Feb. 28, 1939. D. LEWIS 2,148,761

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 1

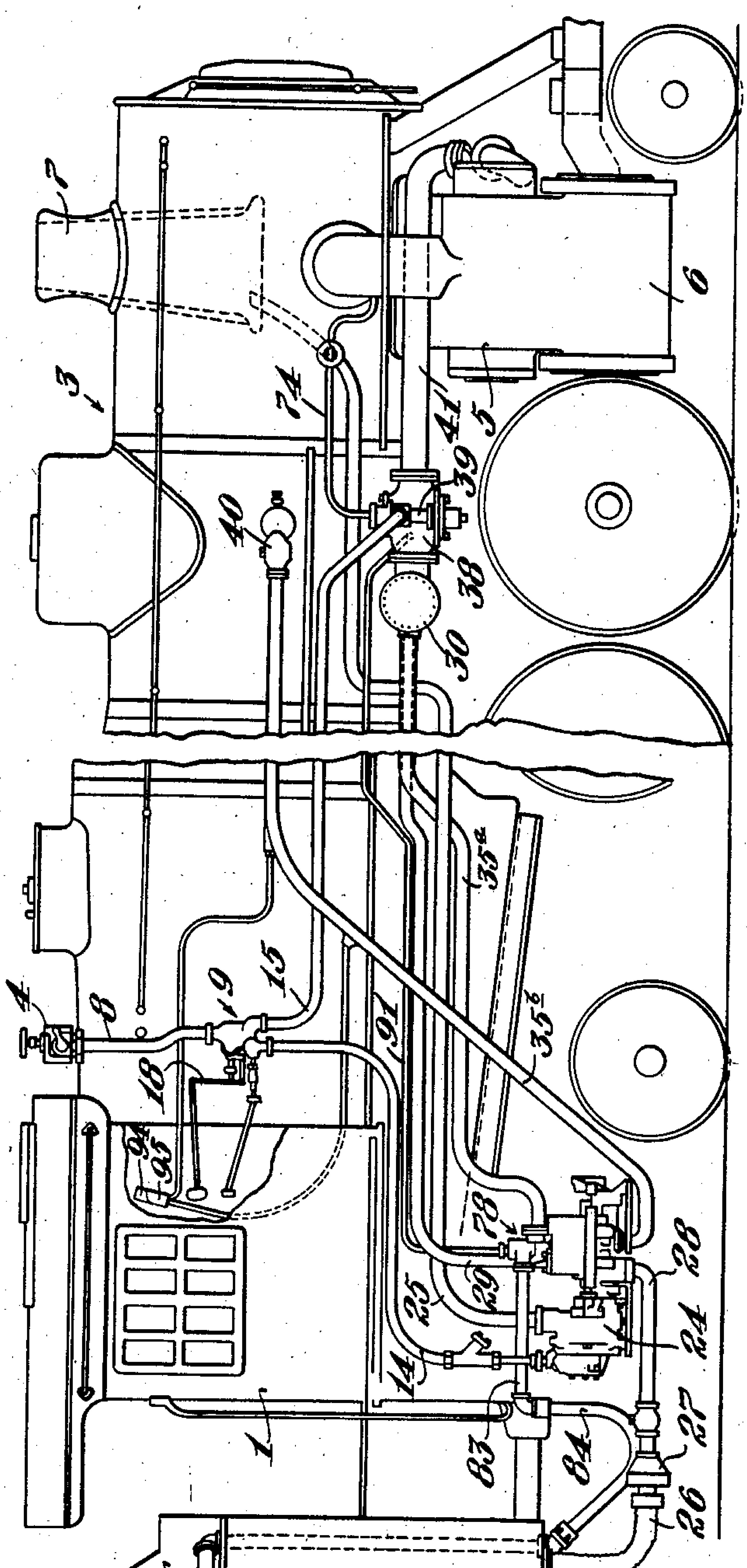
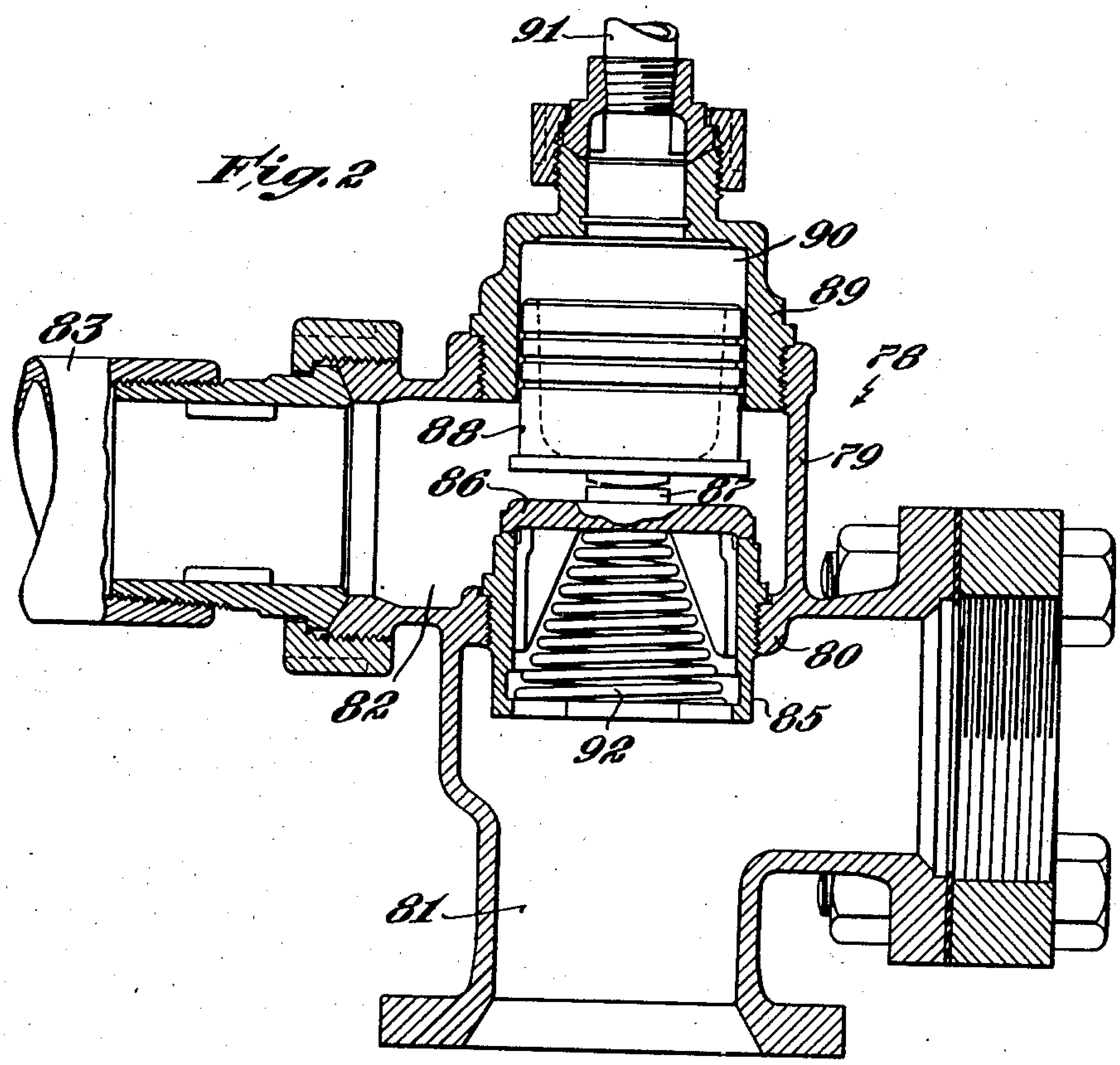


Fig. 1

INVENTOR.
Dartrey Lewis
BY Roberts Cushman & Woodberry
ATTORNEYS.

