

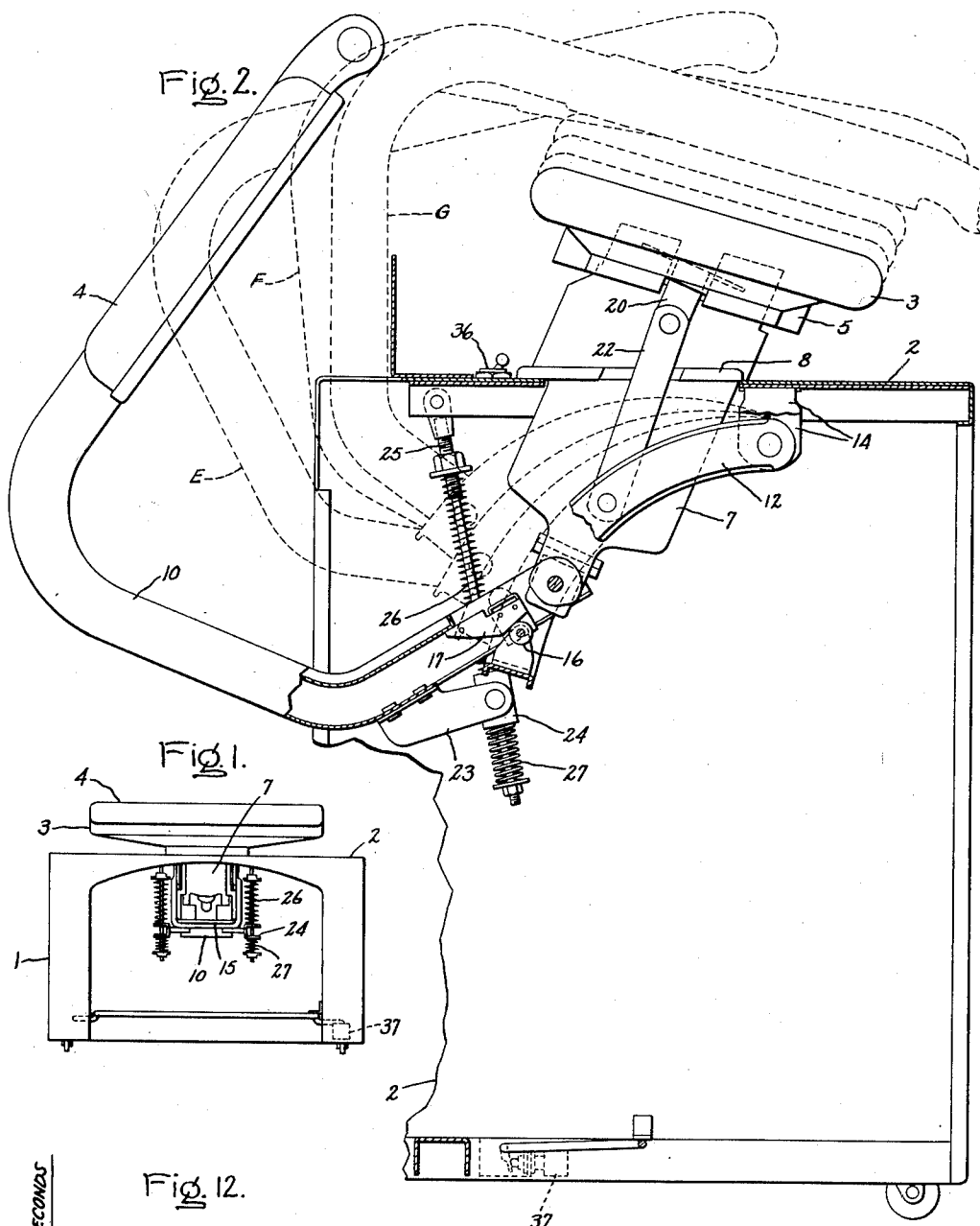
April 1, 1952

C. S. DAYTON
HYDRAULIC PRESS OPERATOR WITH
AUTOMATIC SPEED REGULATION

2,591,410

Filed Sept. 28, 1949

3 Sheets-Sheet 1



