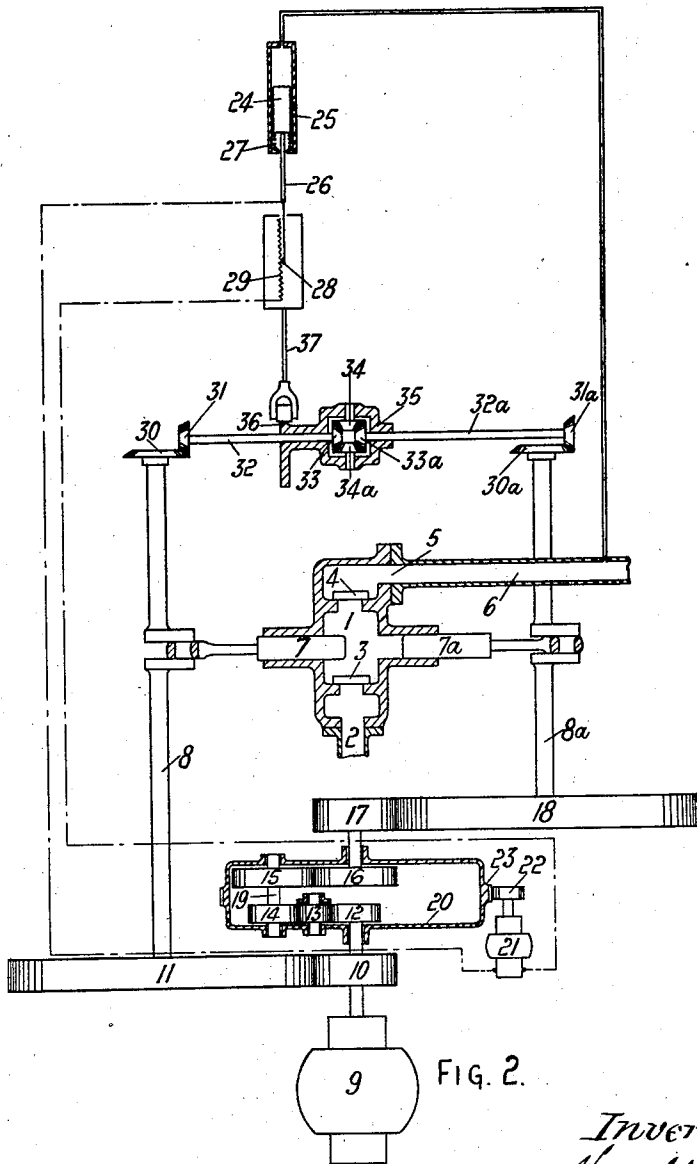
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PUMP CONTROL MEANS

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5 Sheets-Sheet 3

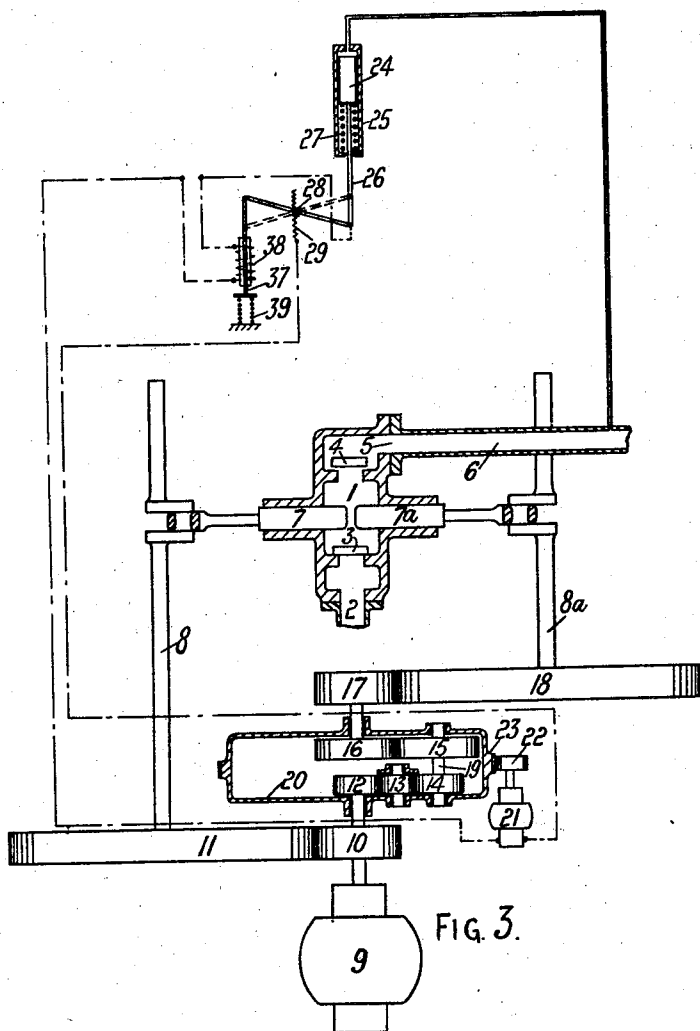


FIG. 3.