Fig. 2



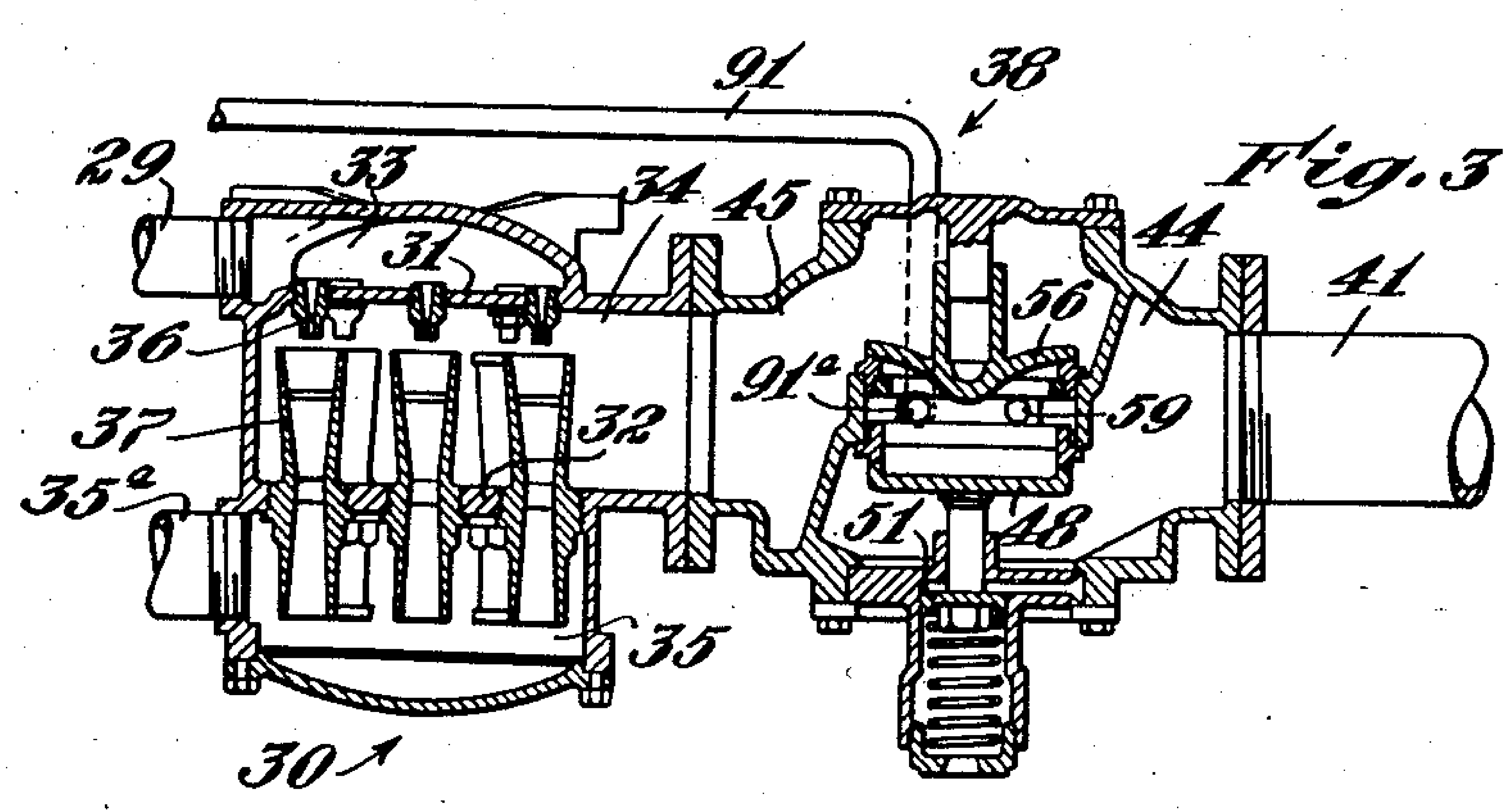


Fig. 3

INVENTOR.
Dartrey Lewis
BY Roberts Cushman & Woodberry
ATTORNEYS.

Feb. 28, 1939. D. LEWIS 2,148,761

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 3

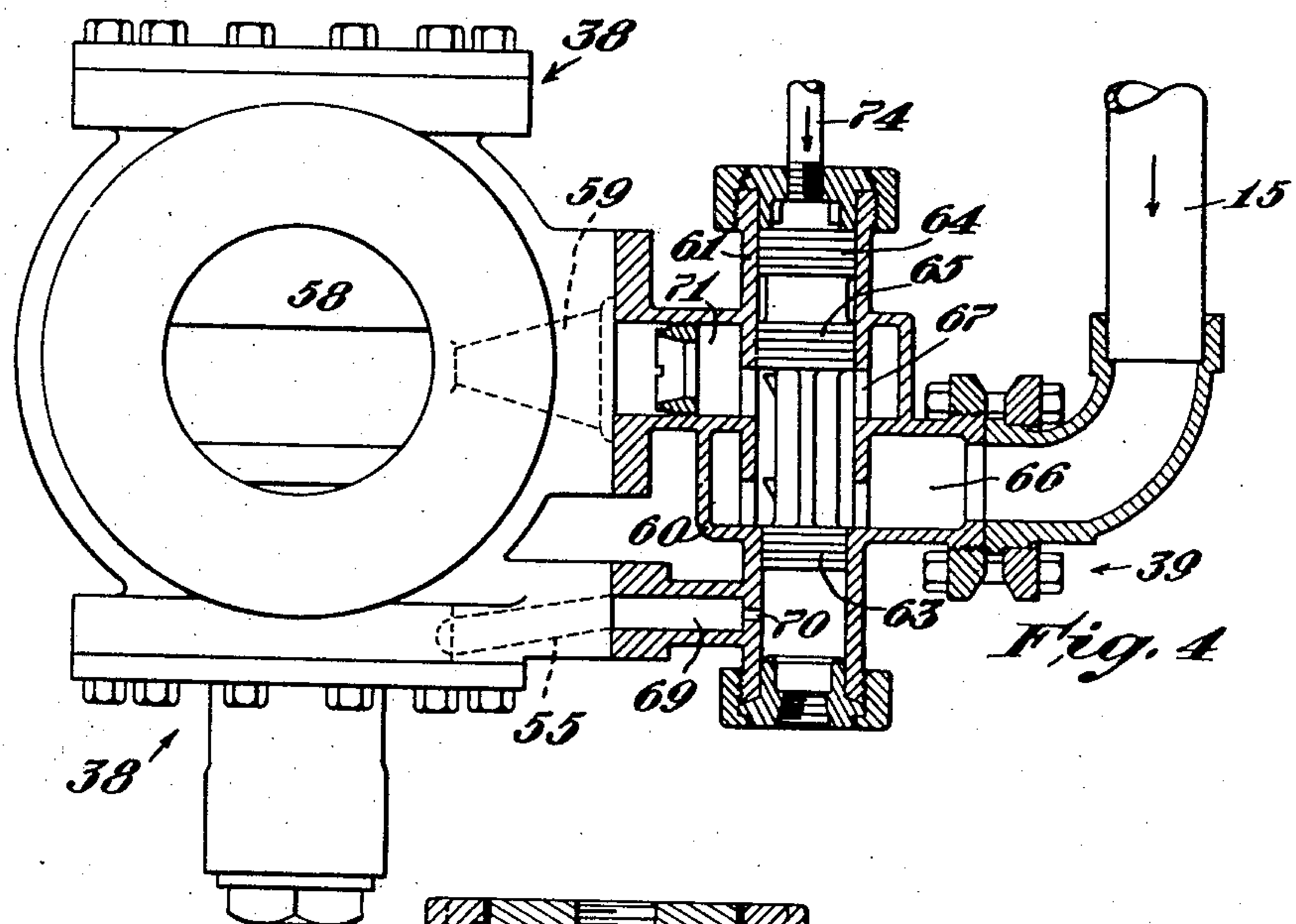


Fig. 4

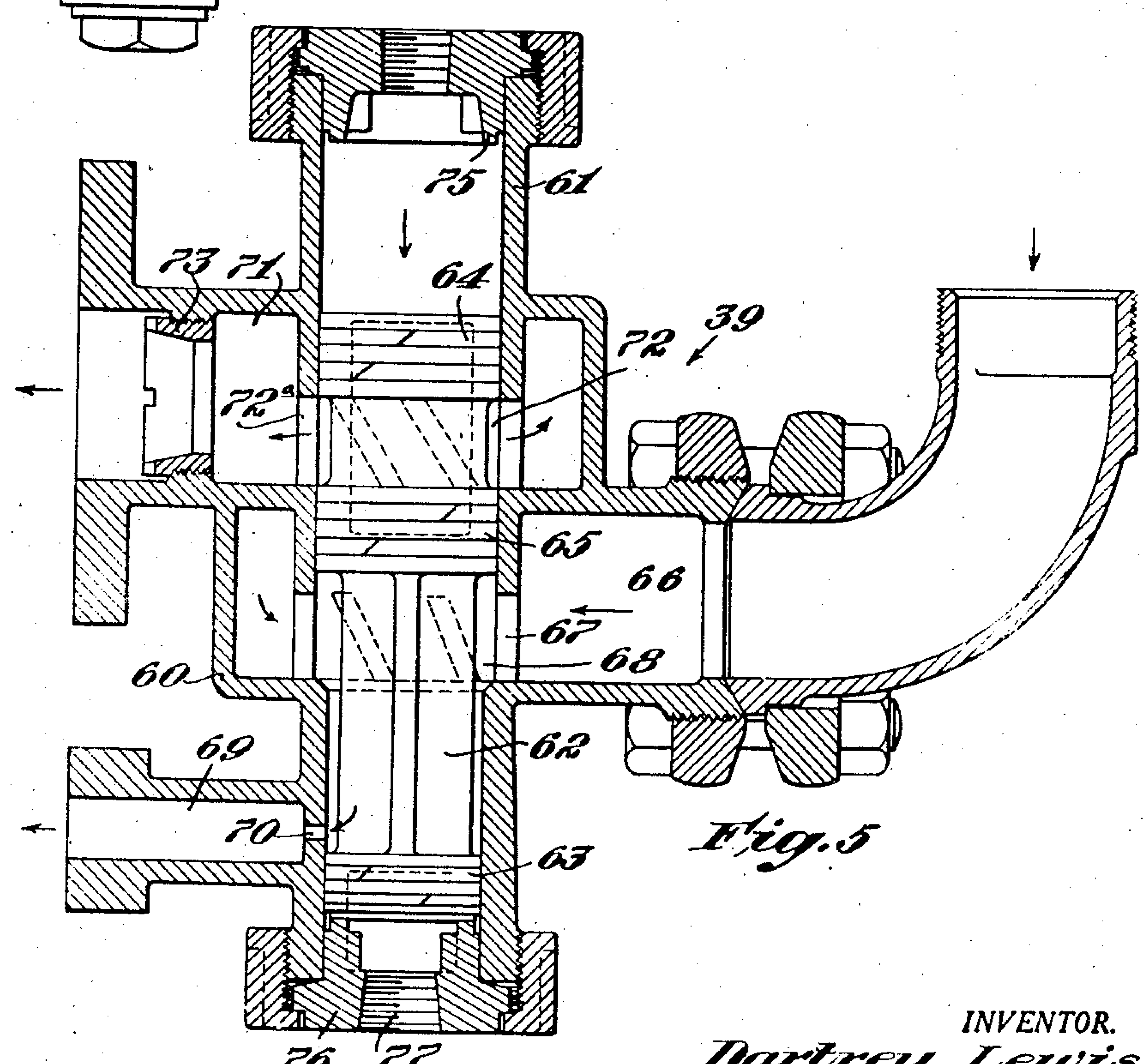


Fig. 5

INVENTOR.
Dartrey Lewis
BY Roberts Cushman & Woodbury
ATTORNEYS.