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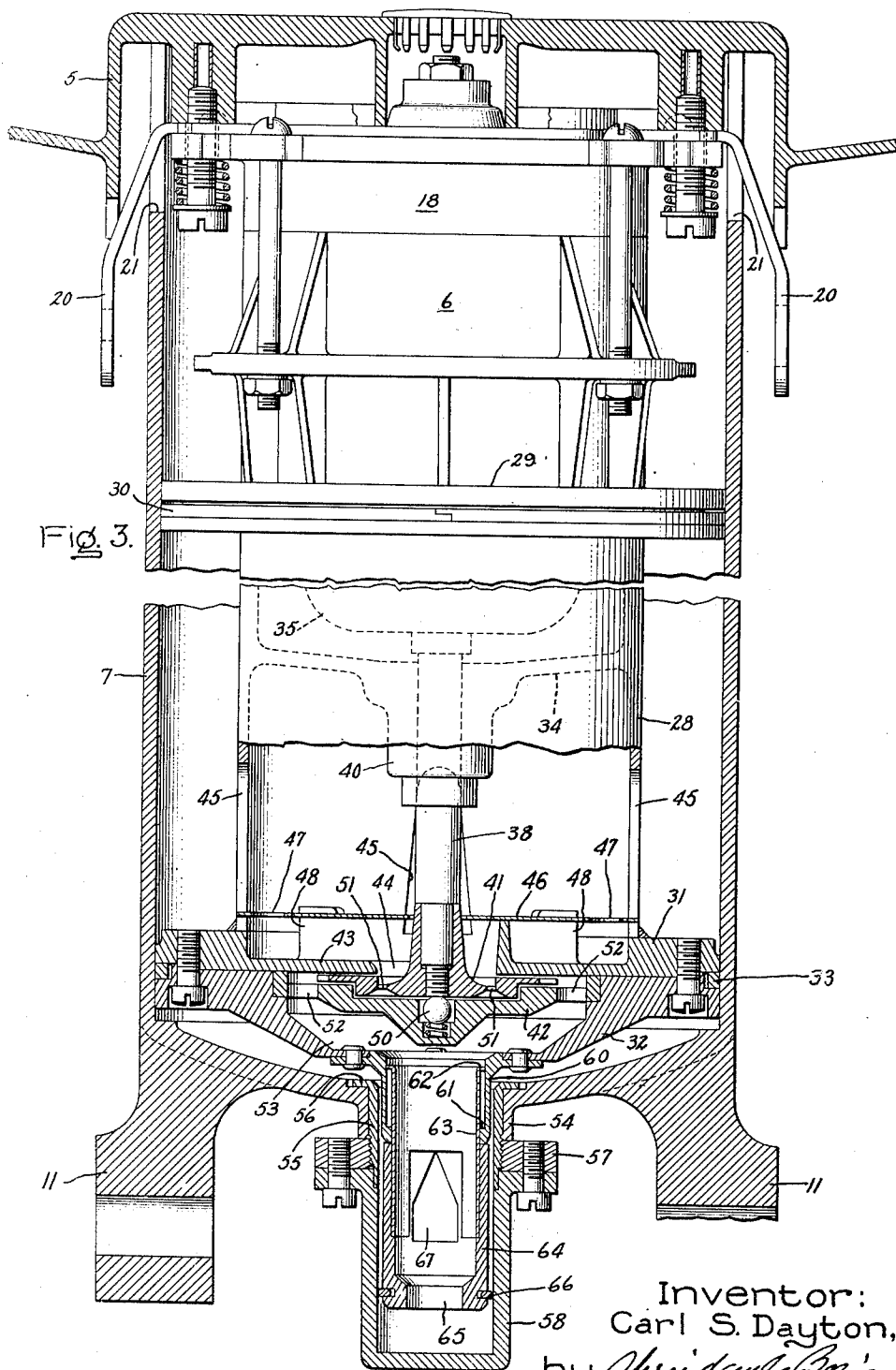
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3 Sheets-Sheet 2



