June 11, 1940. C. A. CAMPBELL 2,204,119

LUBRICATOR

Filed May 9, 1938 2 Sheets-Sheet 1

Fig. 1

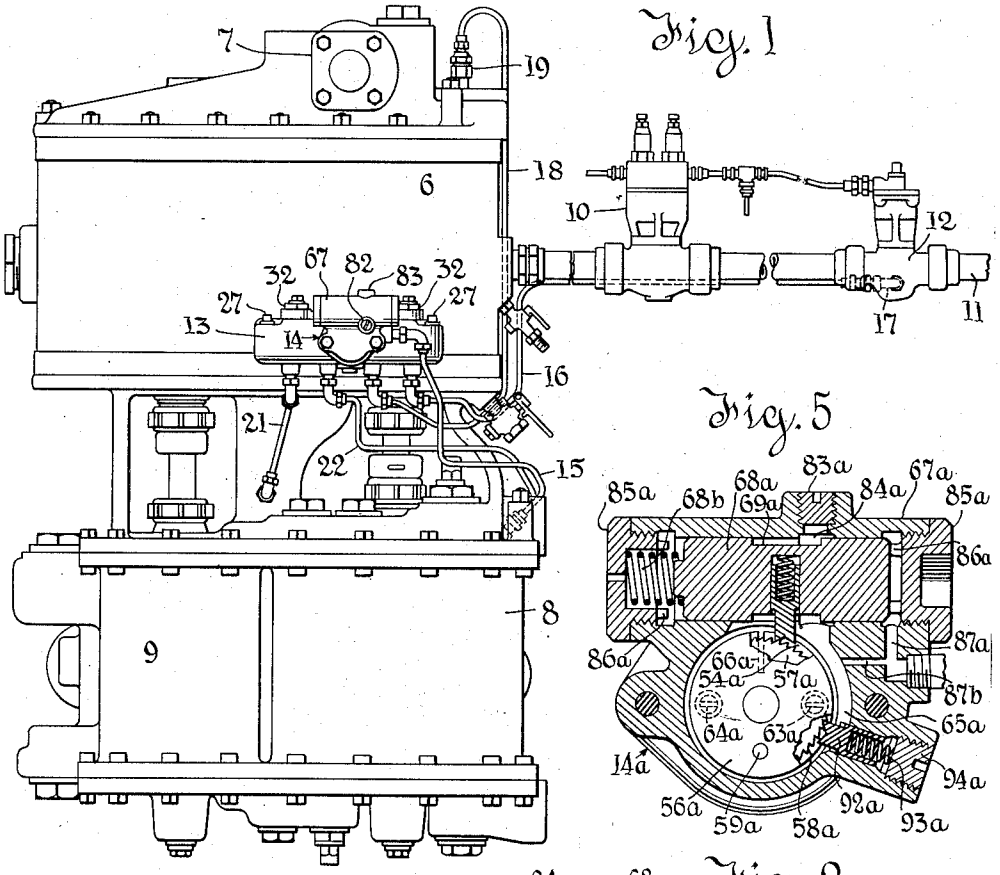
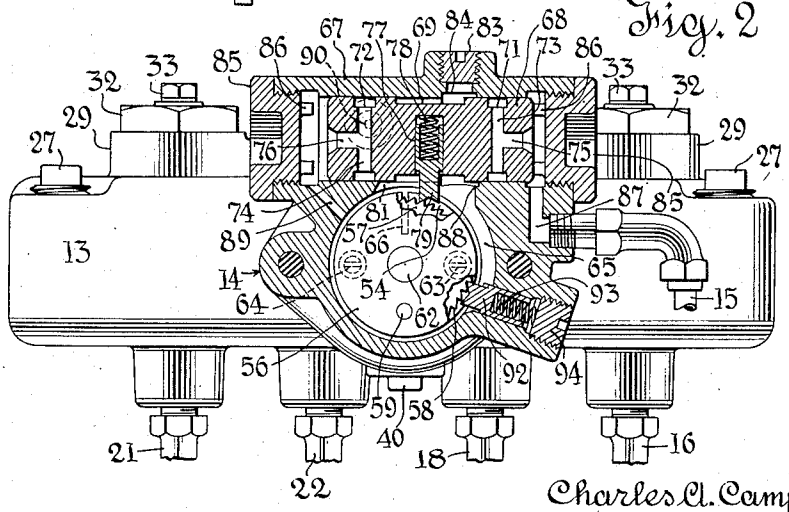