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5 Sheets-Sheet 4

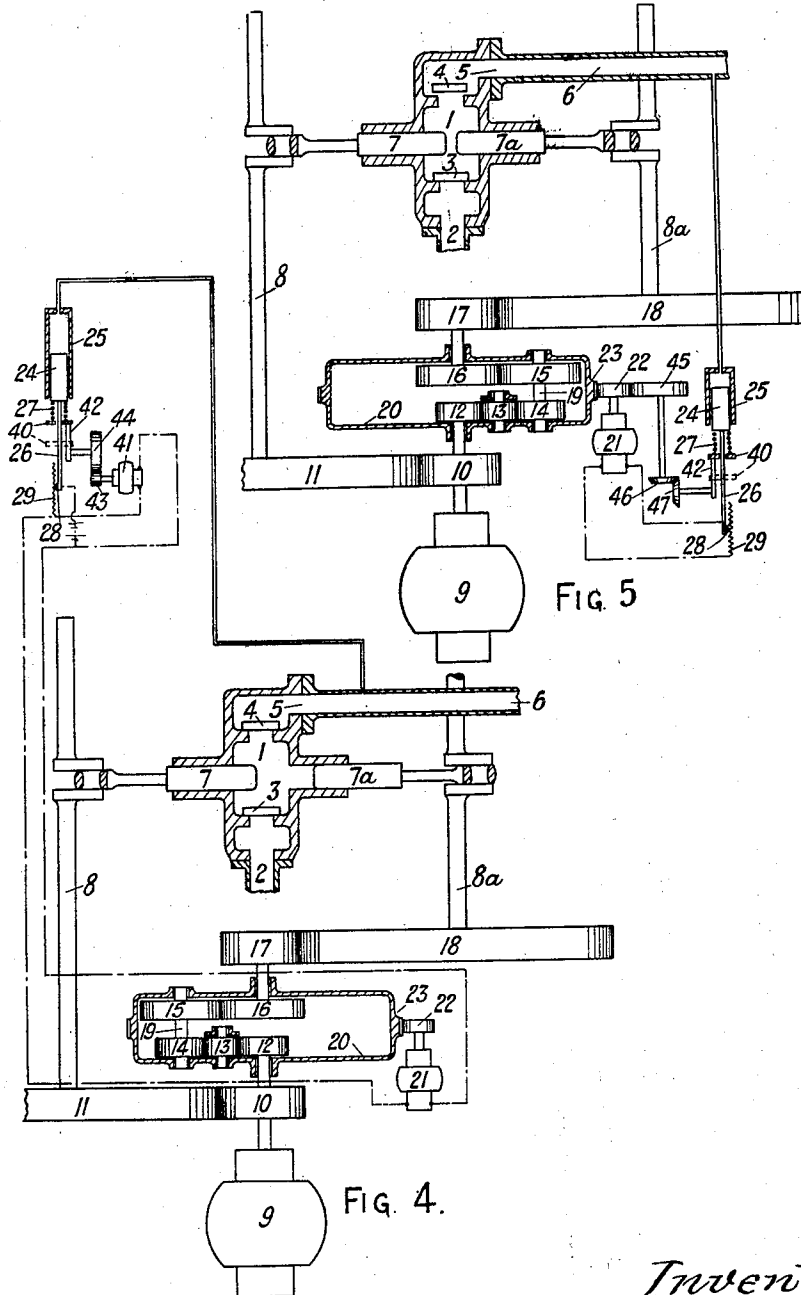


FIG. 5

FIG. 4.