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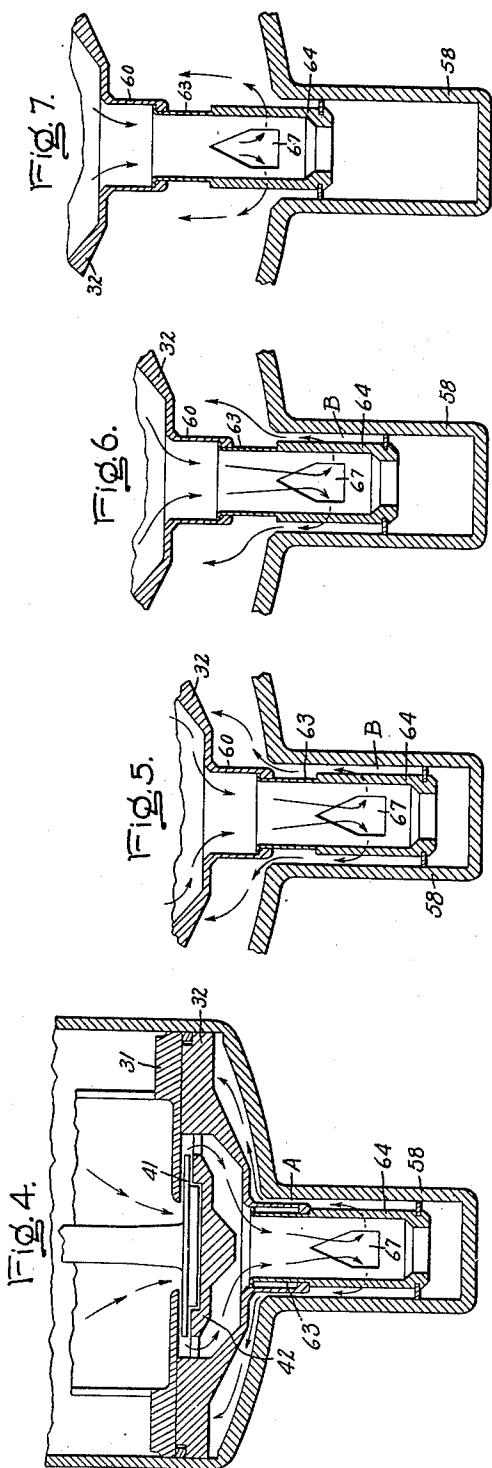
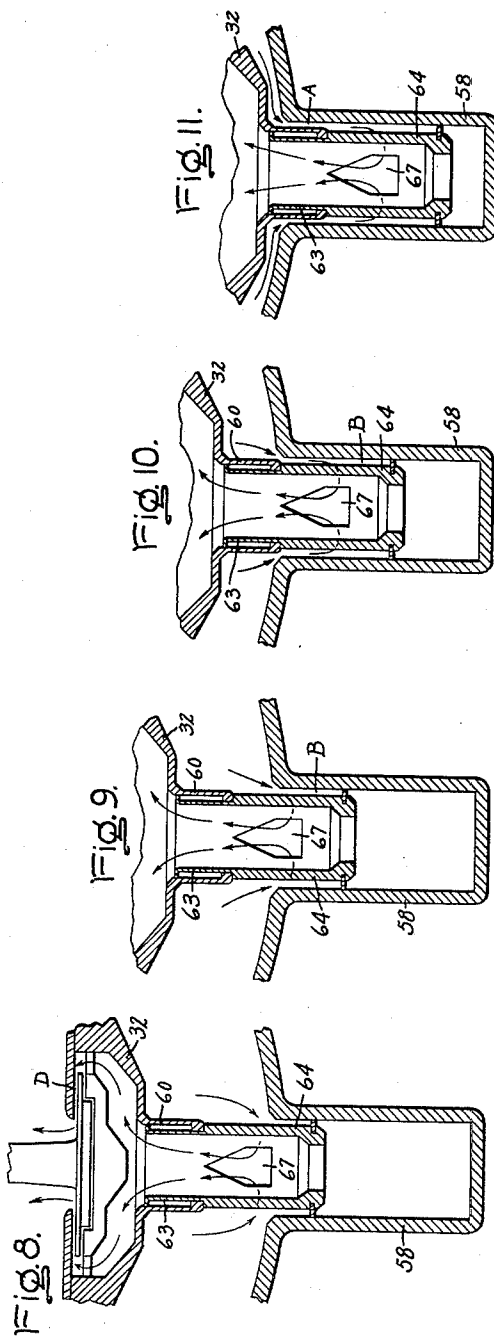
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3 Sheets-Sheet 3



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UNITED STATES PATENT OFFICE

2,591,410

HYDRAULIC PRESS OPERATOR WITH
AUTOMATIC SPEED REGULATIONCarl S. Dayton, Bridgeport, Conn., assignor to
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Application September 28, 1949, Serial No. 118,348

11 Claims. (Cl. 60—52)

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This invention relates to hydraulic power systems, and in particular to an improved hydraulic drive mechanism having an automatic valve assembly which establishes a desired speed pattern of operation for the work devices driven by the power system.