Feb. 28, 1939. D. LEWIS 2,148,761

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 4

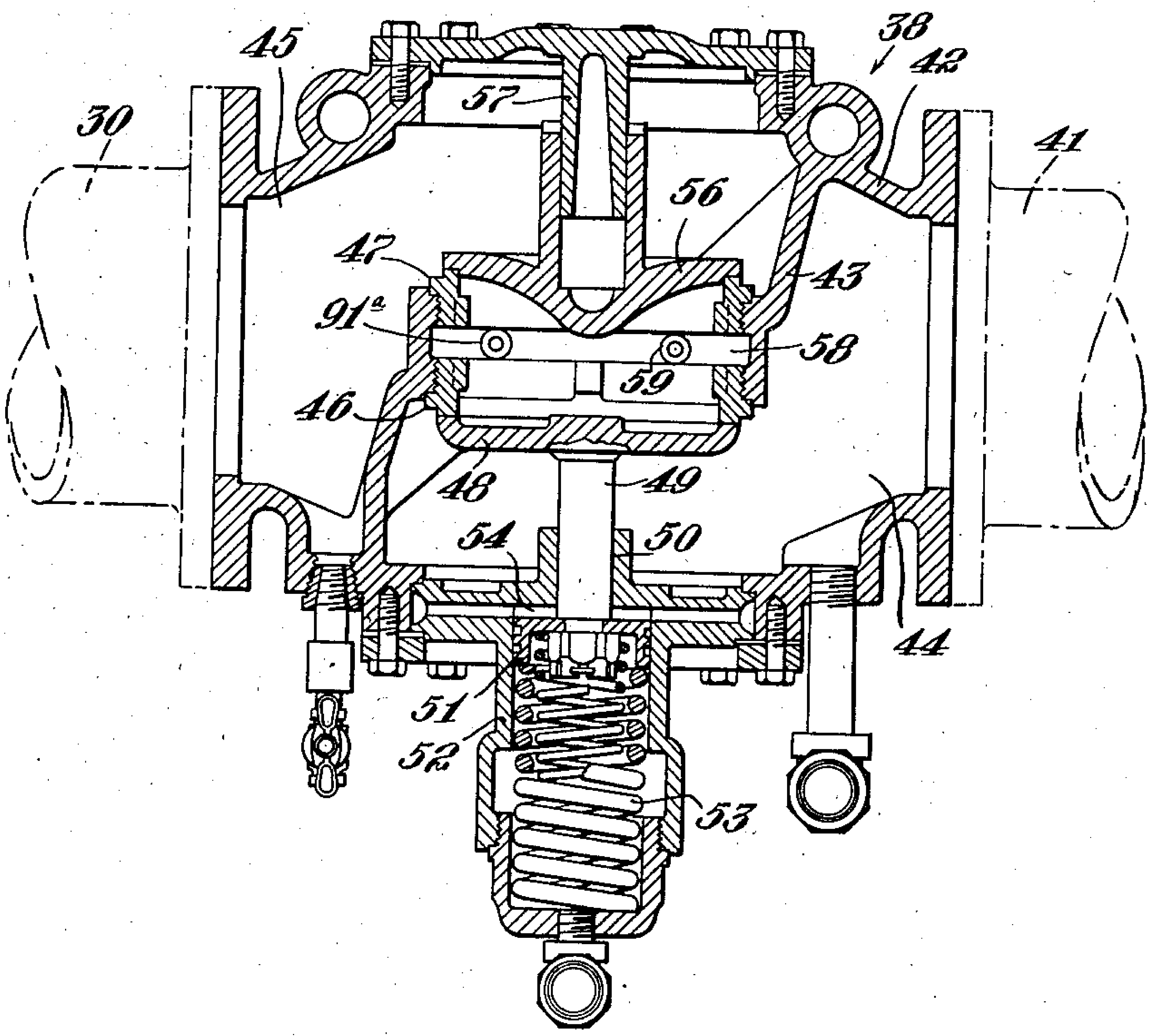


Fig. 6

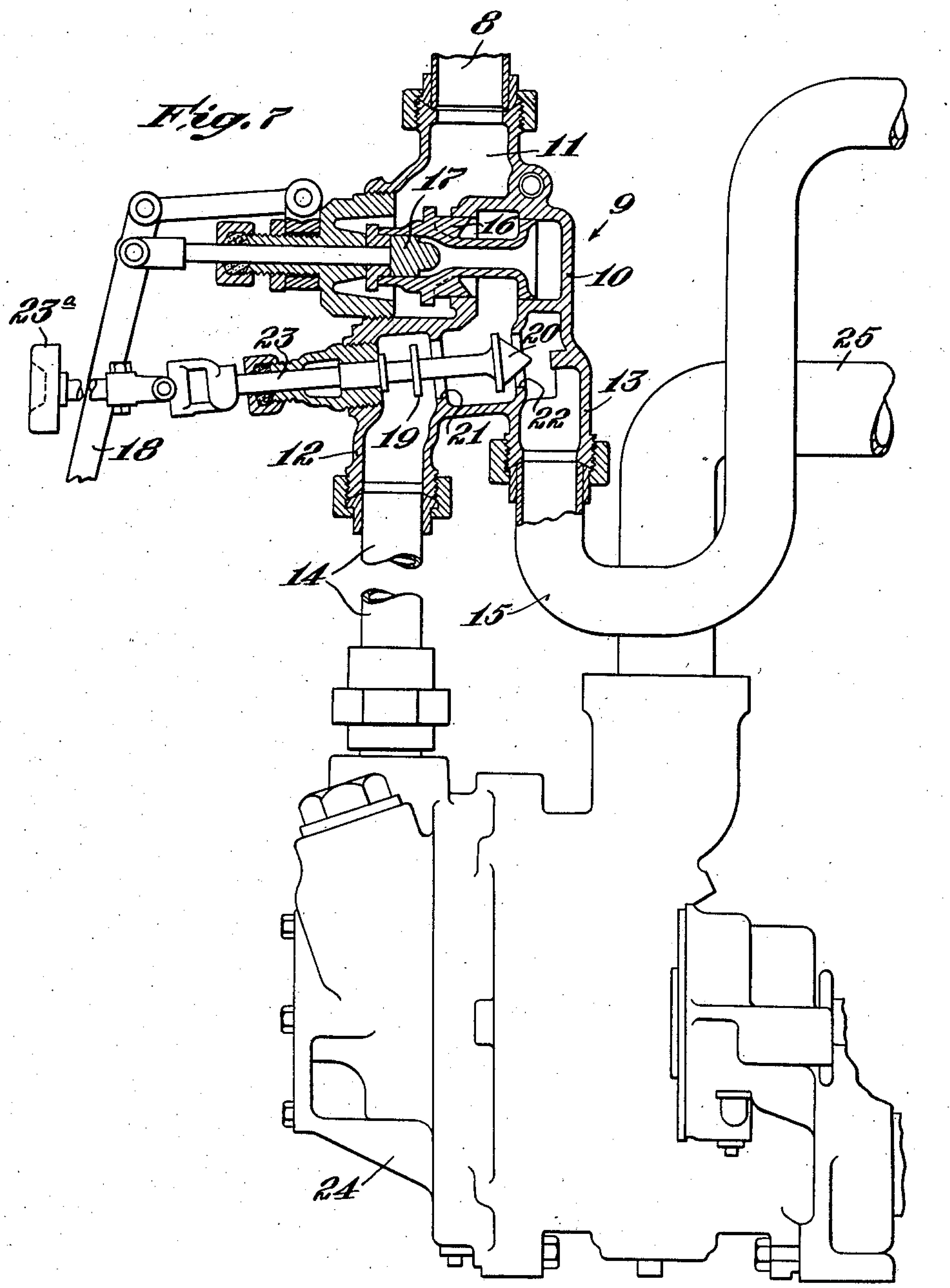
Inventor
Dartrey Lewis
by Roberts Cushman & Woodberry
Att'ys.