Fig. 5

Fig. 2



Inventor

Charles A. Campbell

By

Dodge and Sons

Attorneys

June 11, 1940. C. A. CAMPBELL 2,204,119

LUBRICATOR

Filed May 9, 1938 2 Sheets-Sheet 2

Fig. 3

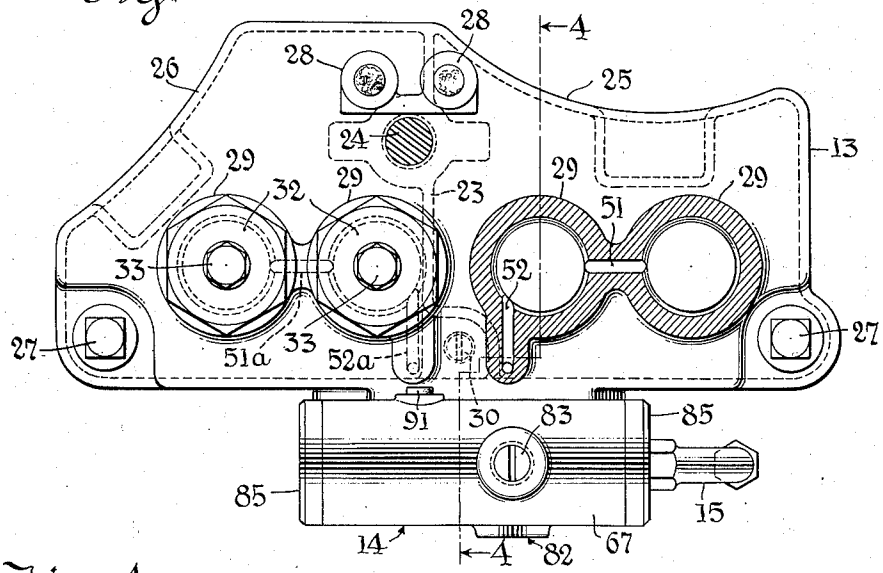
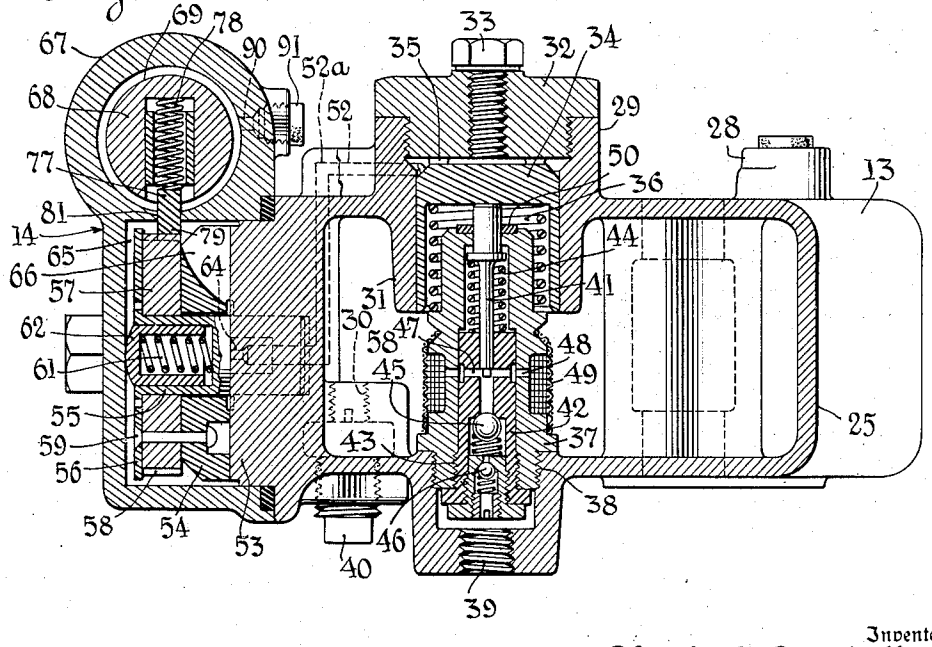


Fig. 4



Inventor

Charles A. Campbell

By

Attorneys

Patented June 11, 1940

2,204,119

UNITED STATES PATENT OFFICE

2,204,119

LUBRICATOR

Charles A. Campbell, Watertown, N. Y., assignor to The New York Air Brake Company, a corporation of New Jersey

Application May 9, 1938, Serial No. 206,861

6 Claims. (Cl. 184—29)

This invention relates to force feed lubricators and particularly to such lubricators for use with direct connected steam driven compressors having no rotary parts. A familiar example is the type of compressor (simple or compound) used as a part of the air brake system in steam locomotives.