I have chosen to illustrate my invention in association with an ironing press having press elements to be brought synchronously into co-operating relationship. A domestic ironing machine which is presently commercially well known includes a padded buck on which the clothes are spread, and a heated shoe arranged to be brought manually from a home position to an operating position above the buck. As the shoe approaches its operating position a circuit is closed to operate a thrust unit constituting a combined piston and fluid pressure generator within a cylinder affixed to the table of the ironing machine. The piston carries the buck frame, and the drive of the piston brings the buck and the clothes thereon into pressing engagement with the ironing shoe. It has been proposed to operate the shoe automatically by driving it, also, from the piston. My invention is particularly well suited for such automatic operation in that within reasonable limits not only the motions of the respective work devices relative to each other may be controlled, but the most advantageous travel speeds thereof both during the work stroke and the return stroke may be established.

I accomplish my general objective of an improved speed control for hydraulic power systems and the more specific objective of providing a mechanically simple valve control for establishing a predetermined speed pattern of operation of ironing press elements, by affixing to the piston head-end wall of the cylinder a well within which operate telescopically associated valve sleeves effecting a continuation of the discharge outlet of the pressure generator. With the combined piston and pressure unit in its home position the valve sleeves are disposed entirely within the well in respectively different spaced relationship therewith so that the initial discharge from the hydraulic generator into the cylinder is throttled by the narrow flow passage through which the pressure fluid must pass before it becomes effective against the piston head. As the piston travels in the cylinder the rate of flow into the cylinder changes, assuming substantially constant output from the pressure generator, as the valve elements move relative to each other and to the well; and at a predetermined position of piston travel to the pressure flow from the im-

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5 peller is without appreciable restraint and the piston completes its movement at a maximum rate of speed. On the return stroke of the piston to its home position, which is accomplished after the pressure generator has been de-energized, the valve elements re-arrange themselves in reverse order of operation, as the result of which the piston will drop rapidly through a portion of its stroke, and then at a predetermined, stepped, deceleration through the remainder. In an ironing press in which the shoe element is mechanically associated with the piston to be driven thereby, the restricted flow at the completion of piston return to home position provides an inherent cushion effect to bring the buck and shoe smoothly to their respective home positions.

These and other objects and features of my invention will be apparent from the following detailed description of a presently preferred embodiment thereof taken in association with the accompanying drawings in which Fig. 1 is a front elevation of a typical flatplate domestic ironing machine embodying the invention; Fig. 2 is a side elevation of the ironing machine, partly in section, and showing the relative operating positions of the shoe and buck; Fig. 3 is a sectional elevation of the hydraulic pressure generator unit embodying the invention; Figs. 4 through 11 are somewhat schematic representations of the various stages of operation of the valve unit during the ascending and descending movement of the combined pressure generator and piston; and Fig. 12 is a speed-distance curve illustrating a typical speed pattern obtained with my invention.

A domestic ironing machine of a presently commercially well-known type is illustrated in front view in Fig. 1 and comprises a frame 1 having a table portion 2, a buck 3 and a shoe 4. The buck includes a frame element 5 which is directly affixed to a combined piston and pressure generator 6 operating within a cylinder 7 extending through the table 2 in angular relation therewith. Suitable means such as the integral flange 8 may be employed to secure the cylinder to the table. The shoe 4 is carried by a U-shaped carriage 10 pivotally affixed to lugs 11 depending from the cylinder 7. The carriage 10 is arranged to be driven by the piston assembly 5. A suitable carriage drive mechanism comprises a saddle 12 having arcuate arms pivotally supported with respect to the table 2 by brackets 14 depending therefrom and a cross member 15 extending below and across the carriage 10. The cross member carries a cam roller 16 which co-

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operates with a cam 17 suitably affixed to the carriage 10. Fixed to a cap 18 secured to the end of piston 6 is a bracket having arms 20 extending outwardly and downwardly through cylinder slots 21. Drive links 22 pivotally connect with the arms 20 and the arms of saddle 12. As desired the carriage 10 may be provided with symmetrically arranged brackets 23 to which are pivotally attached guide sleeves 24 through which extend the hanger rods 25 pivotally affixed to the underside of the table, as shown. Above and below said sleeves 24, and arranged to be adjusted as to tension by the illustrated nuts or equivalent, are the shoe cushioning springs 26, 27. Said springs establish a certain resiliency of operation of the shoe as it comes into its operating position and returns to its home position shown in Fig. 2.