Feb. 28, 1939. D. LEWIS 2,148,761

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 5

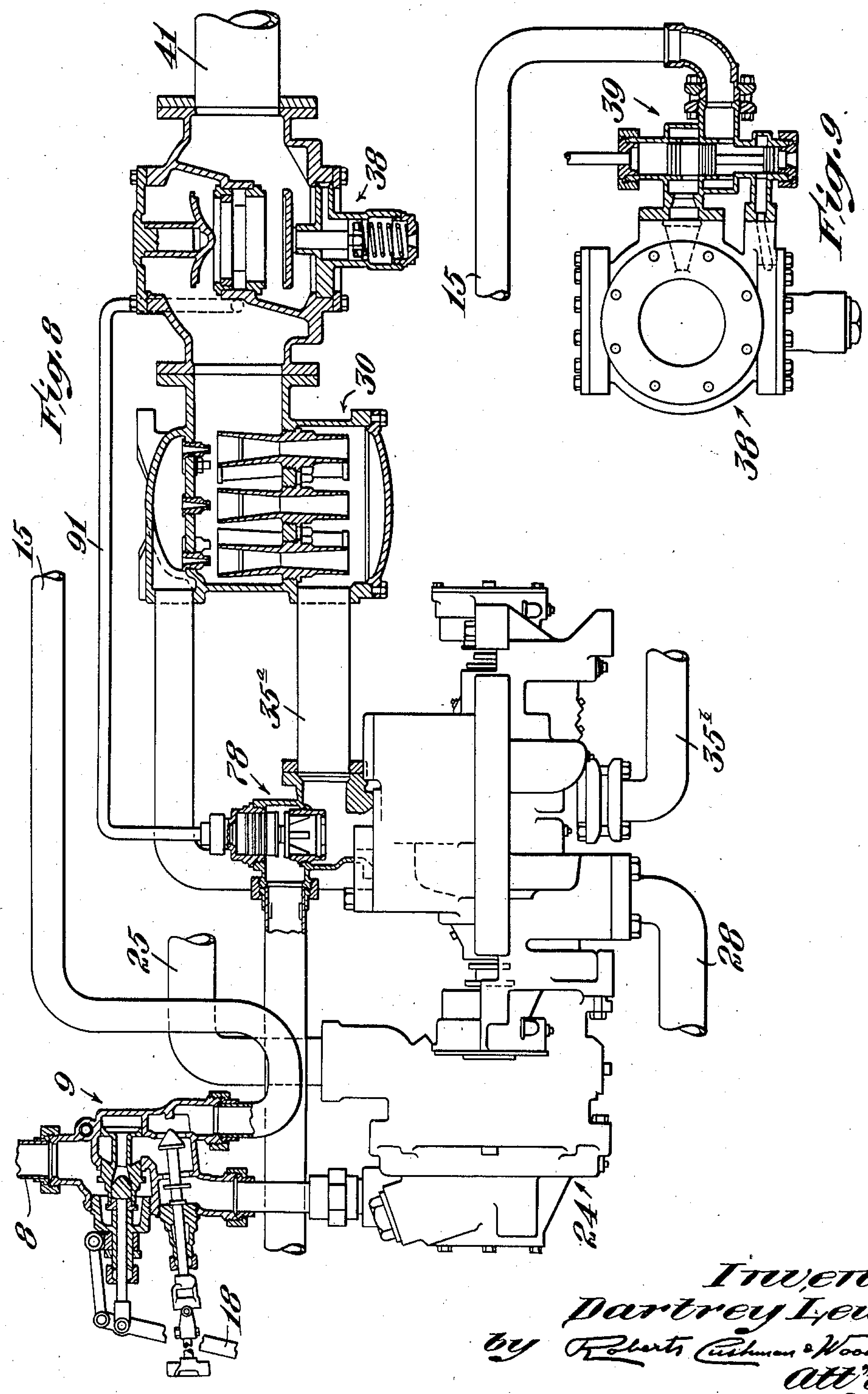
Fig. 7



INVENTOR.
Dartrey Lewis
BY Roberts Cushman & Woodbury
ATTORNEYS.

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 6



Inventor
Dartrey Lewis
by Roberts Cushman & Woodberry
Att'ys.

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 7

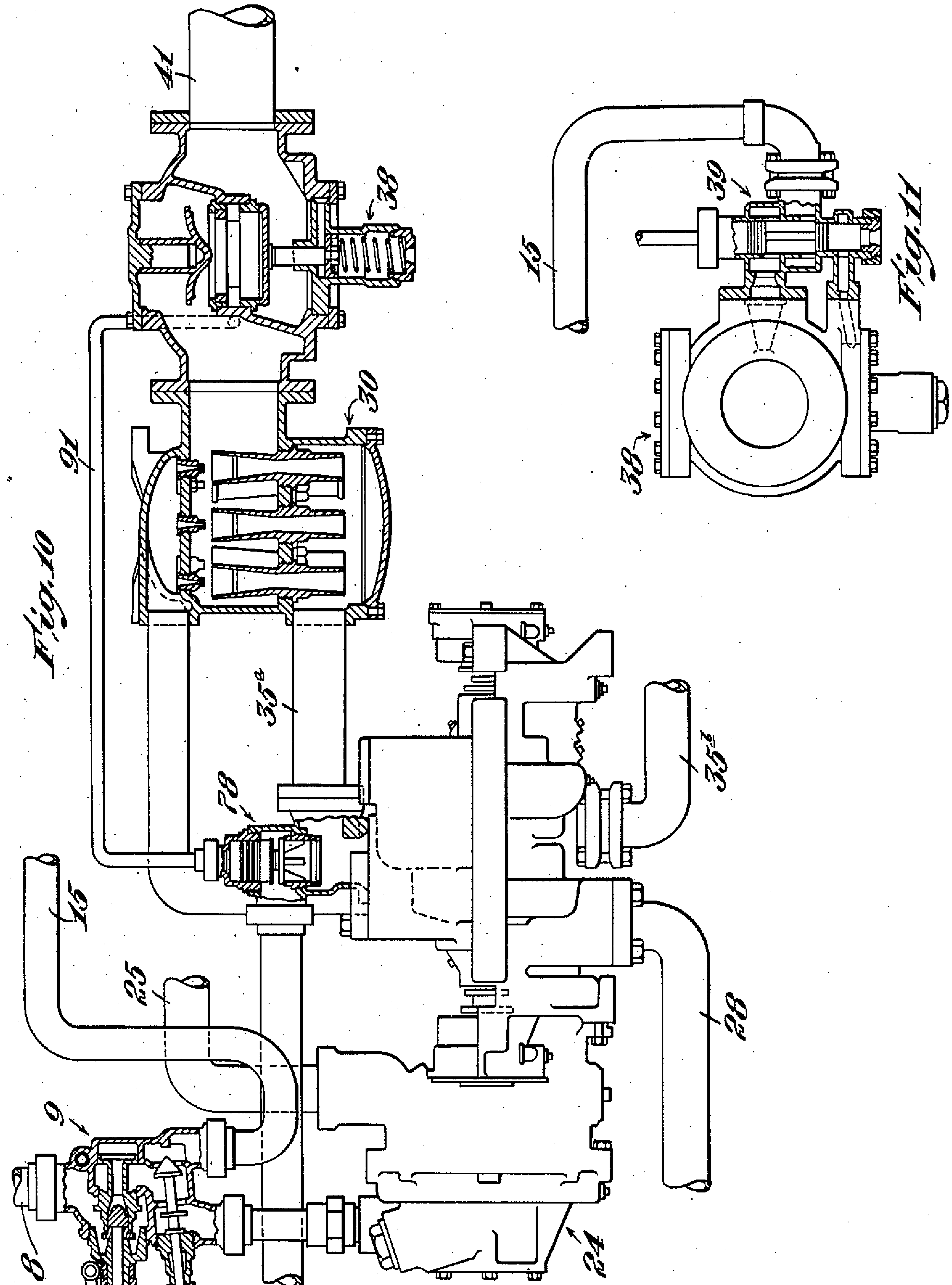


Fig. 10

Fig. 11

Inventor
Dartrey Lewis
by Roberts Cushman & Woodberry
Att'ys.