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PUMP CONTROL MEANS

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5 Sheets-Sheet 5

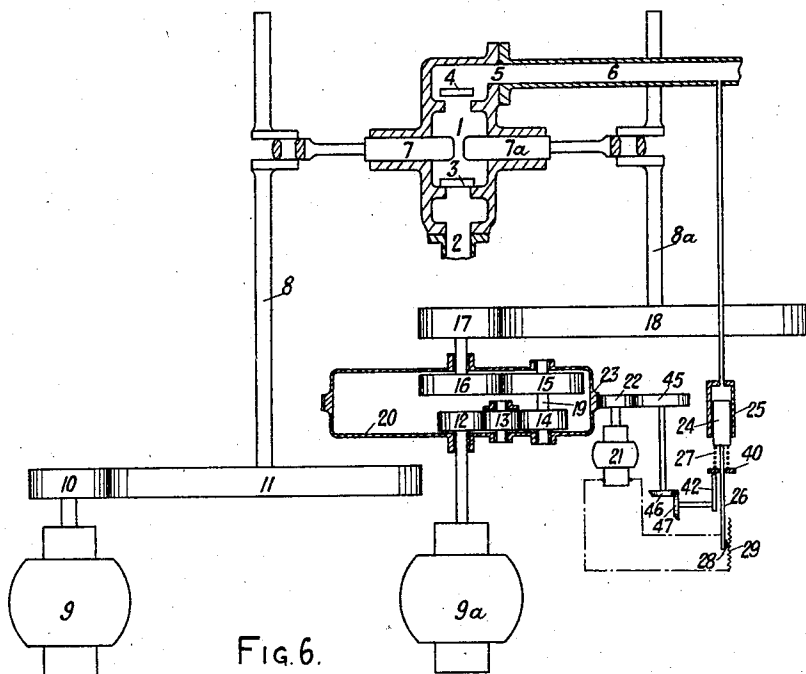


FIG. 6.

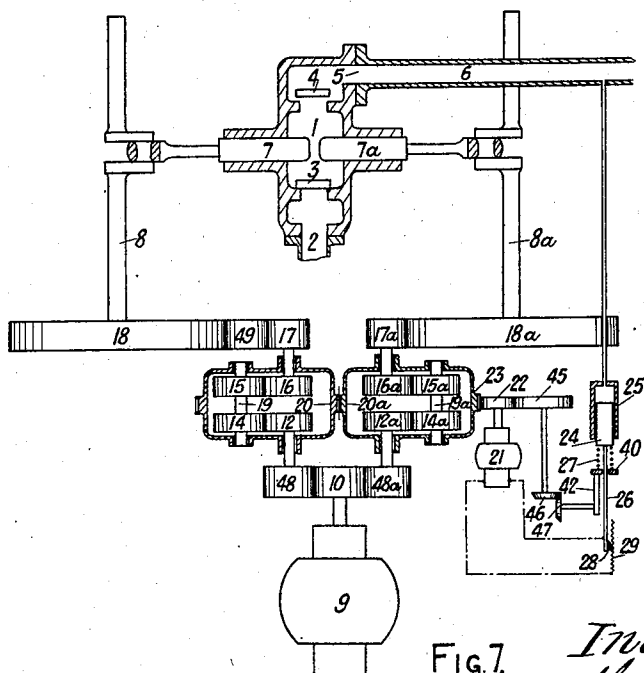


FIG. 7.

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UNITED STATES PATENT OFFICE

1,965,288

PUMP CONTROL MEANS

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In Great Britain October 25, 1932

2 Claims. (Cl. 103—37)

The present invention relates to reciprocating pumps, and, in particular, to reciprocating displacement pumps for discharging feed water into boilers of the class comprising at least one pair of displacers operating in a common chamber and driven by a pair of crank shafts through the medium of a gear train, at least one of said trains including an intermediate shaft mounted in a carrier rotatable by a motor to vary the phase relationship of the crank shafts.

The object of the invention is to provide improved means for driving and controlling such pumps whereby such pumps can be operated stably and satisfactorily in parallel under the control of pressure responsive means the action of which is to bring about variations in the phase relation or relative timing of the two pump shafts to establish a given phase relationship of the two shafts for a corresponding discharge pressure.

According to the present invention the speed of the motor for rotating the carrier is controlled by means comprising a device influenced by the pressure of the discharged fluid and by means operating in synchronism with variation of the phase relationship of the crank shafts.

The arrangement is such that the rotatable carrier is required to perform several rotations in order to change the pump capacity from the maximum to zero. The rotatable carrier may thus be of small dimensions, with a small inertia, so that the time lag in setting the rotatable carrier in motion, or in increasing or reducing its speed, is reduced to a negligible amount and undesirable fluctuations in the pump discharge pressure are avoided.