As more clearly appears in the United States Patent No. 2,135,266 granted November 1, 1938, to George W. Wardwell, Jr., the combined piston and hydraulic pressure generator includes a cylindrical sleeve 28 having the cap 18 bolted thereto. The sleeve has a grooved upper flange 29 which receives a piston ring 30, and a lower flange 31 which cooperates with a pistonhead plate structure 32 similarly to accommodate a piston ring 33. Disposed within a chamber defined by the outer wall of the sleeve 28 and an internal web 34 is a motor 35 having suitable electrical connections (not shown) emerging from the sleeve 28 and forming elements of a circuit which includes a master control switch 36 on the ironer table and a footpedal control switch 37 which is actuated by the operator to control the actual ironing operation. The motor shaft 38 extends through a suitable gland 40 depending from the web 34 and terminates in an impeller 41 operating within a chamber defined by a housing 42 clamped between plate 32 and a central web portion 43 of the flange 31. The cylinder 7 contains a permanent charge of a light oil or other liquid (not shown) to a level suitably below the piston flange 29. Communication between the interior of the piston 7 and the inlet 44 of the impeller chamber is provided by the enlarged side wall openings 45 in the sleeve 28 and a plate 46 having a plurality of apertures 47 about its periphery. Integral with the flange 31 are a group of angularly arranged walls 48 which direct the oil vortically into the impeller housing via the inlet opening 44. The impeller is supported on a spring biased ball thrust bearing 50 and the web of the impeller may have a desired plurality of openings 51 so as to equalize pressure above and below the impeller. The impeller discharges through a suitable plurality of openings 52 into a chamber 53 provided by the plate 32.

In the ironing press disclosed in the said Wardwell patent the full pressure generated by the impeller is immediately effective against the piston head (corresponding to the plate 32 herein) to drive the piston and impeller unit. The resultant substantially fixed rate of piston travel is disadvantageous in many hydraulically powered mechanisms in which cooperating mutually independent work devices are to be brought in a timed synchronism from a home position to an operating position. In domestic ironing machines particularly, an immediately rapid rise of the buck is psychologically disturbing to the operator, particularly when it is associated with a correspondingly rapid rise of the shoe element. An inexperienced or careless operator may, in

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fact, catch her fingers between the cooperating press elements if the initial action thereof is unduly rapid. Pursuant to the present invention, however, I provide means whereby the operation of the press element is characterized by a slow rate of rise of the buck but a relatively rapid rise of the shoe during that portion of the operating cycle in which the shoe comes to its operative position above the buck, and then by a rapid completion of the upward drive of the buck to its pressing position. In the reverse operation the buck drops rapidly while the shoe remains in position, and then the shoe begins its descent to home position. The final movements of both the buck and the shoe are characterized by slow speed, thus avoiding abrupt stops of the respective work devices.

The presently preferred manner of accomplishing this speed control will best be understood by an examination of the typical valve structure of Fig. 3 and the schematic illustrations of the operation of said valve structure shown in Figs. 4-11 inclusive. I provide the cylinder head with a relatively short neck 54 which receives a sleeve 55 having a flange 56 seated within a counterbore in the inner wall of the cylinder head. Preferably threadedly secured to sleeve 55 is a flange 57 to which is suitably affixed a cap 58. Illustratively, the inner wall of the cap is concentric with the inner wall of sleeve 55 so that these elements collectively define a relatively long well of uniform diameter, although if desired, the inner diameter of sleeve 55 may be larger than that of cap 58. Secured to the plate 32 so as to provide a discharge port therefor is a tubular valve element 60 having an end portion which provides an inwardly directed shoulder 61. In telescoping relation with the inner wall of the valve element and having a flange 62 for engagement with the shoulder 61, is a second tubular valve element 63. For manufacturing expediency I prefer to make valve element 63 in two parts, and therefore apply an outer tubular structure 64. The elements 63, 64 may be pressfit together, or brazed or otherwise permanently secured, to operate as a unit. The end of the valve element 64 has a large aperture 65 and carries a friction ring 66 which restrains free movement of the valve element within the cap 58. The preferably diametrically opposed side wall openings 67 are provided in the composite valve element 63, 64 near the distal end thereof to provide for flow between the impeller and the well, as will be obvious. The shape of opening 67 is pursuant to accepted proportional flow sleeve valve practice.

