United States Patent [19]

Bostwick et al.

[11] **3,726,608**

[45] **Apr. 10, 1973**

[54] **FUEL INJECTION PUMP TIMING DEVICE**

[75] Inventors: **Norvan W. Bostwick,** Longmeadow, Mass.; **Warren E. Snyder,** Brookfield, Wis.; **Ignace J. Daborowski,** Springfield, Mass.

[73] Assignee: **AMBAC Industries, Inc.,** Springfield, Mass.

[22] Filed: **Apr. 26, 1971**

[21] Appl. No.: **137,180**

[52] **U.S. Cl.** .....**417/212,** 64/25, 123/139 AP

[51] **Int. Cl.**.....**F16d 5/00**

[58] **Field of Search** .....123/139 AE, 139 AP, 123/139 AQ; 64/25; 417/212

[56] **References Cited**

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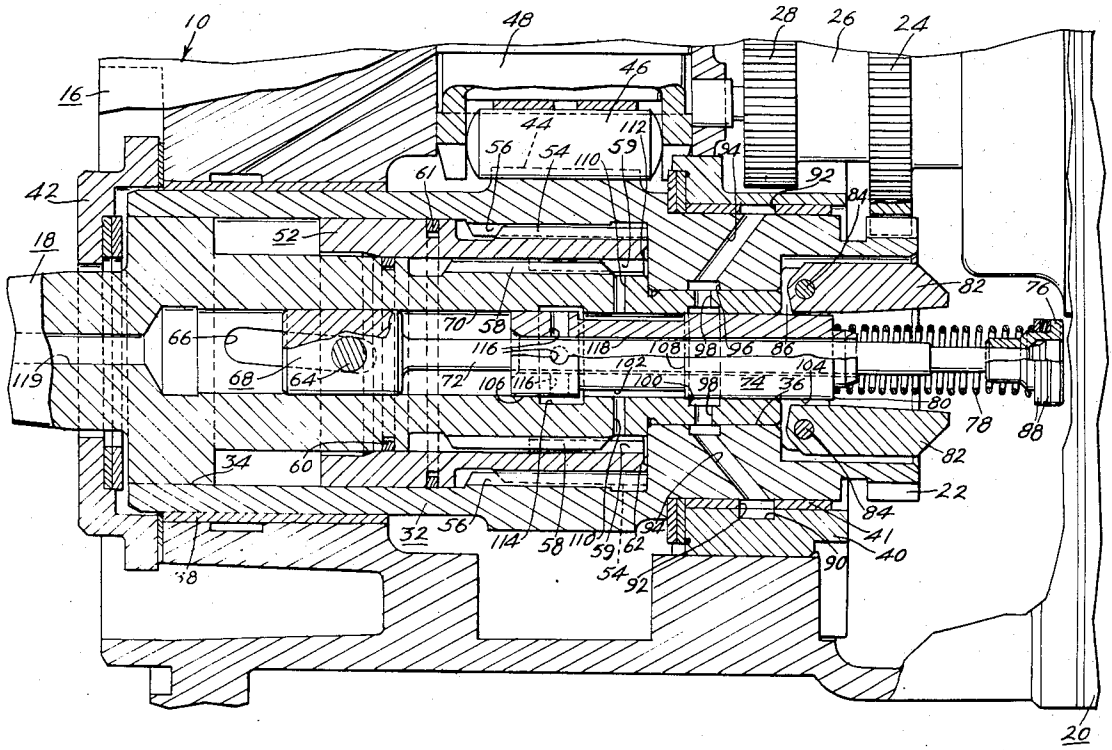
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Primary Examiner—Manuel A. Antonakas
Attorney—Howson and Howson

[57] **ABSTRACT**

An integral timing device for fuel injection pumps including a servo-actuated helically splined sleeve for varying the relative angular position of the pump drive shaft and the cam driven by the drive shaft which serves to reciprocate the pump plunger. The invention includes means for providing a positive trapping of the sleeve actuating fluid during the plunger injection interval to minimize the shifting of the cam towards a retard position and to reduce the pressure requirements of the device. In addition, means are provided for automatically retarding the pump timing during the starting of the engine.

11 Claims, 9 Drawing Figures



PATENTED APR 10 1973

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SHEET 1 OF 5

FIG. 1.