To enable my invention to be clearly understood it will now be described with reference to the accompanying drawings in which the same reference numerals denote similar parts.

Fig. 1 shows diagrammatically a displacement pump of the type described controlled in accordance with the invention, the plungers being shown in phase.

Fig. 2 shows diagrammatically the displacement pump represented in Fig. 1 but with the plungers shown out of phase.

Fig. 3 shows diagrammatically a displacement pump of the type described with the plungers in phase and the controller operating to give minimum and maximum delivery pressures at maximum and minimum outputs, respectively.

Fig. 4 shows diagrammatically a displacement pump of the type described with the plungers out of phase and the second part of the controller

operated by a small motor energized by the flow of current to the secondary prime mover.

Fig. 5 shows diagrammatically a displacement pump of the type described with the second part of the controller operable by gearing from the secondary prime mover.

Fig. 6 shows diagrammatically a displacement pump in which the pump shafts are driven by separate motors.

Fig. 7 shows diagrammatically a displacement pump in which the secondary prime mover imparts oppositely directed rotational movements to two carriers, one included in each shaft driving train.

Referring to Figs. 1 and 2, a pump driven and controlled in accordance with the invention comprises in its simplest form a body 1 containing fluid, a suction branch 2 for the admission of fluid, a suction valve 3 and a discharge valve 4 for controlling the admission and discharge of fluid which is delivered through the outlet branch 5 and delivery pipe 6. A pair of co-acting plungers 7 and 7a operate within the body 1 to vary the fluid content of the same. The plungers 7 and 7a are reciprocated by crankshafts 8 and 8a. When the plungers 7 and 7a are so reciprocated that they commence their displacement stroke at the same moment, the capacity of the pump is a maximum and the phase difference of the two crankshafts is nil. When the plungers 7 and 7a are so reciprocated that one plunger commences its suction stroke at the moment when the other plunger commences its displacement stroke, the capacity of the pump is approximately zero and the phase difference of the two shafts is 180°. By varying the phase difference of the two shafts between zero and 180° the pump capacity can be varied between a maximum and zero. The main prime mover is represented by an A. C. electric motor 9 which drives the crankshaft 8 at the required speed, through spur gears 10, 11 of fixed ratio. The crankshaft 8a is driven by the motor 9 through an epicyclic train of gearing 12 to 18, of which the gear wheels 14 and 15 are mounted on an intermediate shaft 19 carried in a rotatable frame 20 which can be rotated by the secondary motor 21 through the pinion 22 meshing with a gear wheel 23 formed on the rotatable frame 20.

The gear ratios may be arranged so that the frame 20 is stationary when the shafts 8 and 8a are rotating at the same speed, in which case the motor 21 will be started in the appropriate direction by the regulator when it is necessary to alter the phase displacement of the shafts 8 and

8a and stopped when the required phase displacement has been obtained.

Alternatively, the gear ratios may be so arranged that the motor 21 and frame 20 are rotating when the shafts 8 and 8a are operating at equal speeds. The difference in gear ratios to effect this, in practice, is too small to be clearly shown in the drawings.

The capacity of the pump is increased or decreased by making the appropriate change in the speed of the secondary motor 21 until the desired capacity is obtained, when the speed of the secondary motor is restored to the equilibrium speed.

Changes in the speed of the secondary motor are effected automatically by mechanism responsive to the pressure of the discharge of the pump and the phase displacement of the crankshafts. A piston 24 is arranged to operate in a cylinder 25, the upper end of which is subject to the discharge pressure in the pipe 6, and the lower end of which is subject to atmospheric pressure. The lower end of the piston 24 carries a spindle 26 which is surrounded by a spring 27 acting in opposition to the pump discharge pressure. Provision may be made for adjusting the compression of the spring. The spindle 26 is connected to a contact 28 acting on a rheostat resistance 29 in the supply of electric current to the secondary motor 21, so that movement of the contact 28, caused by movement of the piston 24 influenced by the pump discharge pressure acting against the spring 27, varies the rheostat resistance 29 and thereby varies the speed of the secondary motor 21. The crankshafts 8 and 8a are provided with bevel wheels 30 and 30a meshing with bevel wheels 31 and 31a fixed to shafts 32 and 32a. The shafts 32 and 32a carry gear wheels 33 and 33a meshing with loose wheels 34 and 34a carried in a frame 35 which is free to rotate about the shafts 32 and 32a.

