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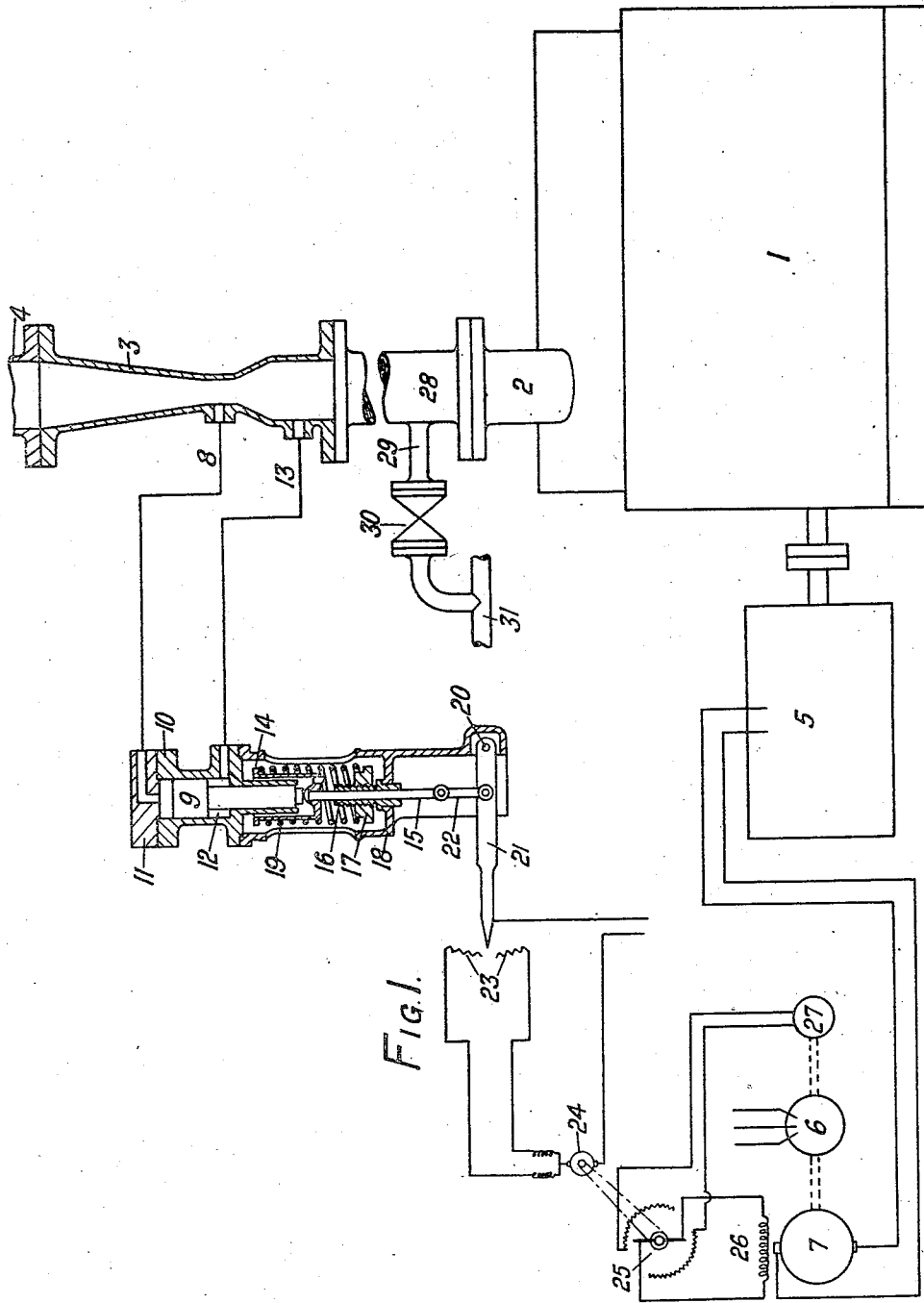
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5 Sheets-Sheet 1