Feb. 28, 1939. D. LEWIS 2,148,761

BOILER FEEDING APPARATUS

Original Filed May 17, 1937 8 Sheets-Sheet 8

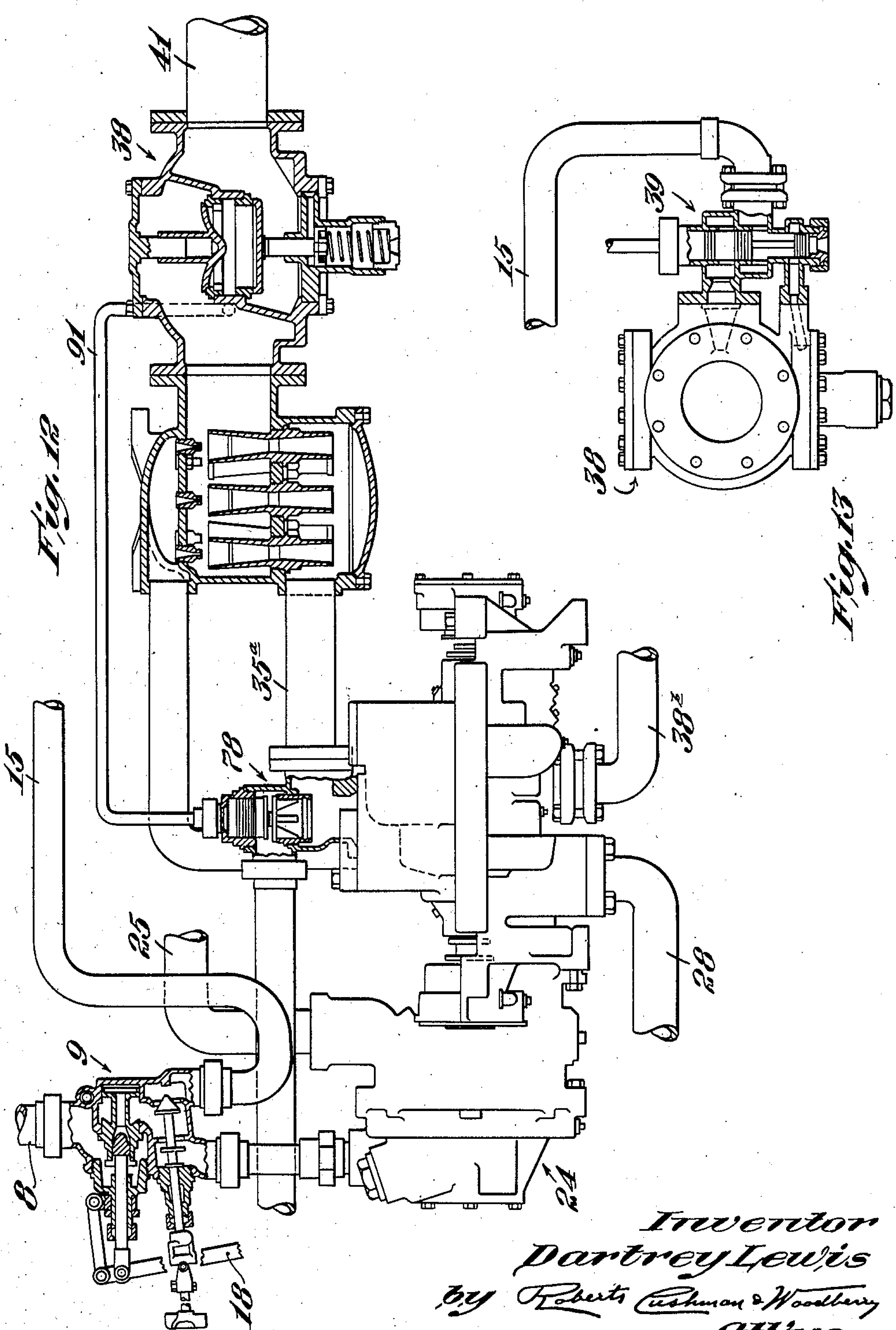


Fig. 12

Fig. 13

Inventor
Dartrey Lewis
by Roberts Cushman & Woodberry
Att'ys.

Patented Feb. 28, 1939

2,148,761

UNITED STATES PATENT OFFICE

2,148,761

BOILER FEEDING APPARATUS

Dartrey Lewis, Boston, Mass., assignor to Manning, Maxwell & Moore, Incorporated, New York, N. Y., a corporation of New Jersey

Original application May 17, 1937, Serial No. 143,131. Divided and this application July 21, 1937, Serial No. 154,813

11 Claims. (Cl. 122—442)

This invention pertains to steam engineering, and relates more especially to improvements in apparatus for delivering feed water to a steam boiler, the present application being a division of the copending application of Dartrey Lewis, Serial No. 143,131, filed May 17, 1937. While in the specific application of the invention, herein chosen for convenience in illustration and description, it is shown as applied to a steam locomotive, it is to be understood that the invention is not in any way necessarily limited to such specific application but is of broad utility wherever it is desired to deliver water to a steam boiler.

While it is customary to deliver feed water to steam boilers by means of jet pumps of the kind known as injectors or inspirators, there are certain situations wherein pumps of mechanical type are required, either as auxiliary to or in substitution for such jet pumps, and in particular where it is requisite to preheat the feed water to a high temperature and to deliver the hot water against a high boiler pressure.

In the patents to Williston et al. No. 1,828,633, dated October 20, 1931, and Allen No. 1,849,900, dated May 15, 1932, desirable forms of apparatus are described, designed economically to preheat feed water to any desired temperature, and to deliver the heated water into the boiler against any desired pressure, the patented apparatus including a multi-stage high-speed turbine-driven centrifugal pump and a water heater of the jet-condenser type having nozzles through which the water passes on its way from the first to the second stage of the pump, and in which it intimately contacts with and condenses exhaust steam from the engine. The heated water is then delivered to the second stage of the pump and by the latter is forced into the boiler.

The apparatus disclosed in the aforesaid patents is highly advantageous as compared with most prior types of feed water pump and heater, being compact and relatively light in weight,—capable of application wherever it is most convenient to place it,—effective to raise the temperature of the feed water through such range as may be desired and to deliver it against the high boiler pressures of present-day practice, and operating with a high thermal efficiency.

The present invention represents an improvement upon the apparatus disclosed in the aforesaid patents and has for its general object the provision of water heating and feeding means which is dependable under all conditions of engine operation; which will automatically adapt itself to changes in operating conditions; which makes it possible to use a jet-condenser low pressure type of heater in combination with a variable speed centrifugal type feed pump without danger of flooding or choking the heater when operating at low capacity; to provide apparatus such that the amount of steam available for heating is always substantially proportional to the speed of the pump; to provide apparatus in which exhaust or live steam may be used alternatively for heating the feed water; to provide means for maintaining a substantially uniform pressure at the inlet at the second stage of the pump under widely varying working conditions; and to provide improved appliances and devices of novel construction and in novel combination whereby the above and other desirable effects are attained.

Other objects and advantages of the invention will be made manifest in the following more detailed description and by reference to the accompanying drawings, wherein

Fig. 1 is a fragmentary, diagrammatic side elevation of a locomotive equipped with feed water apparatus embodying the present invention;

Fig. 2 is a vertical transverse section, to larger scale, illustrating a by-pass valve of a desirable type for use as an element of the water heating and feeding mechanism of the present invention;

Fig. 3 is a fragmentary transverse section, to smaller scale than Fig. 2, showing a control valve and feed water heater of improved type desirable for use in the boiler feed mechanism of the present application;

Fig. 4 is an end elevation, partly in vertical section, illustrating the control valve in association with an automatic heating valve desirable for use as a part of the mechanism of the present invention;

Fig. 5 is a transverse section, to larger scale, showing the heating valve of Fig. 4, but with the parts in a different position;

Fig. 6 is a transverse section showing the control valve of Fig. 3, but to larger scale;

Fig. 7 is a transverse vertical section illustrating a desirable form of operating and regulating valve useful in the mechanism of the present invention;

Fig. 8 is a diagrammatic view, partly in vertical section and partly in elevation, illustrative of the mode of operation of the improved mechanism, and showing the parts as positioned under one set of operating conditions;

Fig. 9 is a fragmentary end elevation, corresponding to Fig. 8; and

Figs. 10 and 12, and 11 and 13, are views similar to Figs. 8 and 9, respectively, but with the various parts differently positioned corresponding to other operating conditions.

Referring to Fig. 1, wherein the invention is shown by way of example as applied to a locomotive, the numeral 1 designates the cab of the locomotive, 2 the tender, 3 the boiler, 4 the turret valve, 5 one of the steam chests, 6 the corresponding cylinder, and 7 the locomotive stack. From the turret valve 4, which receives live steam from the boiler, a pipe 8 leads to the operating valve 9. The operating valve here illustrated is of a type generally resembling the operating valve described in the patent to Walch No. 2,056,698, dated October 6, 1936.

The valve 9 (Fig. 7) comprises a casing 10 having an inlet chamber 11 to which live steam is supplied by the pipe 8. The casing has outlet nipples 12 and 13 to which are connected pipes 14 and 15 leading respectively to the pump actuating turbine and to an automatic water heating valve hereinafter to be described. The flow of steam from the inlet chamber 11 to the outlet nipples 12 and 13 is primarily controlled by a manually actuable balanced valve member having a main head 16 and a relatively movable pilot member 17, said pilot member and the valve head being moved one after the other in succession by the manipulation of a lever 18 having a handle which is disposed within the cab.

Preferably the flow of steam to the respective nipples 12 and 13 is further controlled by a regulating valve comprising spaced heads 19 and 20 which cooperate respectively with annular seats 21 and 22. The heads 19 and 20 are fixed to a stem 23 which is provided with an actuating handle 23^a, also located within the cab. Preferably the regulating valve is so devised, as by the provision of a suitable limiting stop, or by properly dimensioning the valve heads relatively to their seats, that, even when closed as much as possible, sufficient steam will be permitted to pass (assuming that the operating valve 16 is open) to drive the turbine and pump at a rate at which the pump will still deliver a minimum quantity of water to the boiler and also to provide sufficient live steam, if necessary, to heat the feed water.

Assuming that the valve head 16 is unseated, live steam from the boiler will pass through the pipe 14 and enter the casing 24 of the pump operating turbine. An exhaust pipe 25 conveys the exhaust steam from the turbine to some convenient point of discharge; for example, as here shown, to the locomotive stack 7.

The turbine, which may be of any desired type, is direct connected to a multi-stage centrifugal pump, preferably provided with four sets of impeller blades mounted on the same shaft and thus always turning at the same speed, such an arrangement, together with the direct connection of the pump to the turbine, saving space and weight and ensuring proper driving without the use of complicated gearing or other inefficient or uncertain connections. A direct connected turbine and multi-stage pump of this general type is more fully illustrated in the patent to Allen No. 1,849,900 above referred to, but whereas in said patented device the pump comprises but two sets of impellers, it is preferred, in accordance with the present invention, to provide four sets of impellers. However, as in the device of the Allen patent, the first set of impellers constitutes the first distinct stage and, in effect, a separate pump, and this first pump stage delivers feed water to a heater of the jet-condenser type which, in turn, delivers heated water to the first set of impellers of the second stage of the pump. For convenience in further description, the second, third and fourth sets of impellers are regarded as collectively constituting a "second" pump stage, although in actual fact the "second stage" is a three-stage pump in which the pressure of the hot water is successively boosted up to a point such that it may be delivered directly to the boiler against boiler pressure. Since the details of the pump form no essential part of the present invention, and since its general character is clearly disclosed in the Allen patent, no further specific description is here necessary.