It will be noted from Fig. 3 that with the unit in its home position there is only a small clearance between sleeve 55 and the valve member 60, and a larger, but nevertheless small clearance between sleeve 64 and the wall of the well. However, this clearance relationship might be reversed to suit another desired speed pattern. Therefore, until the ports 67 emerge from the well the flow of pressure oil from the impeller into the cylinder is throttled by the small wall clearances between the valve elements and the well, it being understood that in the Fig. 5 position, the flow passage provided between the bottom edge of sleeve 60 and sleeve 55 is greater than that of the annular space about sleeve 64. As typical examples of the area of the annular flow passages and other oil travel paths it may be mentioned that the clearance between the sleeve

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60 and the wall of the well provides an area of .021 sq. in.; the clearance between the valve element 64 and the well provides .045 sq. in. of flow area; the maximum exposed area of the port 67 is approximately .600 sq. in.; and the effective flow passage between the upper surface of the impeller and the web 43 is .405 sq. in. area. The effect of these flow relationships is best shown in the schematic diagrams of Figs. 4 to 11 and the speed-travel diagram of Fig. 12. Fig. 4 represents the piston unit in home position immediately after the impeller has begun its operation. Oil flows through the port 67 and enters the cylinder through passage A causing the buck to travel at a rate and for a distance indicated by the plateau of the curve designated A in Fig. 12. As the piston rises the sleeve member 60 emerges from the well, and through the interval represented by Figs. 5 and 6, during which the sleeve 60 commences to lift the tubular valve members 63, 64 the effective flow area is represented at B and the buck speed picks up slightly and continues its movement along the lift B of Fig. 12. At Fig. 7 the piston has risen to a point where port 67 has become effective and the final upward travel of the buck is along the upwardly curving line C. The gradually accelerating speed represented by line C is, of course, the result of the configuration of the port 67, as will readily be understood.

The down travel is similarly influenced by the telescoping valve elements with the additional effect in the early stages of operation of the .405 sq. in. passage D provided by the impeller, for it will be understood that the oil transfers during the down stroke of the piston from beneath the piston, through the various sleeve valve passages and the impeller inlet to its original disposition above the flange 31. It is for that reason that the down curve shown in dotted line in Fig. 12 does not conform in slope or arrangement to the up curve. It is not until the translation of the piston position from Fig. 9 to Fig. 10 that the throttling effect of the gradually diminishing effective area of ports 67 and the respective annular passages becomes appreciated, for in the positions of Figs. 8 and 9 the flow through the .600 sq. in. valve ports is negated by the necessity of flow through the .405 sq. in. passage D. The lost motion connection afforded by the respective flanges 61, 62 delays the emergence of port 67 from the well during the up stroke and delays its return on the down stroke.

The relationship of the shoe to the position of the buck is controlled mainly by the shape of cam 17 and the multiplying effect of the linkages which operate the shoe arm 10. During the period represented by Figs. 4 and 5 it may be assumed that the buck has risen only $\frac{1}{2}$ in. but the shoe has moved through $\frac{1}{2}$ of its travel as represented by dotted line E in Fig. 2. In the interval between Figs. 5 and 6 the shoe has traveled to its home position although as indicated by the lines F and G the cam has decelerated the shoe action so as to bring it smoothly to its operating position. The shoe has completed its movement at line X of Fig. 12, whereas the buck has not. The time interval represented by Figs. 6 and 7 (portion C of Fig. 12) indicates that the buck rises relatively rapidly to its operating position, and at Fig. 7 full ironing pressure is being applied. On the return movement the buck drops about $\frac{3}{4}$ in. at a relatively fast rate, during which the shoe remains fixed. As the piston position reaches the Fig. 10 stage the buck

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has decelerated during approximately $\frac{1}{2}$ in. more of its return travel, and the shoe has begun a fast return movement. At Fig. 11 the buck has dropped its final $\frac{1}{2}$ in. at a slow rate and the shoe has become fully retracted. The final movement of the shoe is cushioned by the springs 26, 27 and the dashpot effect of the restricted oil flow as the valve returns to its positions of maximum flow restriction.

In the embodiment of my invention from which the above noted passage areas and the speed curves of Fig. 12 were taken, the actual elapsed time of movement of the buck to its full up position was approximately two seconds, and the return trip to home position required about three seconds. The shoe reached its operating position during the more than one second interval in which the buck was rising slowly, and there is thus achieved an efficient timed operational relationship. The operator has full control over the mechanism through either the foot pedal switch 37 or the master switch 36; and upon opening either circuit the motor will be stopped and the buck and shoe return to home position.