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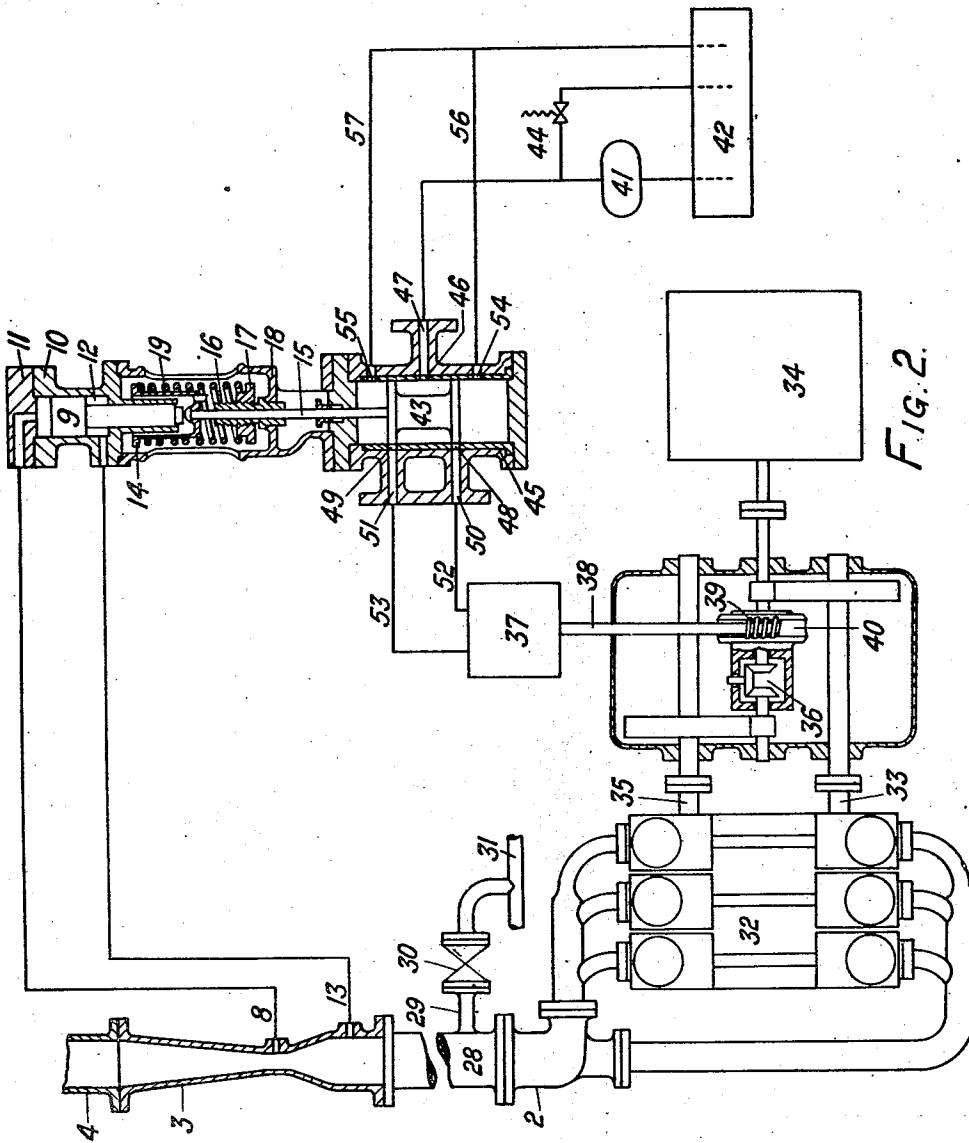


FIG. 2.

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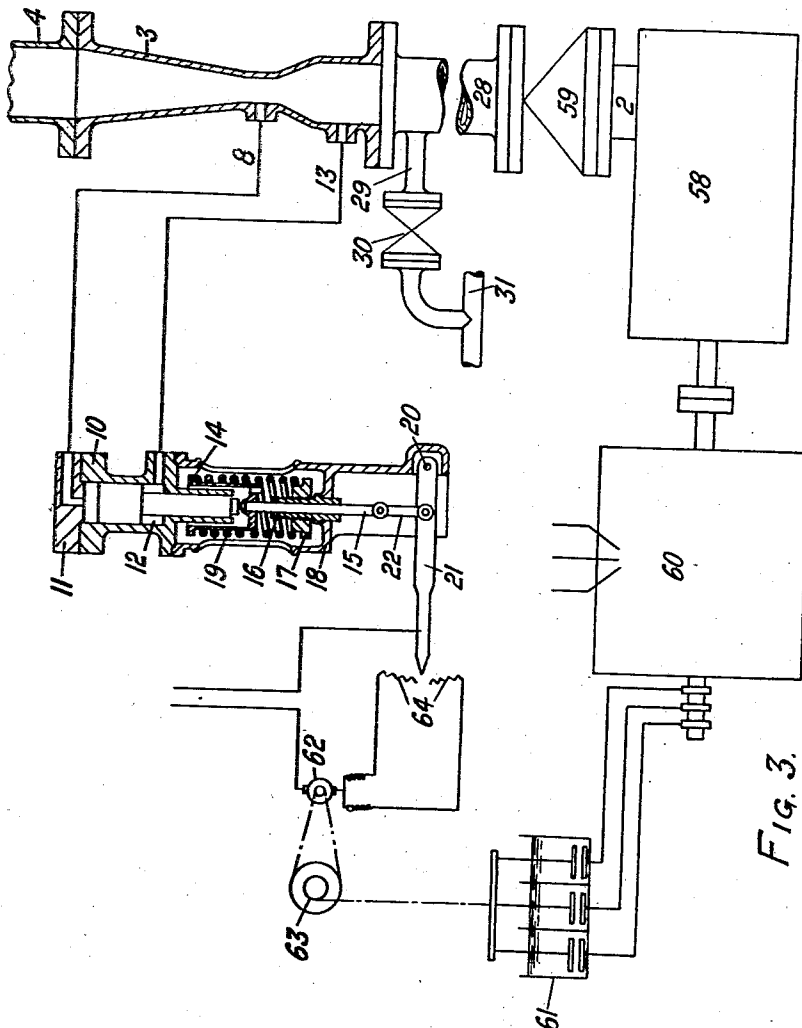


FIG. 3.