In a compressor of this type there is no externally accessible part from which motion can be derived conveniently, and furthermore the amount of oil to be fed per stroke is so small that any oil pump operated in time with the reciprocations of the compressor will afford an intolerably variable rate of delivery. This is because the total displacement of the pump plunger would be so small as to be seriously affected by leakage and by slip through the pump valves. It has been found impossible to hold this slip within extremely close limits.

According to the present invention an oil feeding pump of proportionally large capacity is operated by an impulse motor which receives an air pressure impulse from a low stage working space of the compressor once in a considerable number of strokes. Forty-eight has been selected as a convenient number because it permits satisfactory accuracy of feed and is divisible by 2, 3, 4, 6 and 8 which are common numbers of lubricated points for which individual pumps can be provided.

To time the impulses for one or more lubricating pumps a rotary distributor valve is used and is driven by an impulse plunger subject to impulses in the same working space. The drive to the distributor is conveniently through a pawl and ratchet wheel (having 48 teeth in the preferred example illustrated). This arrangement permits a single impulse connection from the pump working space to the distributor valve actuating piston and to the distributor valve, the latter delivering selected impulses to the impulse motors of selected pumps.

Thus a plurality of pumps or groups of pumps may be operated successively at such intervals as to permit the pumps to be given a delivery displacement per stroke large enough to ensure reasonably precise measurement of oil.

Any suitable number of pumps may be controlled by a single distributor and they may be operated according to the design of the distributor in any desired order, and singly or in groups.

The example chosen for illustration has four pumps arranged in pairs. Thus the pairs are operated in alternation so that alternating pairs are operated every 24 strokes of the compressor. Hence each pump delivers once in 48 strokes.

There are various mechanical details of importance which will be understood as the description proceeds.

In the accompanying drawings:

Fig. 1 is an elevation of a cross-compound compressor with the lubricator applied.

Fig. 2 is an elevation of the lubricator with the body of the distributor sectioned to show the distributor motor, rotary valve and ratchet feed.

Fig. 3 is a plan view, partly sectional.

Fig. 4 is a section on the line 4—4 of Fig. 3.

Fig. 5 is a view similar to a portion of Fig. 2 but showing a slightly modified construction.

Referring first to Fig. 1, the high pressure steam cylinder is to the right of the casing 6 which encloses lagging and the low pressure steam cylinder is to the left therein, while the steam distributing valve mechanism, whose form is immaterial to the present invention, is housed in the casing 7. The first or low stage compressor cylinder is indicated at 8, and the second or high pressure compressor cylinder is indicated at 9. These are equipped with automatic valves whose form is not material to the present invention.

The steam supply line 11 leading to the steam end of the compressor is controlled by a throttle valve 12 and pump governor 10, which last responds to pressure in the usual main reservoir (not shown) to start and stop the compressor. The lubricator is bolted to the lower flange of the high pressure steam cylinder, and a portion of it enters the casing 6 so that it lies partly within the lagging on the steam cylinder and is heated by conduction from the high pressure steam cylinder walls. The purpose of this is to keep the oil in a fluid condition irrespective of atmospheric temperature.

In Fig. 1 the lubricator is generally indicated by the numeral 13 applied to the main body. The numeral 14 indicates the body housing the impulse distributor mechanism.

A tube 15 which leads from the upper working space of the low pressure air cylinder is the connection through which air impulses are delivered to the lubricator. On the upward stroke of the low pressure piston a pressure impulse rising to about 40 pounds gauge is delivered, and on the ensuing downward stroke the pressure drops to a value slightly below atmospheric pressure. The tubes 16 and 18 are oil delivery connections. The tube 16 leads to the throttle valve 12 through

a terminal check valve 17. This is to protect the lubricator against back flow and against the entrance of water. The tube 18 leads through a terminal check valve 19 to the steam inlet of the steam distributing valve in casing 7. The terminal check valve 19 is for the same purpose as the terminal check valve 17.