While I have shown a particular embodiment of my invention, it will be understood, of course, that I do not wish to be limited thereto since many modifications may be made; and I therefore contemplate by the appended claims to cover any such modifications as fall within the true spirit and scope of my invention.

What I claim as new and desire to secure by Letters Patent of the United States is:

1. In combination, a cylinder; a piston therein; a work device to be operated by said piston, and valve means at the entrance to said cylinder for controlling fluid flow therein in relation to the travel of the piston therein, said valve means including a valve body in communication with said cylinder, mutually telescoping tubular valve elements in said body and providing flow paths of different capacities according to the position of said elements with respect to said body, one of said valve elements being mechanically connected to said piston for movement within said body simultaneously with motion of said piston in said cylinder, means for restraining movement of a second valve element with respect to said body, means comprising a lost-motion mechanical linkage between said valve elements to delay movement of said second element until after a predetermined piston movement, and means for introducing pressure fluid into said body through at least one of said valve elements for flow into said cylinder to perform work on said piston.

2. In combination, a cylinder, a piston therein, a work device to be operated by said piston, and valve means for controlling fluid flow into said cylinder in relation to the travel of the piston, said valve means including a valve body in communication with said cylinder, mutually telescoping tubular valve elements in said body in mutually different spaced relationship with the inner wall thereof, one of said valve elements being mechanically connected to said piston for movement within said body simultaneously with motion of said piston in said cylinder and the full work stroke of said piston withdrawing said element entirely from said body; means for restraining movement of a second valve element with respect to said body; means comprising a lost motion connection between said valve elements to delay movement of the second element until after a predetermined piston movement; and means for

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introducing pressure fluid into said body through at least one of said valve elements for flow into said cylinder to perform work on said piston, the rate of flow between said valve body and said cylinder being at least in part related to the minimum annular flow space afforded between a valve element and said body wall.

3. In combination, a cylinder, a piston therein, work devices to be operated by said piston, a valve body fixed to the cylinder wall at the piston head end and opening into said cylinder, mutually telescoping tubular valve elements disposed within said valve body in spaced relationship with the wall thereof, one of said valve elements being affixed to the piston to be withdrawn thereby entirely from said valve body during the work stroke of the piston; means for restraining movement of a second of said valve elements within said body; means for effecting a lost motion connection between said valve elements to delay movement of said second element outwardly of said valve body into said cylinder until after a predetermined movement of said piston, said second valve element having wall ports of substantially greater capacity than the maximum annular space provided between either of said valve elements and said body and arranged for emergence into a space having a greater flow capacity than said ports during movement of said second element; and means including said valve elements for introducing pressure fluid into said cylinder initially solely by way of said annular space and ultimately directly by way of said wall ports.

4. In combination, a cylinder, a piston therein, a work device to be operated by said piston, a valve body fixed to the cylinder wall at the piston head end and opening into said cylinder, mutually telescoping tubular valve elements disposed within said valve body in spaced relationship with the wall thereof, means for moving one of said valve elements within said body at a rate proportional to, and in the direction of, piston movement within said cylinder, said valve element being withdrawn from said valve body into said cylinder during the work stroke thereof; means for restraining movement of a second of said valve elements with said body; means for effecting a lost motion connection between said valve elements to delay movement of said second element outwardly of said valve body into said cylinder until after a predetermined movement of said piston, said second valve element having wall ports of substantially different capacity than the annular space provided between said valve elements and said body and arranged for emergence from said body into said cylinder during movement of said second element; and means including said valve elements for introducing pressure fluid into said cylinder initially by way of said annular space and ultimately directly by way of said valve ports.

5. In combination, a cylinder, a piston therein, a work device to be operated by said piston, a valve body fixed to the cylinder wall at the piston head end thereof and opening into said cylinder, mutually telescoping valve elements disposed within said valve body in spaced relationship with the wall thereof, one of said valve elements being affixed to the piston to be withdrawn entirely from said valve body during the work stroke of the piston; means carried by a second of said valve elements for frictionally engaging the side wall of said body to restrain free movement of said body therein; means for effecting a lost motion connection between said valve ele-

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ments to delay movement of said second element outwardly of said valve body into said cylinder until after a predetermined movement of said piston, said second valve element being configured to pass a greater amount of fluid into said cylinder at a predetermined point of emergence thereinto than could be passed through the annular space between said first element and said valve body; and means including said second valve element for introducing pressure fluid from a source of pressure into said valve body for ultimate passage into said cylinder.