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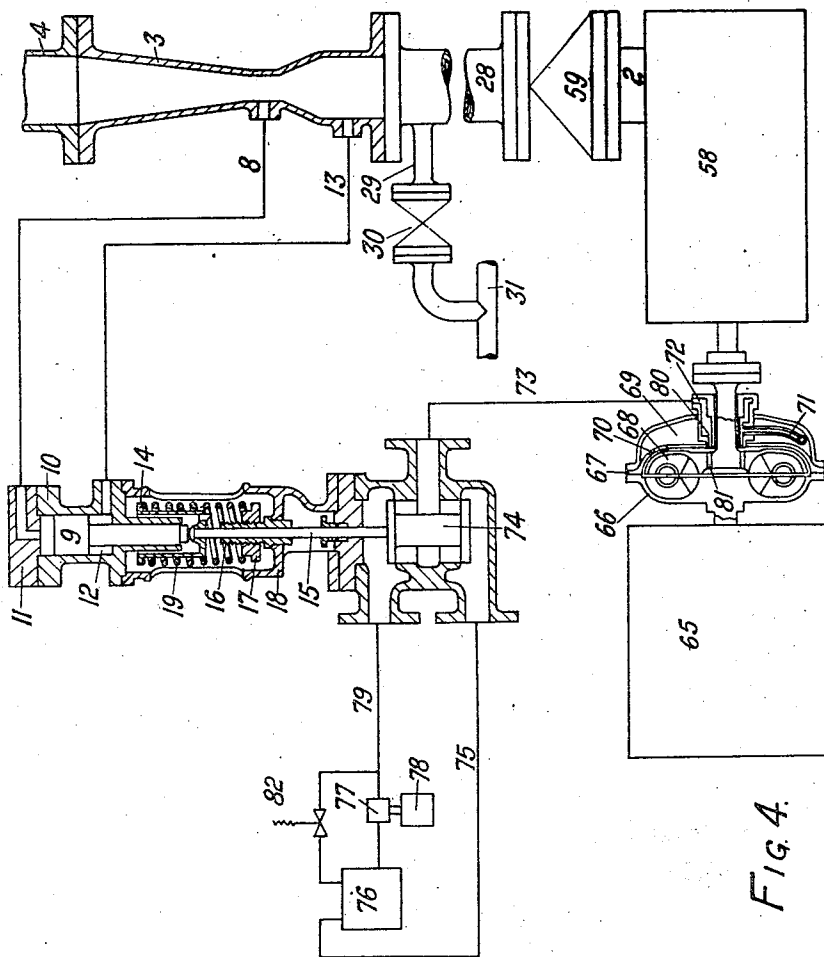
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5 Sheets-Sheet 5

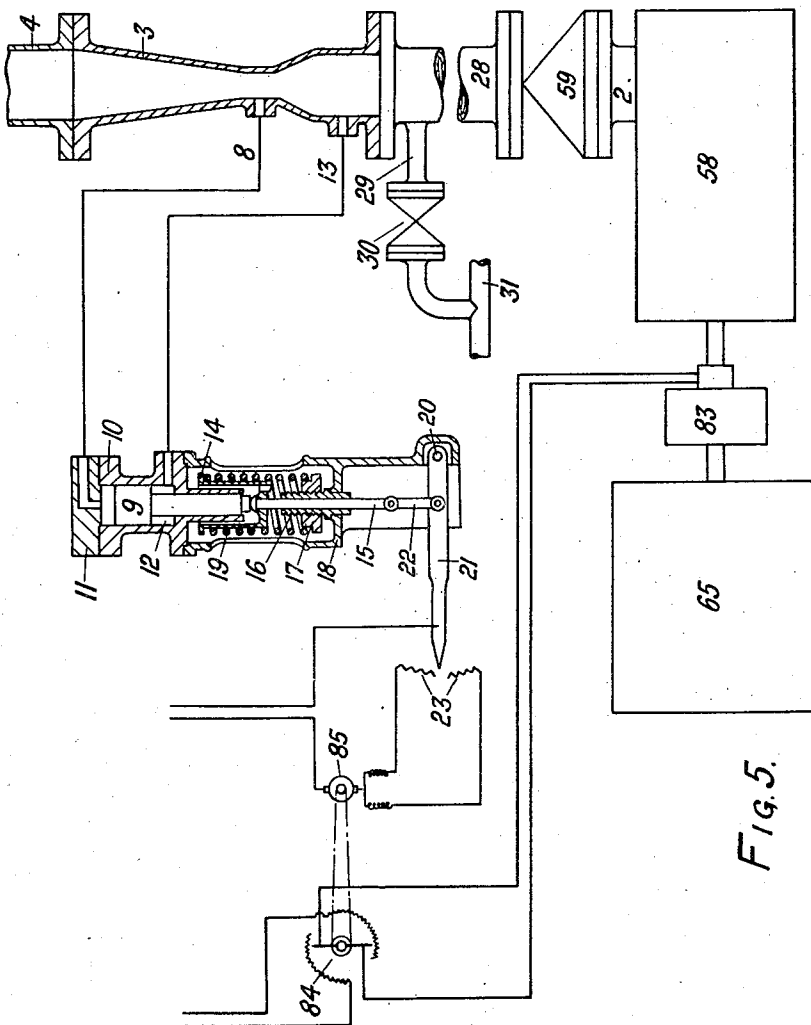


Fig. 5.

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UNITED STATES PATENT OFFICE

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2 Claims. (Cl. 103—35)

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This invention relates to means for controlling the speed of the driving member of a liquid pump, particularly of pumps used for discharging feed water into boilers.

For this duty, the resistance to be overcome when discharging water into a boiler is represented by the summation of the pressure in the boiler, the static lift from the feed pumps to the water level in the boiler, the pressure drop across the boiler feed regulator and the frictional resistances in the piping, heaters and valves through which the water flows from the feed pump to the boiler. These frictional resistances increase approximately as the square of the rate at which water is discharged through the feed system to the boiler. The boiler may be operated to maintain a pressure as nearly constant as possible in the boiler, or it may be so operated as to maintain a constant pressure at the main turbine inlet valve, in which case the boiler pressure rises with the load due to the pressure drop between the boiler and turbine which increases as the turbine load increases.

The static lift to the boiler is constant. Further, to obtain the best results from the boiler feed regulator, the pressure drop across the feed regulator should be maintained as nearly constant as possible.