When the crankshafts 8 and 8a are rotating at equal speeds, the frame 35 is stationary, but, if the speed of the crankshaft 8a is varied relatively to the speed of the crankshaft 8, the frame 35 is caused to perform a movement of partial rotation which is communicated by a device such as a cam 36 and a spindle 37 to the rheostat resistance 29.

Assuming that the pump is running at a given constant capacity and that the boiler feed regulator reduces the area for the flow of water into the boiler, the pump discharge pressure in the pipe 6 rises and causes the piston 24 to move the spindle 26 which moves the contact 28 on the rheostat resistance 29 so as to modify the speed of the secondary motor 21 and vary the phase relationship of the two crankshafts 8 and 8a as necessary to reduce the delivery of the pump. Due to the difference between the speeds of the two crankshafts 8 and 8a the frame 35 will perform a partial rotation and thereby actuate the spindle 37 to move the resistance 29 in the opposite direction until the secondary motor 21 returns to the equilibrium speed and the pump is operating at the reduced capacity corresponding to the reduced opening of the boiler feed regulator.

When the piston 24 is at the bottom of its stroke, the capacity of the pump will be zero, and, when the piston is at the top of its stroke, the pump will be operating at its maximum capacity, intermediate capacities corresponding to intermediate piston positions, the discharge pressure of the pump being determined by the compression of the spring 27.

With this arrangement the discharge pressure of the pump falls from a predetermined high pressure at zero capacity to a predetermined low pressure at full capacity.

The gearing 12 to 18 may be so arranged that the secondary motor is stationary when the crankshafts 8 and 8a are running at the same speed, the rheostat resistance being designed to start the secondary motor in a forward or reverse direction as necessary to effect any desired change in the phase relationship of the crankshafts, the resistance being brought to neutral position to stop the secondary motor when the desired pump delivery is reached.

Referring to Fig. 3, the movements of the spindle 37 are effected by a solenoid 38 acting against a weighted lever or a spring 39, said solenoid being contained in the circuit of the secondary motor 21, the acceleration and deceleration of which are effected by displacement of the contact 28 on the resistance 29.

The plungers 7 and 7a and the crankshafts 8 and 8a are shown in the phase position corresponding to maximum pump output and the means 24 to 29 and 37 to 39 for controlling the pump are so arranged that the delivery pressure of the pump is a predetermined minimum at maximum pump output and a predetermined maximum at minimum pump output, any intermediate pump output corresponding to an intermediate delivery pressure which is determined by the compression of the spring 27.

In the arrangement shown in Fig. 4, the spring 27 presses at one end on the piston 24 and at the other end on a movable spring cap 40. The gearing train 12 to 18 is so arranged that the secondary motor 21 is stationary when the crankshafts 8 and 8a are operating at equal speeds. A small motor 41 is arranged in the circuit of the secondary motor 21 so that the motors 41 and 21 are operated concomitantly. The motor 41 acting through gearing 43 and 44 operates a cam 42 to raise or lower the spring cap 40. The plungers 7 and 7a with the crank shafts 8 and 8a are shown in the opposite phase position at which the pump delivery is practically zero. Assuming that the boiler feed regulator, not shown, opens, the pressure in the delivery pipe 6 will fall and the piston 24 will rise due to the action of the spring 27 which will increase in length because of the reduced pressure acting on the piston 24. The upward movement of the piston 24 will raise the contact 28 on the resistance 29 and allow current to flow and start the secondary motor 21 in the requisite direction to increase the output of the pump. The flow of current to the secondary motor 21 also starts the motor 41 which moves the spring cap 40 to lower the piston 24 and contact 28 until the contact 28 reaches the neutral position on the resistance 29 and the secondary motor 21 stops when the pump is operating at the increased output required by the opening of the boiler feed regulator.

With the cam 42 arranged as shown, the spring cap 40 will be lowered as the pump output is increased, and, since the piston 24 and the contact 28 must return to the same position whenever the crankshafts are operating at equal speeds, the compression of the spring 27, and therefore the delivery pressure of the pump, will vary from a maximum at minimum pump output to a minimum at maximum pump output.

I may so arrange the cam 42 that the spring cap 40 is raised whereby to increase the compression of the spring 27 as the pump output is in-

creased, the delivery pressure of the pump being thereby varied from a minimum at minimum pump output to a maximum at maximum pump output.