6. In combination, a cylinder; a piston therein; work devices for operation by said piston; a closed-ended well fixed to said cylinder at the piston head end thereof, said well opening directly into said cylinder; mutually telescoping tubular valve elements disposed in said well in spaced relationship with the wall thereof, one of said elements being affixed to said piston to be moved during the stroke thereof from a position at least partially within said well to entire removal therefrom and the other of said elements having adjacent its distal end wall ports of substantially greater flow capacity than afforded by the clearance between said first member and said well; means for effecting a lost motion connection between said valve elements to delay the emergence of said enlarged ports into said cylinder until after a piston movement greater than that represented by the spacing of the said ports below the surface of said well; and means within the piston and communicating with the first-named valve element to discharge pressure fluid through said element for eventual passage into said cylinder to propel the piston therein.

7. The combination according to claim 6 in which the annular space between said well and the respective valve elements is substantially different so that as the first element emerges from the well the flow of pressure fluid from said well into said cylinder is at a different rate.

8. The combination according to claim 6 in which the annular space between said well and the respective valve elements is arranged to afford substantially different flow rates from said well into said cylinder as the valve members emerge from the well, and the flow capacity of the ports of said second valve member is greater than that afforded by any of said annular spaces.

9. In an ironing machine having a cylinder, a piston, fluid pressure supply means, and a work device affixed to said piston to be operated thereby; means for returning said piston means to a home position adjacent the base of the cylinder; means for operating said fluid pressure supply means to create pressure fluid to perform work on said piston; and means for regulating the rate of discharge from said pressure supply means into said cylinder according to the position of said piston therein, comprising a well extending from the base of the cylinder and communicating therewith, a sleeve valve member fixed to said pressure generating means at the outflow port thereof and extending into said well in spaced relationship therewith, a second sleeve valve member telescopically associated with said first member and projecting therefrom in different spaced relationship relative to said well, means for restraining free movement of said second sleeve relative to said well, and means comprising a lost motion connection between said sleeves; the relationship of said sleeves to the piston travel being such that in home position both of said sleeves are within said well and on

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movement of said piston the first sleeve substantially completely emerges from the well before any substantial movement of the second sleeve therein, whereby during the last part of piston movement the rate of pressure flow into said cylinder is a function of the clearance between the second sleeve and the well.

10. In an ironing machine according to claim 9 the provision of enlarged ports in the second sleeve near its distal end, said ports having a greater flow capacity than that afforded by the spaced relationship of either sleeve with the well, and said ports emerging from the well for discharge directly into the cylinder during a late portion of piston movement.

11. In an ironing machine having a cylinder, piston means in said cylinder including a motor driven impeller disposed at the head of said piston, said cylinder having a permanent reservoir of liquid to be fed to said impeller through ports including a relatively small area entrance passage to the housing of said impeller; a work device mechanically connected to said piston means to be driven thereby; means for returning said piston means to a home position closely adjacent the piston head whereby the bulk of the liquid charge is behind the piston head; and valve means for controlling the flow of pressure fluid against the piston head and the return of the fluid to the reservoir at rates related to the extent of movement of said piston, comprising a closed-ended well fixed to said cylinder at the piston head end thereof, said well opening di-

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rectly into said cylinder, mutually telescoping tubular valve elements disposed in said well in spaced relationship with the wall thereof, one of said elements affixed to said piston head to receive discharge directly from said impeller and a second of said elements being mechanically associated with said first element by a lost-motion connection; each of said elements when the piston is in home position occupying said well, and with the piston in work position being substantially withdrawn from said well; the second element being configured to pass fluid into said cylinder at a maximum rate during the final portion of the piston work stroke; the flow capacity of said impeller to return liquid to the cylinder reservoir during the piston return to home position being less than the last-named rate but greater than the flow rate afforded by the clearance between the valve elements and the well, whereby the first portion of the return stroke of the piston will be at a slower rate than the final portion of the work stroke.

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The following references are of record in the file of this patent:

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30	Number	Name	Date
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	2,135,266	Wardwell	Nov. 1, 1938
	2,475,304	Bariff	July 5, 1949