The locomotive tender 2 is provided with the usual water supply tank from which the hose connection 26 leads to a strainer 27 from which the cold water supply 28 leads to the intake of the first stage of the pump. The cold water from the first pump stage is delivered through pipe 29 to the feed water heater 30. This feed water heater, which also acts as a condenser for exhaust steam, may be located at any desired and convenient point, but is here shown as arranged near the forward end of the boiler.

The casing of the feed water heater 30 may be of any desired external shape, but is here shown as of drum-like form. The interior of this casing (Fig. 3) is divided by septums 31 and 32 into an inlet chamber 33 into which the water is delivered by the pipe 29, an intermediate chamber 34 which receives the steam for heating the water, and a delivery chamber 35. One or more water delivery nozzles 36, mounted in openings in the septum 31, deliver the water received from the first pump stage, in the form of powerful jets, into corresponding convergent-divergent ejector tubes 37 which are mounted in openings in the septum 32 with their receiving ends in the chamber 34 and their delivery ends in the chamber 35. From the chamber 35 a pipe 35^a leads to the inlet eye of the second stage of the pump, and from the delivery orifice of the second stage of the pump a pipe 35^b conveys the hot feed water to the boiler check valve 40 through which it passes into the boiler.

For maximum heating of the feed water, it has been found that certain definite dimensional relations between the nozzles and ejector tubes are requisite. Thus, experimentally, it has been discovered that by making the diameter of the nozzle throat in a ratio of from one-third to one-fifth of the tube throat diameter and by placing the nozzle so that the distance from its throat to that of the tube is from four to five times the tube throat diameter, it is possible to heat the water to within ten degrees of the theoretical maximum (that is to say, within ten degrees of the temperature of the saturated steam used in heating it) when the pressure at the discharge of the condenser is kept substantially equal to that of the steam used for heating.

The feed water heater 30 is designed to raise the feed water delivered to it by the first pump stage to the desired temperature whether the engine is consuming steam or not (in other words, whether or not exhaust steam is available for heating the water), and to this end the present invention contemplates the provision of automatic means designed to deliver exhaust steam to the heater, so long as such steam is available or, if exhaust steam is not available, then to supply live steam to the heater but to cut off both exhaust and live steam from the heater when the pump is not operating, thereby to avoid any possibility of blowing steam back into the tank. The delivery of exhaust or live steam or the cutting off of both is determined by the action of a control valve 38 (Figs. 3 and 6) and a heater valve 39 (Figs. 4 and 5). These valves 38 and 39 are conveniently located just forward of the heater 30 (Fig. 1), the heater valve 39 receiving live steam from the operating valve 9 through the pipe 15, while valve 38 receives exhaust steam from the exhaust cavities of the valve chest through the pipe 41.

The control valve 38 (Fig. 6) has a casing 42 divided by a septum 43 into an inlet chamber 44 and a discharge chamber 45. Preferably the casing 42 is bolted directly to the casing of the heater 30 so that the discharge chamber 45 is in direct communication with the intermediate chamber 34 of the heater.

The septum 43 has a large opening in which two coaxial valve seat rings 46 and 47 are seated. A valve head 48 cooperates with the seat 46, being mounted on a stem 49 which slides in a fixed guide boss 50 forming a part of the lower head of the casing. The lower end of the stem 49 is furnished with a piston head 51 which slides in a cylinder 52 formed in a downward extension of the lower head of the casing. One or more compression springs 53 tend to raise the piston and thus to hold the valve disk 48 against its seat 46. The space 54 above the piston head 51 communicates by means of a passage 55 (Fig. 4) with one of the chambers of the heater valve 39 (hereinafter more fully described). Through this passage 55 live steam is at times supplied to the space 54, thereby to drive the piston 51 downwardly in opposition to the spring 53 and thus to move the valve disk 48 away from its seat.

A check valve disk 56 cooperates with the seat 47, said disk having a tubular stem which slides on a fixed guide boss 57 projecting downwardly from the upper head of the casing. This check valve 56 tends to seat in response to the action of gravity or by fluid pressure applied to its upper side but lifts in response to pressure below it in excess of the pressure above it. When seated, the valve disks 48 and 56 are spaced apart, thereby providing between them a chamber 58 into which leads a passage 59 which communicates with a chamber of the heating valve 39.

The casing 60 of the heating valve 39 (Figs. 4 and 5) may be integral with the casing 42 of the control valve, or separate from and bolted thereto as may be preferred. The casing 60 has a central portion 61 provided with a cylindrical bore in which slides a differential piston valve comprising a stem having a head 63 (Fig. 5) at its lower end and a duplex head of larger diameter adjacent to its upper end, said duplex head comprising the spaced members 64 and 65. The several heads of this piston may be provided with packing rings is desired. As shown, the lower head 63 is of substantially smaller diameter than the upper head members 64, 65,—the cylindrical bore being smaller at its lower end to cooperate with this head 63.

Any suitable ratio of areas between the large and small ends of this differential piston valve may be employed. For example, if L equals the area of the large head and S equals the area of the small head, and P^1 equals the pressure which acts beneath the large head and P^2 equals the pressure which acts beneath the small head of this differential piston valve, then the forces on this valve are balanced when

$$\frac{P^2}{P^1}=\frac{L-S}{L}$$

The piston valve will move up or down when the ratio of P^1 to P^2 is out of balance sufficiently to overcome the friction of the piston. A suitable ratio of P^2 to P^1 is one to four and one-half.

The casing 60 has an inlet chamber 66 which receives live steam from the operating valve through pipe 15, and the wall of the cylinder 61 has ports at 67 through which steam from the chamber 66 may enter the space 68 below the valve head 65. The casing also has a chamber 69 which at times communicates by means of a small port 70 with the chamber 68 and which is connected by passage 55 (Fig. 4) with the space 54 above the head of piston 51 of the control valve (Fig. 6) as above described.

The casing 60 also has another chamber 71 which, at times, communicates by means of ports 72ᵃ with the space 68 between the piston heads 65 and 64. Preferably, an orifice ring 73 of predetermined capacity restricts the flow of steam from the outlet chamber 71 to a predetermined maximum amount. Steam which passes through the orifice 73 flows through the passage 59 (Figs. 4 and 6) in the casing of control valve 38 and thus enters the space 58 between the control valve disks 48 and 56.

A pipe 74 conducts live steam from the steam chest 5 to the space within the cylinder 61 above the valve head 64, so that the latter is always subjected to high steam pressure, so long as the engine throttle valve is open. The cylinder 61 is provided with stops 75 and 76 at its upper and lower ends, respectively, to limit movement of the differential piston valve. Preferably a drain opening 77 is provided beneath the piston head 63 to prevent an accumulation of pressure fluid beneath such head.

Operation

When the engine is running and exhausting steam in an amount sufficient to heat the feed water (Figs. 8 and 9), and assuming that the operating valve has been opened to admit live steam through pipes 14 and 15 to the turbine and heater valve respectively, and further assuming that the pressure of live steam from the steam chest, acting on the valve head 64, has pushed the differential valve downwardly until its lower head engages the stop 76, as shown in Fig. 5, live steam is admitted through the ports 67 to the chamber 68 and thence through the port 70 to the space 69 from which it flows through the passage 55 into the space 54 above the piston 51 of the control valve, thus pushing said piston downwardly and moving the control valve disk 48 away from its seat. At this time the head 65 of the heater valve piston cuts off communication between the chambers 68 and 71 so that no live steam can enter the latter chamber. However, chamber 44 of the control valve is now supplied with exhaust steam from the exhaust cavities of the steam chest, and this steam passes between the lowered valve disk 48 and its seat and enters the space 58 and lifts the check valve disk 56 from its seat. The exhaust steam then passes through the chamber 45 and into the intermediate chamber 34 of the heater device.

At the same time, live steam admitted through the pipe 14 starts the turbine and thus drives the pump, the speed of the turbine and pump being determined by the amount of steam admitted, which is regulated by the setting of the regulating valve comprising the heads 19 and 20. Upon admission of exhaust steam to the chamber 34 of the heater, its first effect is to tend to clear the chamber of water by forcing it through the pipe 35ª into the second pump stage. By this time the pump has picked up speed sufficient to draw water from the tank and to deliver it at substantially tank temperature and at a pressure of the order of fifty pounds per square inch, for example, to the nozzles 36 of the water heater. From these nozzles the water is delivered in high velocity jets into the convergent combining sections of the ejector nozzles 37. These jets of relatively cold water entrain the exhaust steam by an ejector action, condensing the steam, and thereby very effectively heating the water, some of the heat energy of the steam being converted into pressure as the water and condensate pass out through the divergent delivery ends of the tubes into the chamber 35. From this chamber the water, now heated, for example to a temperature nearly approximating the temperature of the saturated steam admitted to chamber 34, enters the intake eye of the second pump stage. In passing through this second pump stage, the pressure of the hot water is raised sufficiently to force it through the check valve 40 into the boiler.

So long as the engine continues to supply exhaust steam and so long as the exhaust valve remains open to supply steam for actuating the pump, the above operation continues.