5 The rheostat resistance 29 is so arranged that movement of the contact 28 effected by movement of the piston 24 due to change in delivery pipe pressure causes the requisite motion of the secondary motor 21 to alter the pump output until
10 the delivery pressure changes and allows the piston 24 to restore the contact 28 to the neutral position with the compression of the spring 27 modified by the spring cap 40 to give the desired delivery pressure corresponding to the altered pump
15 output.

Referring to Fig. 5, the spring 27 presses at one end on the piston 24 and at the other end on a movable spring cap 40. The gearing train 12 to 18 is so arranged that the secondary motor 21 is stationary when the crankshafts 8 and 8a are operating at equal speeds. The secondary motor is connected by gearing 45 to 47 to the cam 42 which raises or lowers the spring cap 40.

25 The plungers 7 and 7a with the crankshafts 8 and 8a are shown in the in-phase position in which the pump delivery is a maximum.

Assuming that the boiler feed regulator, not shown, closes slightly, the pressure in the delivery pipe 6 will rise and the piston 24 will descend against the action of the spring 27 which will contract because of the increased pressure acting on the piston 24. The downward movement of the piston 24 will lower the contact 28 on the resistance 29 and allow current to flow and start the secondary motor 21 in the requisite direction to reduce the output of the pump. The motion of the secondary motor 21 communicated through the gearing 45 to 47 to the cam 42 will lower the spring cap 40. The pump output will continue to be reduced until the delivery pressure falls to that corresponding to the lower output required by the reduced opening of the boiler feed regulator, when the contact 28 will reach the neutral position on the resistance 29 and the secondary motor 21 will stop, the pump then operating at the required reduced output.

With the cam 42 arranged as shown, the spring cap 40 will be lowered as the pump output is reduced, and, since the piston 24 and the contact 28 must return to the same position whenever the crankshafts are operating at equal speeds, the compression of the spring 27, and therefore the delivery pressure of the pump, will vary from a maximum at maximum pump output to a minimum at minimum pump output.

60 The cam 42 and the spring cap 40 may be so arranged as to reduce the compression of the spring 27 as the pump output is increased, the delivery pressure of the pump being thereby varied from a maximum at minimum pump output to a minimum at maximum pump output.

In Fig. 5 the contact 28 is shown in the neutral position.

Fig. 6 shows an arrangement differing only from the arrangement shown in Fig. 5 in that the crankshafts 8 and 8a are driven by separate motors 9 and 9a, respectively.

Fig. 7 shows an arrangement including two intermeshing carrier frames 20 and 20a, the motor 9 driving the crankshaft 8 through the gear train 10, 48, 12, 14, 15, 16, 17, 49 and 18 associated with the frame 20 and driving the crankshaft 8a through the gear train 10, 48a, 12a, 14a, 15a, 16a, 17a, and 18a associated with the frame 20a. As will be understood, the arrangement is such that the secondary motor 21 imparts oppositely directed rotational movements to the two carrier frames 20 and 20a.

The device responsive to the delivery pressure of the pump has in all cases been represented by a piston acting against a spring, but it will be understood that any known pressure responsive device may be used to actuate the means for controlling the speed of the secondary prime mover.

It is to be understood that the invention in its broadest aspects is not limited to the arrangements described, since many changes may be made in the details of the parts without departure from the essence of the invention as defined in the appended claims.

I claim:—

1. A reciprocating pump, comprising, in combination, a pump chamber, at least one pair of displacers operating in said chamber, a pair of crank shafts for driving said displacers, prime mover means for effecting rotation of said crank shafts, gear trains interposed between said prime mover means and said crank shafts, at least one of said trains including an intermediate shaft, a carrier for said intermediate shaft, said carrier being rotatable to vary the phase relation of said crank shafts, the gear ratio of one train being different from the gear ratio of the other train such that said carrier is required to rotate not less than one complete revolution to effect 180° relative angular displacement of said crank shafts, a secondary prime mover for rotating said carrier which rotates continuously when the said crank shafts are operating at equal speeds, means for regulating the speed of said secondary prime mover, said speed regulating means including a part influenced by the delivery pressure of the discharged fluid, and a follow-up mechanism operating in synchronism with variation of the phase relation of said crank shafts.

2. A reciprocating pump according to claim 1 in which said speed regulating means includes a part controlled by the delivery pressure of the discharged fluid and a part comprising a differential gear operated by the phase displacement of the crank shafts.

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