It will be seen that the characteristic of the resistances to be overcome rises from a minimum when an extremely small quantity of water is flowing into a boiler to a maximum when the boiler feed pump is discharging its maximum quantity of water into the boiler and is thus a rising characteristic with a rise increasing in proportion to the square of the rate of feed flow.

When the feed pump is of the centrifugal type, the pump impeller or impellers must be so designed that the discharge pressure of the pump, when driven at a constant speed, falls continuously from that developed by the pump when no water is being discharged to that developed by the pump when discharging water at its maximum capacity. The relation between the discharge pressure of the pump and the quantity of water discharged by the pump is known as the pressure capacity characteristic and is termed a falling characteristic because it falls continuously from no load to full load.

Centrifugal boiler feed pumps must have a falling characteristic when operating at constant speed if they are to operate stably when running solo and stably when operating in parallel with

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other feed pumps which may be sharing the duty of supplying water to a battery of boilers.

The difference in pressure between the rising characteristic of the resistance in the feed system, neglecting the feed regulator, and the falling characteristic of the feed pump, must be absorbed by the feed regulator so that the drop in pressure across the feed regulator is very much greater at low loads than at the maximum load. This gives rise to difficulties in the operation of a boiler feed regulator at low loads, and a considerable amount of power is expended in excess of the minimum necessary to feed the boiler.

The minimum expenditure of energy is obtained when the feed pumps develop a discharge pressure characteristic which corresponds with the rising resistance characteristic of the feed system. For a centrifugal feed pump to develop a rising characteristic, it is necessary for the driving member of the unit, whether it be an electric motor or a steam turbine, to vary the speed of the centrifugal pump from a minimum at no flow to a maximum speed at the maximum flow, the speed rising continuously between these two points as necessary to give the desired discharge pressure at any given flow corresponding to the resistance to be overcome in the feed system to ensure that the water passes into the boiler. This desired variation in speed of the centrifugal pump can be obtained by the use of a steam turbine, a variable speed motor, when electrical supply is used as the driving power, or a variable speed coupling of the magnetic or hydraulic type interposed between a constant speed motor and the centrifugal feed pump.

Reciprocating feed pumps develop an output which is proportional to their speed of operation, and, at the same time, develop the pressure which is necessary to overcome the resistance in the feed system at any given flow.

In one arrangement of reciprocating feed pump, the pump comprises two rows of direct-acting plungers, each row being driven by a separate crankshaft through gearing which is interposed between the driving unit and the crankshafts. The plungers on each crankshaft are arranged in pairs, one plunger on each shaft co-acting with another plunger on the other shaft in a common chamber with common suction and discharge valves, so that the discharge from such pair of plungers is a maximum when the strokes of the two plungers are in phase and a minimum when the strokes of the two plungers are in opposite phase.

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The output of the pump is varied by varying the phase position of one crankshaft relative to the other crankshaft. The phase position of the crankshaft mentioned is determined by a secondary prime mover such as an electric motor or an oil motor driving a gear train interposed in the gearing between the main driving unit and the said crankshaft. The output of the feed pump in such an arrangement is, therefore, varied by starting and stopping the secondary prime mover or varying its speed on either side of a speed which gives equilibrium of the relative phase position of the crankshafts.

In another arrangement of a reciprocating feed pump, the speed of the feed pump is varied by use of a variable speed electric motor.

The present invention provides improved means for controlling boiler feed pumps where variation in output is obtained by varying the speed of the driving unit or a member of the driving mechanism, so that such pumps can be operated stably and satisfactorily when running solo or in parallel and with a minimum expenditure of power for the duty required.

The invention is illustrated in the accompanying drawings in which—

Fig. 1 shows diagrammatically a simple type reciprocating pump driven by a variable speed motor with means for controlling the speed of the motor in accordance with our invention.

Fig. 2 shows diagrammatically a reciprocating pump of the phase-changing type in which the phase position of two crankshafts is controlled by means in accordance with our invention.

Fig. 3 shows diagrammatically a centrifugal boiler feed pump driven by a variable speed motor the speed of which is controlled by means in accordance with our invention.

Fig. 4 shows diagrammatically a centrifugal boiler feed pump driven by a constant speed electric motor through an hydraulic coupling, the driven member of which varies in speed, the variation in speed being controlled by means in accordance with our invention.

Fig. 5 shows diagrammatically a centrifugal boiler feed pump driven by a constant speed motor through an electro-magnetic coupling, the speed of the pump being controlled by means in accordance with our invention.

Referring to Fig. 1, a reciprocating feed pump 1 discharges feed water through a discharge branch 2. Provided in the delivery line is a convergent-divergent Venturi tube 3 through which the feed water passes to the feed line 4 connecting to the boiler. The pump 1 is driven by an electric motor 5 in a variable speed Ward-Leonard arrangement comprising an induction motor 6 supplied with alternating current and driving a direct current generator 7 which supplies direct current to the motor 5. From the throat of the tube 3 a connection 8 is led to the top of a piston 9 having two diameters, the upper diameter being larger than the lower diameter. The piston 9 is movable in a cylinder 10 the upper end of which is closed by a cover 11 provided with an inlet communicating with the throat of the tube 3, so that the pressure prevailing in the throat of the tube is exerted on the top face of the larger diameter portion of the piston 9. The annular space 12 surrounding the lower part of the piston 9 within the cylinder 10 is connected to the inlet side of the tube 3 by a pipe 13, so that the annular area of the piston is subjected to the pressure at the inlet side of the tube 3. The smaller diameter portion of the piston 9 passes

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through the smaller bore of the cylinder 10 and engages an upper spring cap 14. The lower extremity of the piston 9 is subjected to atmospheric pressure.