A tube 21 leads to and delivers through a port in the intermediate frame of the compressor leading to the interior of the high pressure cylinder 9. The tube 22 leads to the interior of the low pressure air cylinder 8. In these two connections no terminal check valves need be used for the reason that they lead downward and there is no danger of the entry of water into the connections from the compressor.

Reference should now be made to Figs. 2, 3 and 4. It will be observed that the main body 13 of the lubricator is hollow and is divided into two oil reservoir chambers by an internal partition 23 through which partition the mounting bolt 24 extends. This is the bolt that attaches the lubricator to the flange of the compressor cylinder.

Referring to Fig. 3, the arcuate vertical face 25 conforms to the contour of the high pressure steam cylinder, and the arcuate face 26 conforms similarly to the low pressure steam cylinder. There are two filling plugs 27, one for each oil chamber, and there are two air vent connections 28 each equipped with a felt dust excluding filter, there being one such vent for each of the two oil reservoirs. It is considered better to use two distinct reservoirs, but this construction is not absolutely essential as all of the pumps could draw from a single reservoir instead of having two reservoirs, one for each pair of pumps as hereinafter explained.

Removal of plug 30 in an offset portion of partition 23 (see Figs. 3 and 4) will connect the reservoirs. Access to plug 30 may be had by removing plug 40.

Formed in the upper wall of the housing 13 are four cylinder forming sleeves each comprising coaxial cylindrical bosses, an upwardly extending boss 29 and a downwardly extending boss 31. The upper end of each upper boss is closed by corresponding cylinder head 32 with a removable screw plug 33.

Since the pump mechanisms now about to be described are identical, the description will proceed with reference to a single one, that is, the third one from the left in Fig. 3, which is shown in axial vertical section in Fig. 4.

A cup-shaped or thimble piston 34 works in the cylinder just described and is limited in its upward motion by stand-off bosses or lugs 35. It is urged upward by a coiled compression spring 36 which is housed within the skirt of the piston and reacts between the piston and a shoulder or spring seat formed on the body 37 of the oil pump unit.

The pump body 37 is threaded at 38 into the lower wall of body 13 and beneath the unit is a tapped connection 39, which in the case of the pump shown in section in Fig. 3, is intended to receive the oil tube 18 already mentioned. The upper end of the body 37 serves as a guide for the upper end of a pump plunger 41 which works in a cylinder bushing 42 inserted through the lower end of the body 37 with which it makes a slip-fit and with which it is connected by threads indicated at 43. The cylinder bushing 42 is axially bored to receive the plunger 41 and this is urged upward by a coil compression spring 44 which reacts between the upper end of the cylinder bushing and a flange on the plunger as clearly shown in Fig. 3.

The lower end of the cylinder bushing 42 is counterbored to receive a double ball check assembly, the ball check valves serving as the discharge valves of the pump. These ball check valves are indicated at 45 and 46 and the construction is so clearly shown in Fig. 4 that it seems necessary merely to state that the ball checks are seated by coiled compression springs and that the retainer and spring seat for the upper and larger ball check serves as a housing and seat for the lower and smaller ball valve. A still smaller threaded plug acts as a retainer and spring seat for the smaller ball check valve. Oil displaced by the plunger 41 flows past the valves to the connection 39 and thence by the tube 18 and terminal check 19 to the interior of the steam distributing valve.

The cylinder bushing 42 is formed with a plurality of radial ports 47 which communicate with the peripheral groove formed in the bushing and arranged to communicate with radial ports 48 in the pump body 37. A screen 49 protects the pump against the entrance of any solids which might be carried by the oil. When the piston 34 is forced down pneumatically, it forces down the pump plunger 41. In the first portion of its downward excursion it closes the inlet ports 47 and then displaces oil past the valves 45 and 46 to the point of use. When the pressure impulses above the piston fade out, the spring 36 restores the motor piston 34 and the spring 44 restores the pump plunger 41.

In its upward motion the pump plunger tends to draw a vacuum in the oil pump cylinder so that when the inlet port 47 is cleared, oil promptly enters from the reservoir. The shim indicated at 50 is used to adjust the delivery capacity of the oil pump. This shim limits the downward motion of the piston 34 and consequently the displacing stroke of the oil pump. By inserting shims of different thicknesses or by eliminating the shim altogether, the displacing stroke of the oil pump may be changed to suit the requirements of the particular part being lubricated. To make the interchange it is necessary to remove the cap 32 above the piston 34, withdraw the piston, place the proper shim, and restore the parts.