If, when the engine is running and delivering exhaust steam, the operating valve be closed so as to stop the pump (Figs. 12 and 13), the supply of steam through both pipes 14 and 15 is simultaneously cut off. Since live steam is now no longer available to act on the piston 54 of the control valve, the spring 53 raises the piston 54 and closes the valve disk 48 against its seat, thus cutting off communication between the condenser and the exhaust cavities of the engine. However, since valve disk 48 is held to its seat by spring pressure, it acts as a relief or safety valve in response to any accidental excess pressure which might develop in chamber 58. It is to be noted that at this time steam from the steam chest is still available to act on the upper head 64 of the heater valve, thus holding the differential piston valve down in the position of Fig. 5. As soon as the valve disk 48 is seated, the check valve 56 returns to its seat and so prevents feed water from flowing into the control valve chamber 58.

It is frequently necessary to feed water to the boiler when the engine is not running or at least is not consuming steam (as, for example, when a locomotive is drifting) but, as above pointed out, it is highly undesirable to deliver cold water to a hot boiler, although under the conditions just referred to, no exhaust steam is available for heating the water. However, in accordance with the present invention, and by the automatic operation of the appliances above described, the failure of the exhaust steam supply, while the operating valve is open, immediately results in the delivery of live steam to the water heater in sufficient quantity to heat the feed water. Thus, let it be assumed, as illustrated in Figs. 10 and 11, that the operating valve is open so that live steam is being delivered to the turbine, and is also free to pass through the pipe 15 to the chambers 66. However, at this time live steam is no longer supplied by the pipe 74 to act on the upper head 64 of the differential valve as the throttle is now closed. Thus the steam pressure in the space 68 reacts against the lower side of the head 65 of the unbalanced differential valve and raises this valve to the position shown in Fig. 4. In this position the lower head 63 closes the port 70, while at the same time the space 68 is put into communication with the chamber 71 by means of port 72ª.

Live steam now flows through the pipe 15, chamber 68, and the ports 72ª into the chamber 71, and through passage 59 into the space between the valve disks 48 and 56 of the control valve. The valve 48 is now closed by the spring 53 (since steam is cut off from the chamber 54 by the valve head 63), but the check valve 56 is lifted by the live steam in the chamber 58 which is now free to flow into the chamber 34 of the heater where it is entrained by the water jets and heats the water in the same way as the exhaust steam as above described.

By suitably designing the heads 19 and 20 of the regulating valve, it is possible to provide suitable amounts of live steam for heating the water in accordance with the amount of water being pumped. For instance, with the pump operating at 100% capacity, sufficient live steam may be provided to raise the water temperature through 120°; at 50% capacity, sufficient steam may be provided to raise the water through 80°; and at 25% capacity, sufficient steam may be admitted to raise the water through 60°, etc. Obviously other proportions may be provided for by suitably relating the sizes of the valve heads 19 and 20 and the orifices with which they cooperate. As already noted, it is preferred to make the valve heads 19 and 20 of such size relatively to the passages through the valve seats as to ensure turbine driving steam sufficient to create a delivery pressure such as to force some water into the boiler so long as the operating valve is open.

As soon as the throttle is opened to admit steam to the steam chest and cylinder to start the engine, pressure is applied to the upper end of the differential valve 64, thus forcing the latter valve down and reopening port 70 while cutting off the passage of live steam to the chamber 71, and at the same time readmitting exhaust steam to the chamber 41 of the control valve.

While the apparatus as above described is operative for the intended purpose and without adjunctive features, it is preferred to provide an automatic pressure equalizer for the delivery chamber 35 of the heater in order to ensure optimum conditions of operation at all capacities of the pump.

If the pump be considered as consisting of two separate pumps, one of which delivers to the nozzles of the heater and the second of which delivers to the boiler, it will be clear that the second pump delivers against a substantially constant head; that is to say, the boiler pressure, but receives its supply from a source of pressure which may vary in accordance with the speed of the first pump. On the other hand, the first pump receives its supply at a substantially constant head but delivers into a heater in which the pressure may vary substantially in accordance with the speed of the second pump.

Moreover, since the pump is of the centrifugal type, there is a definite minimum speed at which the second stage will deliver any water at all to the boiler, since at any lesser speed the water is merely churned by the impeller blades and remains within the pump casing. Preferably the pump is so designed that at a selected speed, for example, maximum pumping capacity, the first stage delivers such an amount of water to the heater nozzles as will (when combined with the condensed heating steam) substantially equal the amount withdrawn from the heater by the second stage of the pump for delivery to the boiler. However, since the impellers of both the first and second stages of the pump are mounted on the same shaft and necessarily turn at the same speed, any reduction in the speed of the second stage, for cutting down the supply to the boiler, results in a similar reduction in speed of the first stage.

To obtain a clear conception of what happens when the speed of the pump is reduced, it is convenient to consider that extreme condition which exists when the speed is reduced just to the point at which the second stage of the pump will no longer deliver any water to the boiler. Manifestly, when this condition obtains, the first pump stage continues to discharge water into the heater, although at a lesser rate, and since no water is now withdrawn from the heater by the second pump stage, pressure rapidly builds up in the delivery chamber 35 of the heater. As one result of this condition, the entire heater and the passages leading to it soon fill with water so that no further condensation can take place.

As a matter of fact, the amount of pressure against which a jet-condensing heater of the type above described will operate is very strictly limited. Moreover, within the range of pressures within which such a jet-condensing heater will actually operate without choking, the amount of steam condensed rapidly grows less as the back pressure in the delivery chamber in the heater increases, and although in theory the building up of a high back pressure at the inlet eye of the second pump stage (at low speeds) will tend eventually to equalize the output of the first and second stages of the pump, the choking of the condenser constitutes the real limit which determines the minimum practical speed of operation.

On the other hand, while excessive back pressure at the intake eye of the second pump stage is not permissible, insufficient back pressure at this point is also undesirable since at the high temperature of the feed water, low pressure results in foaming and cavitation, with loss of pump capacity and efficiency. Preferably the back pressure at the inlet of the second pump stage should be in excess of the boiling pressure corresponding to the water temperature, and when using a jet-condenser heater designed as above described, this temperature is approximately within ten degrees of the temperature of saturated steam at the pressure supplied to the heater for heating the water.

Accordingly, it is highly desirable to provide automatic means for maintaining a predetermined pressure in the delivery chamber 35 of the heater. To this end, the present invention contemplates the provision of an automatic by-pass valve operating to relieve the pressure in the delivery chamber 35 whenever, during the operation of the pump, it tends to rise excessively.

One desirable form of by-pass valve is shown at 78 in Figs. 1 and 2. This valve is conveniently located adjacent to the pump and is here illustrated as mounted directly upon the pump casing, although this is not necessary. This by-pass valve comprises a casing 79, the interior of which is divided by a septum 80 into an inlet chamber 81 and an outlet chamber 82. The inlet chamber 81 is always in communication by means of pipe 35^a with the delivery chamber 35 of the heater, preferably communicating with pipe 35^a just where the latter enters the inlet of the second stage of the pump. The outlet chamber 82 is connected by means of a pipe 83 and a hose connection 84 to the water tank in the tender.

The septum 80 of valve casing 79 has an opening for the reception of a cylindrical guide 85 whose upper edge constitutes an annular valve seat with which cooperates a by-pass check valve disk 86. This check valve disk has guide wings which slide in the guide 85, and the valve disk also preferably has an upstanding central boss 87 for engagement by the lower end of a loading piston 88 which slides in a bore in a hollow plug 89 forming the top of the casing 79. Above the piston 88 is a space 90 which communicates by means of a pipe 91 with the chamber 58 between the valve disks 48 and 56 of the control valve 38, the pipe entering said chamber at 91^a. Normally, the valve disk 86 is held to its seat by the weight of piston 88, assisted by the fluid pressure in the space 90, but in response to excess pressure at the intake of the second pump stage, the valve 86 rises and allows water to escape from the chamber 35 through the pipe 35^a and thence through the chambers 81 and 82 and the pipe 83 to the tank in the tender.

By drawing the steam which applies pressure to the piston 88 from the chamber 58 of the control valve, it is assured that the pressure in the chamber 90 will never exceed that of the steam supplied for heating. At times, due to improper operation of the heater, the pressure in the intermediate chamber 34 of the heater may be higher than that of the heating steam, but the check valve disk 56, which is interposed between the chamber 34 and the inlet 91^a to the pipe 91, effectively prevents any higher pressure than that of the heating steam from acting on the piston 89. Normally the heating steam has a pressure which may vary from zero to twenty-five pounds per square inch, it being noted that even when live steam is being used for water heating, such steam is so throttled in passing through the various pipes and valves and through the orifice 73 that when it reaches the chamber 58, its pressure is not substantially higher than that of the exhaust steam which is used under other conditions.

If the piston 88 and the valve 86 be of the same effective diameter, the check valve 86 would, in theory, open as soon as the pressure in chamber 81 even slightly exceeds the pressure of the heating steam. Since there is some drop in pressure between the chamber 35 of the heater and the chamber 81 of the by-pass valve, it may be desirable, in order to maintain the pressure in the heating chamber 35 equal to that of the heating steam used, to provide a spring 92 to react with a predetermined upward pressure on the valve 86 so as to compensate for the pressure drop between the chambers 35 and 81. This spring may, for example, be so arranged as to exert a pressure corresponding to a pressure of from one to five pounds per square inch acting over the effective area of the valve 86.

While the check valve disk 56 of the control valve device forms a convenient check to prevent excessive pressure from entering the chamber 90, any other check valve appropriately arranged may be employed in so far as maintenance of uniform pressure in the by-pass valve chamber 90 is concerned.