The piston 9 rests on a spindle 15 which passes through the upper spring cap 14 and through a screw-threaded sleeve 16. The sleeve 16 is screwed into a lower spring cap 17 and is provided with a collar resting on the bottom of a spring cage casing 18.

The spring cap 17 constitutes a nut engaging the sleeve 16. Rotation of the sleeve 16 thus increases or reduces the stress of a spring 19.

Upward movements of the piston 9 are effected by the spring 19 in combination with the pressure at the inlet of the tube 3 applied to the annular area of the differential piston and atmospheric pressure on the lower end of the piston 9.

Downward movements of the piston 9 are effected by the pressure at the throat of the tube 3 applied to the top of the piston 9 in opposition to the upward forces.

The casing 18 carries a fulcrum 20 for a lever 21 which is connected by a link 22 with the spindle 15.

One end of the lever 21 works in contact with a rheostat 23 to control the direct current supply to the field of a servo-motor 24 which is arranged to position a rotary rheostat 25 controlling the field 26 of the generator 7 energised by the exciter 27 of the Ward-Leonard arrangement.

In the position corresponding to equilibrium of the pumping arrangement, the lever 21 is in the neutral position on the rheostat 23. With the lever 21 in this position, the speed of the motor 5 is constant. If the lever 21 is raised into the forward position of the rheostat 23, the effect is to increase the speed of the motor 5 and, conversely, if the lever 21 is lowered into the reverse position of the rheostat 23, the effect is to reduce the speed of the motor 5. An increase in the speed of the motor 5 increases the output of the pump 1, and a reduction in the speed of the motor 5 reduces the output of the pump 1.

The proportions of the convergent-divergent tube 3, the diameters of the differential piston 9, and the characteristic of the spring 19, are so chosen in relation to the discharge pressure characteristic desired from the pump 1 that they are always in balance at the neutral position of the lever 21 at any given flow between zero flow and the maximum flow required from the pump 1. We thereby constrain the pump 1 to follow the desired predetermined rising pressure capacity characteristic and the control is such that, for any given constant feed flow, the whole of the driving mechanism is maintained at a corresponding constant speed.

When the feed regulator on the boiler opens and thereby demands an increase in the quantity of water discharged by the pump, the discharge pressure of the pump falls slightly, and the spring 19 raises the lever 21 into the forward position of the rheostat 23, thereby speeding up the motor 5 and increasing the feed quantity discharged by the pump 1. As the quantity discharged by the pump increases, the feed discharge pressure rises and the increasing pressure acting upon the piston 9 causes the lever 21 to fall until it reaches the neutral position on the rheostat 23, when equilibrium speed of the feed pump is obtained in relation to the quantity of feed water required and the discharge pressure corresponding to that flow on the predetermined discharge pressure characteristic.

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If the feed supply to the boiler is reduced by the action of closing the boiler feed regulator, the pressure rises in the tube 3 and the piston 9 moves down against the action of the spring 19 and causes the lever 21 to move into the reverse position of the rheostat 23, thereby reducing the speed of the motor 5 and the quantity of water discharged by the pump 1 until the discharge pressures in the tube 3 fall until the spring acting against the piston 9 lifts the lever 21 into the neutral position again on the rheostat 23, when equilibrium conditions will again obtain with respect to the speed at which the reciprocating pump is running, the quantity of water required by the boiler, and the discharge pressure at that flow on the predetermined pressure capacity characteristic.

If two or more pumps are required to share the load in parallel, the section 28 of the discharge pipe between the pump discharge branch 2 and the tube 3 is provided with a branch 29 and an isolating valve 30 leading to a cross-connecting pipe 31, the pipe 31 being common to all the pumps which may be required to operate in parallel.

If the pump 1 is driven at a higher speed, and therefore delivers more water than another pump or pumps with which it is operating in parallel, then, owing to the piping resistance, the pressure in the section 28 of the discharge line of the pump 1 will be higher than in the corresponding sections of the discharge lines of the other pump or pumps. This higher pressure will induce flow through the branch 29 and valve 30 to the inter-connecting pipe 31, thereby reducing the flow through the tube 3. The reduced flow will reduce the pressure drop between the inlet and the throat of the tube 3 and will increase the pressure communicated by the connection 8 to the top of the piston 9, thereby causing the lever 21 to be lowered into the reverse position of the rheostat 23, reducing the speed of the motor 5 and reducing the quantity of water discharged by the pump 1. Similarly, if the pump 1 is driven at a lower speed and delivers less water than the other pump or pumps with which it is operating in parallel, then the pressure in the section 28 of the discharge line of the pump 1 will be lower than in the corresponding sections of the discharge lines of the other pump or pumps. This will cause water to flow from the inter-connecting pipe 31 through the valve 30 and the branch 29, thereby increasing the flow through the tube 3. This increased flow will increase the pressure drop between the inlet and the throat of the tube 3 and will reduce the pressure communicated by the connection 8 to the top of the piston 9, thereby causing the lever 21 to be raised into the forward position of the rheostat 23, increasing the speed of motor 5 and increasing the quantity of water discharged by the pump 1. In this way, therefore, the pump 1 is constrained to deliver its due share of the total flow required and the pumps in operation are constrained to operate stably and satisfactorily in parallel.

The branch 29 may be connected to the discharge branch 2 of the pump 1 instead of to the section 28 of the discharge line.

Referring to Fig. 2, a reciprocating feed pump 32 is of the phase-changing type in which there are two crankshafts, each driving a line of direct-acting plungers.