The operative characteristics of the oil pumping unit just described, are of substantial practical importance. In the first place the displacing stroke and the suction stroke are both rather rapid so that any leakage or slip will be minimized. Second, the motor piston acts in simple thrust on the oil piston. Consequently, there are no lateral stresses to cause wear or leakage.

Examination of Fig. 3 will indicate that the pumps are arranged in pairs, two pumps of a pair drawing from a corresponding one of the two chambers of the oil reservoir in housing 13. The two pumps of each pair are operated in unison, for the working spaces above the two pistons 34 of a pair are connected by a cross-port 51 and the pressure surge connection 52 consequently delivers to both pump motors. The other pair is similarly arranged. (See posts 52a, 51a on Fig. 3.)

The four pumps illustrated in Fig. 3 are shown in alignment, but one of the important features of the invention is that since each pump has its own motor, the location of the pump may be modified to accord with the space available. This idea of individual pumps pneumatically op-

erated gives very great flexibility as to arrangement.

The distributing valve mechanism for delivering the impulses to the various oil pump motors which will now be described with reference to Figs. 2 and 4, is illustrated as mounted in the housing 14 bolted directly to the front of the housing 13. Since the distributor mechanism merely delivers pneumatic impulses to the oil pump motor, it is obviously possible to mount the distributor mechanism at any convenient point, either on the housing 13 or elsewhere. The only operative relation which it has to the old pump motor is afforded by pressure transmitting passages and obviously these can be provided in a variety of ways so that the specific arrangement here shown, while preferred, nevertheless should be considered merely as illustrative of one convenient arrangement.

Formed on the front face of the body 13 is a circular boss 53 which serves as a seat for a rotary distributor valve 54. The rotary valve 54 is centered by pin 55 which has a flange or head 56. Between head 56 and valve 54 and centered on pin 55 is the ratchet toothed drive disc 57 having in the example illustrated forty-eight teeth 58. Rivets 59 hold the parts 54, 56, 57 together. A spring 61 in cup-shaped plunger 62 acts to hold valve 54 against seat 53.

Formed in seat 53 are two ports, the right-hand one of which, 63, leads to port 52 for the right-hand group of pumps shown in Fig. 3 and the left-hand one of which, 64, leads to port 52*a* for the left-hand group. Ports 63 and 64 are 180° apart with respect to the rotation to the valve and each is connected once in each rotation of valve 54 to chamber 65 in which valve 54 is mounted, by means of a radial slot port 66. (See Fig. 4.)

The ports 63 and 64 are shown (see Fig. 2) as slots radial with respect to the rotary valve, and conveniently are formed by pressing slotted bushings into round holes formed in the seat 53. This is simply a convenient way to form the slot ports which register accurately with the radial slot port 66.

In the upper portion of housing 14 is cylinder 67 for the double-acting impulse piston 68. Piston 68 is relieved or reduced in diameter at its mid-length as shown at 69 so that it has two end head portions which are peripherally grooved at 71 and 72 respectively. Cross ports 73 and 74 with axial ports 75, 76 connect grooves 71 and 72 with respective working spaces at the opposite ends of the piston.

A transverse guide passage receives ratchet plunger 77 and spring 78 and is located at mid-length of the piston so that the latter is symmetrical. Plunger 77 has a fish-tail toothed end 79 to coact with ratchet teeth 58. Hence piston and ratchet may be assembled in reversed positions without effect on function. In this way faulty assembly is precluded.

The end of ratchet plunger 77 works in slot 81 connecting cylinder 67 with valve chamber 65. Plug 82 (Figs. 1 and 3) and plug 83 (Figs. 2 and 3) permit removal of the ratchet plunger. To do so, the plugs are removed, the plunger is pushed back, so that its end may turn in groove 84 as piston 68 is rotated in the cylinder through 180°. This permits removal of the plunger through the tapped hole in which plug 83 is normally mounted, without dismounting the rotary valve 54 and drive disc 57. If these parts be removed from the housing the ratchet plunger 77 may be inserted from the valve chamber 65 where the intersection of the groove 84 produces an enlargement of the slot 81.

The ends of cylinder 67 are closed by two identical heads 85 which are screwed into place, each having stand-off lugs 86 to limit the reciprocation of piston 68.