While, as shown, the spring 92 is a compression spring disposed beneath the valve disk 86, it may be, as well, a tension spring arranged to act upwardly on the piston 88. In fact, this latter arrangement has certain advantages; for example, it leaves the valve 86 always free to seat in response to any tendency whatever of fluid to flow in reverse direction from the chamber 82 to the chamber 81.

Preferably a pressure gauge 94 and a thermometer device 95 are disposed within the cab and connected to the delivery pipe 35^b of the heater so as to inform the engineer, at all times, of the pressure and temperature conditions of the feed water when the pump is in operation.

While a certain desirable embodiment of the invention has herein been shown and described by way of example, and while certain specific forms of valve have been illustrated as of utility, it is to be understood that the invention is not necessarily limited to the precise arrangements herein illustrated nor to the specific valve constructions indicated as desirable for performing certain particular functions, but it is to be regarded as of broad scope and as inclusive of any and all equivalents, both as respects the apparatus as a whole and the individual units which in combination coact to bring about the desired result.

I claim:

1. Apparatus of the class described including a multi-stage, variable-speed centrifugal feed pump, a single steam turbine for driving the pump, and a water heater of the jet-condenser type having nozzles through which water passes on its way from the first to the second stage of the pump, characterized in having pressure relief means operative to prevent flooding of the water heater, said pressure relief means being located between the heater and the second pump stage and constructed and arranged automatically to reduce the pressure at the inlet of the second pump stage when the pressure at said inlet exceeds a predetermined value and regardless of the speed of the pump.

2. Apparatus of the class described including a multi-stage, variable-speed centrifugal feed pump, a single steam turbine for driving the pump, a water heater of the jet-condenser type having nozzles through which water passes on its way from the first to the second stage of the pump, and a conduit for conveying the water from the heater to the inlet of the second pump stage, and a regulable steam admission valve for delivering steam to the turbine whereby the speed of the latter may be varied, characterized in having a normally closed relief valve constructed and arranged automatically to open, when the pressure in the conduit leading from the heater to the inlet of the second pump stage reaches a predetermined value, and discharge water from said conduit thereby to prevent flooding the heater regardless of the speed of the pump.

3. In a boiler feed apparatus designed to deliver preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a variable-speed, multi-stage centrifugal pump, a single steam turbine for driving the pump, a heater of the jet-condenser type to which the water from the first stage of the pump is delivered and from which the water after heating is conveyed to the second stage of the pump and by the latter is forced into the boiler, in combination, pressure relief means comprising a normally closed valve so constructed and arranged that it tends automatically to open in response to the pressure of the hot water on its way from the heater to the second pump stage and loading means normally holding the valve closed, said loading means being so constructed and arranged that when the pressure of the hot water on its way to the second pump stage reaches a predetermined value it permits the valve to open and allow hot water to escape thereby to prevent excessive pressure at the outlet of the heater and consequent flooding of the latter.

4. Boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage pump, a heater of the jet-condenser type comprising a water inlet chamber, a water delivery chamber and water nozzles arranged to discharge water from the inlet chamber into convergent-divergent ejector tubes discharging into the water delivery chamber, the water from the first stage of the pump being delivered to the heater and, after heating, being conveyed to the second stage of the pump and by the latter being forced into the boiler, a conduit for conveying heated water from the heater to the inlet of the second pump stage, and means for supplying heating steam to the heater, in combination, relief valve means operative, when the pressure in the delivery chamber of the heater reaches a predetermined value, to provide for escape of water from the conduit which leads from the heater to the second pump stage, thereby to lower the pressure in the delivery chamber of the heater to prevent flooding of the latter, and fluid pressure means for loading said relief valve, said loading means being exposed to the pressure of the heating steam.

5. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage centrifugal pump, the first stage of which receives water from a supply tank, a heater of the jet condenser type to which the water from the first stage of the pump is delivered and from which the water, after heating, is conveyed to the second stage of the pump and by the latter is forced into the boiler, in combination, means providing a by-pass passage leading from the inlet of the second pump stage to the supply tank, a relief valve comprising a valve disk for closing said by-pass passage, and a fluid pressure loaded piston normally holding the relief valve closed, the load imposed by the piston being such that the relief valve opens in response to such a pressure at the inlet of the second pump stage as to prevent flooding of the heater.

6. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage, centrifugal pump arranged to receive feed water from a

source of supply, a heater of the jet-condenser type comprising an inlet chamber, a delivery chamber and water nozzles arranged to discharge water from the inlet chamber into convergent-divergent ejector tubes discharging into the delivery chamber, the water from the first stage of the pump being delivered to the heater and the heated water being conveyed to the second stage of the pump and by the latter being forced into the boiler, and a conduit for carrying heated water from the heater to the second pump stage, in combination, means for supplying steam for heating the water in said heater, means providing a by-pass passage for conducting heated water from the inlet of the second pump stage back to the source of supply, a relief valve normally closing said by-pass passage, the relief valve being operative, in response to pressure in excess of a predetermined value in the delivery chamber of the heater, to open the by-pass passage and allow hot water from the heater to return to the water supply source, and a pressure loaded element exposed to the pressure of the heating steam normally holding said relief valve closed.

7. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam heat of an engine for driving the latter, said feed apparatus being of the kind which includes a turbine driven, multistage centrifugal pump, a heater of the jet-condenser type to which the water from the first stage of the pump is delivered and from which the water, after heating, is conveyed to the second stage of the pump and by the latter is forced into the boiler, in combination, a source of steam for heating the water in said heater, relief valve means operative in response to pressure, in excess of a predetermined value, at the inlet of the second pump stage, to open and relieve said pressure, a fluid pressure-responsive device for loading said relief valve means, a conduit for conducting steam from the source of heating steam to said fluid pressure-responsive device, and means operative to prevent flow of pressure fluid from the heater to said conduit.

8. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage, centrifugal pump, a heater of the jet-condenser type to which the water from the first stage of the pump is delivered and from which the water, after heating, is conveyed to the second stage of the pump and by the latter is forced into the boiler, in combination, a relief valve comprising a valve disk and a seat with which it cooperates, said seat defining an orifice leading from the inlet of the second pump stage, a fluid pressure-responsive device for loading said relief valve disk so as normally to keep it seated, said disk lifting from its seat in response to pressure in excess of a predetermined value at the inlet of the second pump stage, thereby to relieve such pressure at the inlet of the second pump stage, means defining a chamber to which heating steam is delivered so long as the pump is running and from which steam flows to the heater for heating the water therein, a conduit leading from said chamber to the fluid pressure-responsive device whereby the latter is exposed to the pressure of the heating steam thereby to determine the times at which the relief valve may open, and means operative to prevent flow of fluid from the heater to said chamber.

9. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage, centrifugal pump which receives water from a supply tank, a heater of the jet-condenser type to which the water from the first stage of the pump is delivered and from which the water, after heating is delivered to the second stage of the pump and by the latter is forced into the boiler, in combination, a conduit for conveying water from the inlet of the second pump stage to the supply tank when the pressure at said intake exceeds a predetermined amount, a relief valve normally closing said conduit but which is arranged to open when the pressure at the inlet of the second pump stage exceeds a predetermined value, a fluid actuated piston for loading said relief valve, means defining a chamber into which heating steam is delivered so long as the pump is in operation and from which steam flows to the heater, a pipe for conducting steam from said chamber to act on the piston thereby to maintain a load on the relief valve corresponding to the pressure in said chamber, and a check valve interposed between the heater and said chamber and constructed and arranged to prevent flow of pressure fluid from the heater to said chamber.

10. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a variable speed, turbine-driven, multi-stage, centrifugal pump, a heater of the jet-condenser type comprising an inlet chamber, a delivery chamber and water nozzles arranged to discharge water from the inlet chamber into convergent-divergent ejector tubes delivering into the delivery chamber, the water from the first stage of the pump being delivered to the inlet chamber of the heater and after heating being conveyed from the outlet chamber of the heater to the second stage of the pump and by the latter being forced into the boiler, and a conduit for conveying the heated water from the heater to the second pump stage, in combination, means for delivering live steam to the turbine for operating the pump, means for regulating the amount of steam so delivered thereby to change the speed of the pump, means operative to deliver steam to the heater, and control means operative to maintain a predetermined pressure at the outlet of the heater definitely proportioned to the pressure of the steam delivered to the heater regardless of the speed of the pump, said control means including a movable pressure-responsive part exposed to the pressure fluid in the conduit which leads from the heating means to the second pump stage.

11. In a boiler feed apparatus for delivering preheated feed water to a steam boiler which supplies steam to the steam chest of an engine for driving the latter, said feed apparatus being of the kind which includes a multi-stage pump, a heater of the jet-condenser type comprising an inlet chamber, a delivery chamber, and water nozzles arranged to discharge water from the inlet chamber into convergent-divergent ejector tubes delivering into the delivery chamber, the water from the first stage of the pump being delivered to the inlet chamber of the heater and, after heating, being conveyed from the outlet

chamber of the heater to the second stage of the pump and by the latter being forced into the boiler, and a conduit for conveying the heated water from the heater to the second pump stage, in combination, means operative to deliver steam to the heater, and automatic control means including a relief valve operative to maintain a predetermined pressure at the outlet of the heater definitely proportioned to the pressure of the steam delivered to the heater, said control means including a movable pressure-responsive valve-controller exposed to the pressure fluid in the conduit which leads from the heating means to the second pump stage, said valve-controller being so designed and arranged that the relief valve opens and allows hot water to escape from the conduit whenever the pressure in the delivery chamber of the heater exceeds a fixed and definite value.

DARTREY LEWIS.