One crankshaft 33 is driven at constant speed by a motor 34 through gearing which is interposed between the driving unit and the crank-

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shaft. The driving unit may be a constant speed electric motor or a constant speed steam turbine. The other crankshaft 35 is driven by gearing from the same driving unit, but a gear train 36 of the differential type is interposed in the gearing between the driving unit and the crankshaft.

The plungers on each crankshaft line are arranged in pairs, one plunger on each shaft co-acting with another plunger on the other shaft in a common chamber with common suction and discharge valves, so that the discharge from such pair of plungers is a maximum when the strokes of the two plungers are in phase and a minimum when the strokes of the two plungers are in opposite phase relation.

The output of the feed pump is thus varied by modifying the phase position of the crankshaft 35 relative to the crankshaft 33 by means of the differential gear train 36. The phase position of the crankshafts in the arrangement shown in Fig. 2 is varied by means of a reversible hydraulic motor 37 through a driving shaft 38 and a worm 39 meshing with a worm wheel 40 on the differential gear 36.

An oil pump 41 is arranged to draw oil from a sump 42 and discharge through a control valve 43 to the hydraulic motor 37 from where it flows back through the control valve 43 to the oil sump 42.

The discharge system of the oil pump 41 is provided with a relief valve 44 set at a suitable pressure to enable the oil pump to discharge back to the oil sump in excess of the requirements of the hydraulic motor.

The feed pump 32 discharges feed water through the discharge branch 2 and there is provided in the delivery line a convergent-divergent tube 3 through which the feed water passes to the feed line 4 connecting to the boiler.

From the throat of the convergent-divergent tube 3 a connection 8 is led to the top of the piston 9 having two diameters, the upper diameter being larger than the lower diameter. The piston 9 is movable in a cylinder 10, the upper end of which is closed by a cover 11 provided with an inlet communicating with the throat of the tube 3, so that the pressure prevailing in the throat of the tube is exercised on the top face of the piston 9. The annular space 12 within the cylinder 10 is connected to the inlet side of the tube 3 by a pipe 13 so that the annular area of the piston is subjected to the pressure at the inlet side of the tube 3. The smaller diameter portion of the piston 9 passes through the smaller bore of the cylinder 10 and engages the upper spring cap 14. The lower extremity of the piston 9 is subjected to atmospheric pressure.

The piston 9 rests on the spindle 15 which passes through the upper spring cap 14 and through a screw-threaded sleeve 16. The sleeve 16 is screwed into the lower spring cap 17 and is provided with a collar resting on the bottom of a spring cage casing 18.

The lower spring cap 17 constitutes a nut engaging the screw-threaded sleeve 16. Rotation of the sleeve 16 thus increases or reduces the stress of the spring 19.

Upward movements of the piston 9 are effected by the spring 19 in combination with the pressure at the inlet of the tube 3 applied to the annular area of the differential piston and atmospheric pressure on the lower end of the piston 9.

Downward movements of the piston 9 are effected by the pressure at the throat of the tube

3 applied to the top of the piston 9 in opposition to the upward forces.

The valve 43 is shown in the neutral position when the forces acting on the piston 9 are in balance against the load of the spring at the predetermined position of the valve 43 for equilibrium of feed pump output. The valve 43 operates in a liner 45 which has a port 46 in communication with an oil inlet branch 47 in the discharge line from the oil pump 41. The liner also has ports 48 and 49 in communication with branches 50 and 51 respectively connected by pipes 52 and 53 with the hydraulic motor 37. Ports 54 and 55 are also provided in the liner and connected to pipes 56 and 67 through which oil can be discharged to the oil sump.

In the neutral position shown, the valve 43 covers the ports 48 and 49 and no oil is discharged to the hydraulic motor 37, the output of the oil pump 41 being discharged through the relief valve 44 to the sump 42.

When the feed regulator on the boiler opens and admits an increase in the quantity of water discharged by the pump, the discharge pressure of the pump 32 falls slightly, the spring 19 raises the valve 43, thereby admitting oil to flow from the branch 47 by way of the ports 48 and 49 to the hydraulic motor 37, the exhaust from the motor 37 passing by way of the pipe 52, and ports 48 and 54, to the oil sump 42.

The motion of the hydraulic motor 37 is such that the differential mechanism 36 is driven in the direction necessary to increase the quantity of feed water discharged by the pump 32. As the quantity discharged by the pump increases, the feed discharge pressure rises, and the increasing pressure, acting upon the piston 9, causes the valve 43 to return to the neutral position when the ports 48 and 49 are closed and the motion of the hydraulic motor ceases, equilibrium being obtained between the quantity of feed water discharged by the pump 32 and the discharge pressure corresponding to that flow on the predetermined discharge pressure characteristic.

If the feed supply to the boiler is reduced by the action of closing the boiler feed regulator, the pressure rises in the tube 3 and the piston 9 moves down against the action of the spring 19 and causes the valve 43 to admit oil from the discharge of the oil pump 41 by way of the ports 46 and 48 to the motor 37, which is driven in the reverse direction, oil exhausting from the motor 37 passing by way of the pipe 53, the ports 49 and 55, and the pipe 57 to the oil sump.

The hydraulic motor drives the differential gear 36 in the direction required to reduce the quantity of water discharged by the pump 32 until the discharge pressures in the Venturi tube fall until the spring acting against the piston 9 lifts the valve 43 into the neutral position, thereby cutting off the supply of operating oil to the motor 37 when equilibrium obtains between the quantity of water discharged by the pump, the quantity of water required by the boiler, and the discharge pressure at that flow on the predetermined pressure capacity characteristic.