Connection 15 from the upper low pressure compressor working space leads to the right end of cylinder 67 by passage 87. On the discharge stroke pressure rises so that piston 68 moves to the left turning valve 54 counter-clockwise. The motion is so limited by lugs 86 that the valve is turned one tooth only.

Just before the limit of motion is reached groove 71 opens past shoulder 88 to chamber 65 so that compressor pressure is admitted to this chamber. Port 89 then admits the pressure developed in the chamber to act against the left end of piston 68, but by this time the piston will have moved full stroke. If port 66 then registers with either port 63 or 64 the corresponding group of oil pumps will be actuated.

On the suction stroke in the compressor working space there is back flow through passage 87 and connection 15, so that piston 68 shifts back to the right. Just before it reaches its right-hand limit of motion groove 72 opens to vent port 90 which leads to atmosphere through filter plug 91. This causes pressure in chamber 65 to drop to atmospheric, allowing pistons 34 to move up if they have been depressed and conditioning the whole device for the next cycle of the compressor.

A ratchet plunger 92 identical with plunger 77, urged by spring 93 and removable upon removal of plug 94 prevents retrograde motion of valve 54 by its engagement wth teeth 58.

The impulse motor just described has important functional advantages. The chamber 65 is at atmospheric pressure until a step rotation of valve 54 has been nearly completed. This reduces the friction of the rotary valve and ensures that the rotary valve opens and closes ports 63 and 64 before pressure is established in chamber 65, because the pressure is not admitted to the chamber until groove 71 overtravels shoulder 88, and this occurs near the end of the piston stroke when the advance of the rotary valve has been substantially completed. Hence double actuation of an oil pump or group of oil pumps can never occur. Since the impulse piston is pressure-actuated in each direction no return spring is needed, and one would be required if a single acting plunger were adopted. Such a spring is considered objectionable because it is subject to possible failure.

However, a satisfactory lubricator can be constructed by using a single acting piston with a spring return of the piston, and further, it is not strictly essential that the piston have any valve function, i. e. it is not necessary that the piston shall time the impulse. It can be timed with sufficient precision by the rotary valve.

A construction of this type is shown in Fig. 5. In this figure parts corresponding to similar parts in Fig. 2 are identified by the reference numerals used in Fig. 2 each with the distinguishing letter *a*. No detailed description of such parts is considered necessary.

The differences are as follows. There are no peripheral grooves corresponding to the grooves 71 and 72; no cross ports such as the port 73 or 74; and no axial ports such as 75 or 76. The two cylinder heads 85*a* are not identical, but the

one to the left is formed to receive a piston return spring 68b so that the piston is forced to the left during the delivery or discharge stroke in the upper working space of cylinder 8 and returns to the right under the urge of the spring upon the ensuing suction stroke.

The passage 87a, which is the surge passage, has a constantly open branch of rather small area indicated at 87b and leading from the surge passage directly to the rotary valve chamber 65a. Thus the pressure surges are communicated directly to the rotary valve chamber with only such delay as is incident to the restriction at 87b. The purpose of this delay is to cause the piston 68a to shift the rotary valve to its new position on rise of pressure before this rising pressure becomes effective in the chamber 65a. It is possible, by suitably proportioning the restricted port 87b, to make sure that the oil pumps will not be caused to make two successive delivery strokes, as it might happen if pressure could rise rapidly in the chamber 65a.

This double actuation is absolutely prevented in the structure of Fig. 2 because the piston in Fig. 2 must shift the rotary valve to its new position before any pressure surge is admitted to the rotary valve chamber. Thus the construction just described with reference to Fig. 5 is operative and commercially useful, but the refined embodiment shown in Fig. 2 is preferred because it avoids the use of the return spring and because the timing function in the piston prevents double actuation, as explained.

In the arrangement shown in the drawings the two groups of two pumps each are operated alternately at uniform intervals. Stated differently, for 23 strokes the impulses merely feed the rotary valve. On the 24th stroke they feed the rotary valve and deliver an actuating impulse to one group of oil pumps. Then for 23 strokes the rotary valve advances and finally on the next (48th) stroke the valve completes a rotation and delivers an impulse to the other group of oil pumps.

It is possible to prime any pump manually without affecting any other pump by removing the cap screw 33 and actuating the piston 34 by means of a small push rod.