The proportions of the convergent divergent tube 3, the diameters of the differential piston 9, and the characteristic of the spring 19 are so chosen in relation to the discharge pressure characteristic desired from the pump 32 that they are always in balance at the neutral position of the valve 43 at any given flow between zero flow and the maximum flow required from the pump.

We thereby constrain the pump 32 to follow the desired rising pressure capacity characteristic, and the control is such that, for any given constant feed flow, the whole of the driving mechanism is maintained at a corresponding constant speed.

Any variation in the quantity of the feed water required will call into play forces which will move the valve 43 as necessary to restore the equilibrium position for the required new rate of flow of feed water to the boiler.

If two or more pumps are required to share the load in parallel, the section 28 of the discharge pipe between the pump discharge branch 2 and the tube 3 is provided with a branch 29 and an isolating valve 30 leading to a cross connecting pipe 31, the pipe 31 being common to all the pumps which may be required to operate in parallel, as described with reference to Fig. 1.

Referring to Fig. 3, the feed pump 58 is of the centrifugal type discharging water through the discharge branch 2, a non-return valve 59, the convergent-divergent tube 3, and the feed discharge pipe 4 connecting to the boiler.

The pump 58 is driven by a variable speed electric motor 60 which may be of the slip ring alternating current type controlled by a liquid rheostat 61. The position of the rheostat 61 is controlled by the operation of servo-motor 62 through pulley mechanism 63, starting and stopping of the servo-motor 62 being effected by a rheostat 64 which is controlled by the lever 21 actuated by the differential pressure regulator which was described with reference to Fig. 1.

As previously described with reference to Fig. 1, a demand for an increased quantity of feed water causes the lever 21 to rise, thereby starting the servo-motor to alter the position of the liquid controller 61 to increase the speed of the motor 60 and the pump 58 until the required quantity of water is being delivered by the pump 58, at which point the lever 21 will be restored to the neutral position, and the servo-motor 62 will be stopped again. Conversely, if a smaller quantity of water is required by the boiler, the lever 21 will be lowered and the servo-motor 62 will operate in the opposite direction to lower the speed of the motor 60 and reduce the quantity of water supplied by the pump 58 until the lever 21 is restored to neutral position and the servo-motor 62 is stopped when the centrifugal pump 58 is discharging the required quantity of water to the boiler.

If two or more pumps are required to share the load in parallel, the section 28 of the discharge pipe between the non-return valve 59 and the tube 3 is provided with a branch 29 and an isolating valve 30 leading to a cross connecting pipe 31, the pipe 31 being common to all the pumps which may be required to operate in parallel as described with reference to Fig. 1.

Referring to Fig. 4, a centrifugal feed pump 58 discharges water through the discharge branch 2, the non-return valve 59, the convergent-divergent tube 3, and the discharge pipe 4 connecting with the boiler.

The pump is driven by a motor 65 running at constant speed coupled directly to the driving impeller 66 of a hydraulic coupling 67, of which the runner 68 is coupled directly to the shaft of the centrifugal pump 58.

The impeller 66 and the runner 68 are operatively interconnected hydraulically by oil which is circulated between the impeller 66 and the runner 68. When the hydraulic coupling is full

of oil, the speed of the runner 68 is approximately only 1½ per cent less than the speed of the impeller 66. As the quantity of oil in the coupling is reduced, the speed of the runner 68 decreases and is a function of the quantity of oil remaining in the coupling. Oil in the impeller 66 and the runner 68 has access to a space 69 by way of a small passage 70 of determinate cross sectional area from which flows a certain amount of oil under the action of centrifugal force. Immersed in the rotating oil within the space 69 is a stationary scoop 71 communicating with a chamber 72 surrounding the shaft of the runner 68. The arrangement is such that the scoop 71 collects and discharges oil from the space 69 into the chamber 72, whence the oil is discharged by way of a pipe 73, a control valve 74, and a pipe 75, to an oil reservoir 76.

An oil pump 77 driven by a motor 78 is arranged to draw oil from the reservoir 76 and to pump it back into the coupling 67 by way of a pipe 79, the control valve 74, the pipe 73, the chamber 72 and passages 80 and 81 into the runner 68.

The control valve 74 is actuated by a differential pressure regulator of the type described in detail with reference to Fig. 1. When the control valve 74 is in the neutral position as shown, there is no flow of oil from or to the coupling 67 by way of the pipe 73, and the discharge of the oil pump 77 is discharged by way of the relief valve 82 to the reservoir 76. When, however, a demand for an increased quantity of feed water causes the control valve 74 to be raised, oil flows by way of the pipe 79, the control valve 74, the pipe 73, etc., into the hydraulic coupling 67, thereby increasing the speed of the runner 68 and the speed of the centrifugal pump 58 until the required increased quantity of water is being delivered by the pump 58, at which point the control valve will have returned to the neutral position and the flow of oil into the hydraulic coupling will be cut off. Conversely, if a smaller quantity of water is required by the boiler, the differential pressure regulator will cause the control valve 74 to fall whereby oil is discharged from the coupling by way of pipe 73, the control valve 74 and the pipe 75 to the oil tank. A reduction in the quantity of oil in the hydraulic coupling 67 will cause the speed of the runner 68 to decrease and consequently the speed of the centrifugal pump will also be reduced until the quantity of water discharged to the boiler falls to the required quantity, at which point the differential pressure regulator will have restored the control valve 74 to its neutral position and flow of oil from the coupling will be cut off.

If two or more pumps are required to share the load in parallel, the section 28 of the discharge pipe between the non-return valve 59 and the convergent-divergent tube 3 is provided with a branch 29 and an isolating valve 30 leading to a cross connecting pipe 31, the pipe 31 being common to all the pumps which may be required to operate in parallel, as described with reference to Fig. 1.

Referring to Fig. 5, the centrifugal feed pump 58 discharges through the discharge branch 2, the non-return valve 59, the convergent-divergent tube 3, and the discharge pipe 4 connecting with the boiler.

The pump is provided with a motor 65 running at a constant speed and driving the centrifugal pump 58 through an electro-magnetic coupling 83. The speed of the driven half of the electro-

magnetic coupling 83 is controlled by a rheostat 84 which is operated by a servo-motor 85 which operates intermittently and is controlled by a rheostat 23. The rheostat 23 is actuated by a lever 21, the movements of which are effected by a differential pressure regulator as described with reference to Fig. 1.

As previously described with reference to Fig. 1, a demand for an increased quantity of water causes the lever 21 to rise, thereby starting the servo-motor 85 to alter the position of the rheostat 84, to increase the speed of the driven portion of the electro-magnetic coupling 83 and, therefore, the speed of the centrifugal pump 58, until the required quantity of water is being delivered, when the lever 21 will be restored to the neutral position and the servo-motor 85 will be stopped again. Conversely, if a smaller quantity of water is required by the boiler, the lever 21 will be lowered and the servo-motor 85 will operate in the reverse direction to lower the speed of the driven half of the electro-magnetic coupling 83 and so reduce the speed of the centrifugal pump 58, thereby reducing the quantity of water supplied by the pump until the lever 21 is restored to the neutral position and the servo-motor 85 is stopped when the centrifugal pump 58 is discharging the required reduced quantity of water to the boiler.

If two or more pumps are required to share the load in parallel, the section 28 of the discharge pipe between the non-return valve 59 and the convergent-divergent tube 3 is provided with a branch 29 and an isolating valve 30 leading to a cross connecting pipe 31, the pipe 31 being common to all the pumps which may be required to operate in parallel, as described with reference to Fig. 1.

In these several typical applications which have been described showing the application of our invention to control the appropriate driving member of a feed pump, it will be seen that, when the governing device is in the predetermined neutral position, the speed of the appropriate driving member is in equilibrium and the output of the feed pump is constant.

The proportions of the convergent-divergent tube 3, the diameters of the differential piston 9, and the characteristic of the spring 19, are so chosen in relation to the discharge pressure characteristic desired from the pump that they are always in balance at the neutral position of the governing device at any given flow between zero flow and maximum flow desired from the feed pump, the feed pump being constrained to follow a desired predetermined rising pressure capacity characteristic and, for any change in the demand of the quantity of feed water, the governing device speeds up or slows down the appropriate driving member until equilibrium is again obtained at the required flow of feed water and the corresponding discharge pressure.

We claim:

1. An apparatus for controlling the delivery pressure of a liquid pump at a predetermined value for each and every given rate of flow of the pump, comprising a cylinder, a piston reciprocable in said cylinder, one end portion of said piston being of larger diameter than the other, the end portion of smaller diameter extending through the adjacent end of the cylinder and being exposed to atmospheric pressure, said piston having an annular portion complementary to said portion of smaller diameter, a spring operative to oppose outward movement of the smaller

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end portion of the piston, a Venturi nozzle through which discharge from the pump passes, a conduit connecting the space in the end of the cylinder opposite the free end of the larger portion of the piston with a region of low pressure in the Venturi nozzle, a conduit connecting the space in the other end of the cylinder in which the annular portion of the piston reciprocates with a region of high pressure in the Venturi nozzle, whereby variations of pressure in said regions will cause movement of the piston in the cylinder, a rotary motor member connected to the pump, and means operated by movement of the piston to control the speed of the rotary motor member and thereby to control the pump from a given pressure datum against the atmosphere, the proportions of the Venturi nozzle, the larger and smaller diameters of the piston, and the characteristic of the spring being such with respect to a rising pressure capacity characteristic desired from the pump that the piston is stationary in a predetermined position for any given liquid flow at the corresponding discharge pressure of the pump on the pressure capacity characteristic.

2. An apparatus for controlling the delivery pressure of a liquid pump at a predetermined value for each and every given rate of flow of the pump, comprising a cylinder, a piston reciprocable in said cylinder, one end portion of said piston being of larger diameter than the other, the end portion of smaller diameter extending through the adjacent end of the cylinder and being ex-

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posed to atmospheric pressure, said piston having an annular portion complementary to said portion of smaller diameter, a spring operative to oppose outward movement of the smaller end portion of the piston, a Venturi nozzle through which discharge from the pump passes, a conduit connecting the space in the end of the cylinder opposite the free end of the larger portion of the piston with a region of low pressure in the Venturi nozzle, a conduit connecting the space in the other end of the cylinder in which the annular portion of the piston reciprocates with a region of high pressure in the Venturi nozzle, whereby variations of pressure in said regions will cause movement of the piston in the cylinder, a rotary pump-driving motor, means for driving the pump from said motor including a coupling having a driven member, and means operated by movement of the piston to control the speed of the driven member of the coupling and thereby to control the pump from a given pressure datum against the atmosphere, the proportions of the Venturi nozzle, the larger and smaller diameters of the piston, and the characteristic of the spring being such with respect to a rising pressure capacity characteristic desired from the pump that the piston is stationary in a predetermined position for any given liquid flow at the corresponding discharge pressure of the pump on the pressure capacity characteristic.

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