The invention offers, as explained, the utmost flexibility in the location of the oil pump units and in the location of the distributor with its actuating motor. The only actuating connection from the compressor is the air connection 15. The pumps are not subject to damage by side thrust, they cannot be overloaded. All the components of the mechanism are simple in form and a large proportion of them are susceptible of manufacture on automatic machines. The use of a pneumatically actuated distributor to secure spaced pneumatic impulses for operating the oil pump makes it possible to increase the stroke displacement of the pumps to a point at which valve slip and leakage become negligible as disturbing factors.

In the construction shown in Figs. 1 to 4 the rotary valve determines when an oil pump motor is to receive an impulse but the piston 68, by its groove 71 actually times the delivery of the impulse to the pump motor. This is a desirable but not strictly indispensable feature, for the timing can be performed by the rotary distributing valve as indicated in Fig. 5.

While the lubricator particularly meets the requirements of the railway air brake service and meets conditions peculiar to that service, it is believed to have utility in other fields and is susceptible of modification to meet conditions peculiar to particular installations. It can be used to lubricate any device having a working space in which there are recurrent pressure impulses.

What is claimed is:

1. A lubricator for devices having a working space in which a pulsing fluid pressure is developed as an incident to operation, said lubricator comprising in combination a lubricant delivering force pump; a pressure-operated motor of the expansible chamber type capable of responding to said pulsing fluid pressure and connected to actuate said pump; a distributing valve comprising a rotary disc valve mounted in a chamber, said valve having a recurrent series of successive positions in at least one of which positions it establishes a connection between said chamber and said motor and in the others of which it interrupts such connection; a second motor of the expansible chamber type having a working space connected with the working space of said device, said second motor being capable of reciprocating in response to said pulsing fluid pressure; an intermittent drive connection between said second motor and said rotary disc valve whereby the motor rotates the valve step by step through said series of positions; and means establishing a connection between the working space in which the pulsing fluid pressure is developed and the chamber of said distributing valve, said connection being arranged to delay the development of a pressure pulse in said chamber whereby the second motor will respond to a given pressure pulse before the same becomes substantially effective in said valve chamber.

2. The combination defined in claim 1 in which the connection for delaying the development of pressure in the distributing valve chamber comprises valve means operated by said second motor means.

3. The combination defined in claim 1 in which the connection from the working space to the second motor is relatively large and that to the chamber of the distributing valve is relatively restricted.

4. A lubricator for devices having a working space in which a pulsing fluid pressure is developed as an incident to operation, said lubricator comprising in combination a lubricant delivering force pump; a pressure operated motor of the expansible chamber type capable of responding to said pulsing fluid pressure and connected to actuate said pump; a distributing valve comprising a valve member shiftable in a chamber through a recurrent series of positions in at least one of which it offers a connection between its chamber and said motor and in the others of which it interrupts such connection; a reciprocating piston subject to and reciprocable by said pulsing pressure; an intermittent drive connection through which said piston shifts said distributing valve step by step through its series of positions; and valve means associated with said reciprocating piston and controlling a vent from said valve chamber and also controlling a connection between said valve chamber and said working space, said valve means being so timed relatively to the reciprocation of the piston that in the normal position of the piston the valve opens said vent and interrupts said connection and as the piston completes its excursion in re-

sponse to a rising pressure surge, the valve closes the vent and establishes said connection.

5. The combination defined in claim 4 in which the reciprocating piston is double acting and subject on one end to pressure in said distributing valve chamber.

6. A lubricator for devices having a working space in which pulsing fluid pressure is developed as an incident to operation, said lubricator comprising in combination, means forming an oil reservoir; a force pump mounted in said reservoir, said pump being connected to receive charges of oil from said reservoir and to discharge it to the point to be lubricated; a piston motor connected with said pump to actuate the same, said motor being capable of responding to pressure pulsations in said working space when connected therewith; a distributing valve comprising a ported rotary valve mounted in a valve chamber and having a series of positions in at least one of which its port establishes a connection between said valve chamber and said motor and in the others of which the valve interrupts such connection; a movable abutment subject to pressure in said working space and capable of reciprocating through a limited range in response to the pulsing fluid pressures therein; valve means associated with said abutment and arranged to open and close a communication between said working space and said rotary valve chamber as the abutment reciprocates; and a ratchet mechanism interposed between said abutment and said rotary valve and adapted to serve as a drive connection through which said abutment shifts said rotary valve step by step through a recurrent series of positions.

CHARLES A. CAMPBELL.