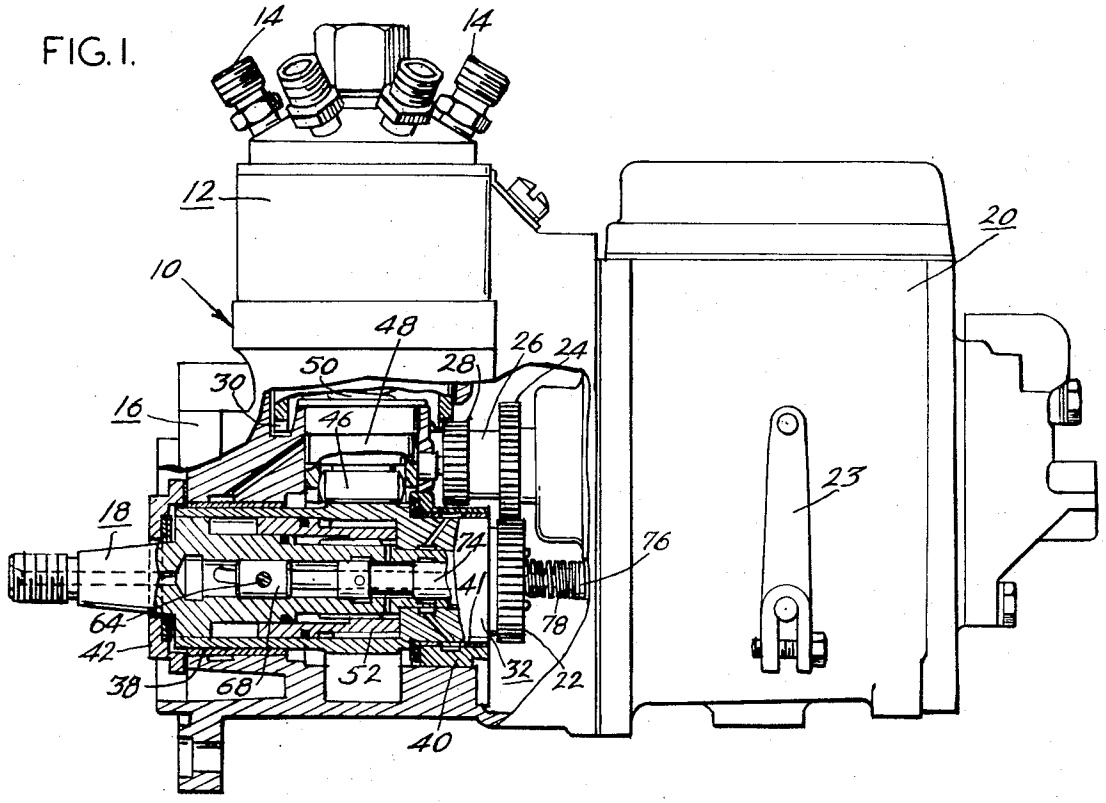
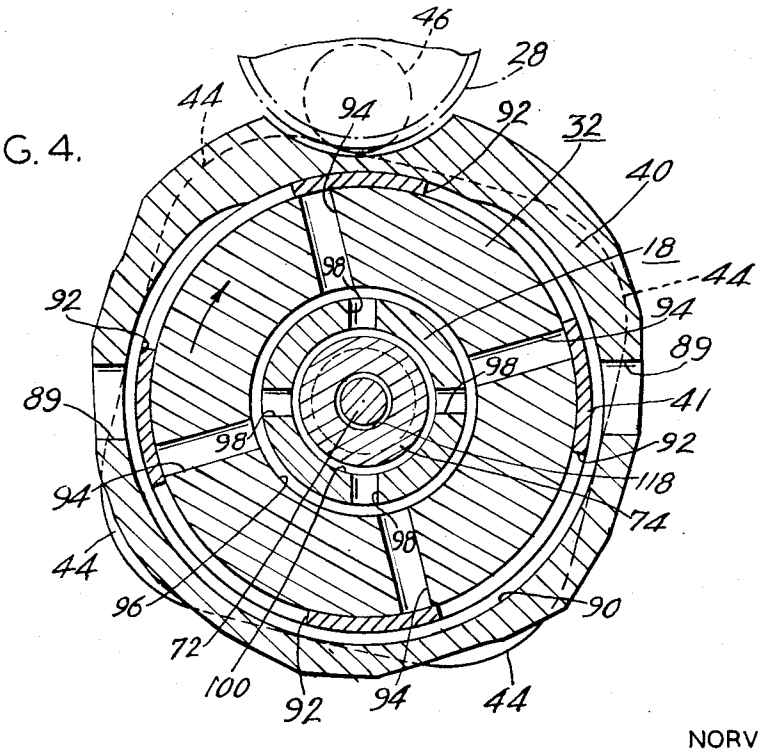


FIG. 4.



INVENTORS:
NORVAN W. BOSTWICK
WARREN E. SNYDER
IGNACE J. DABOROWSKI
BY Howson & Howson
ATTYS.

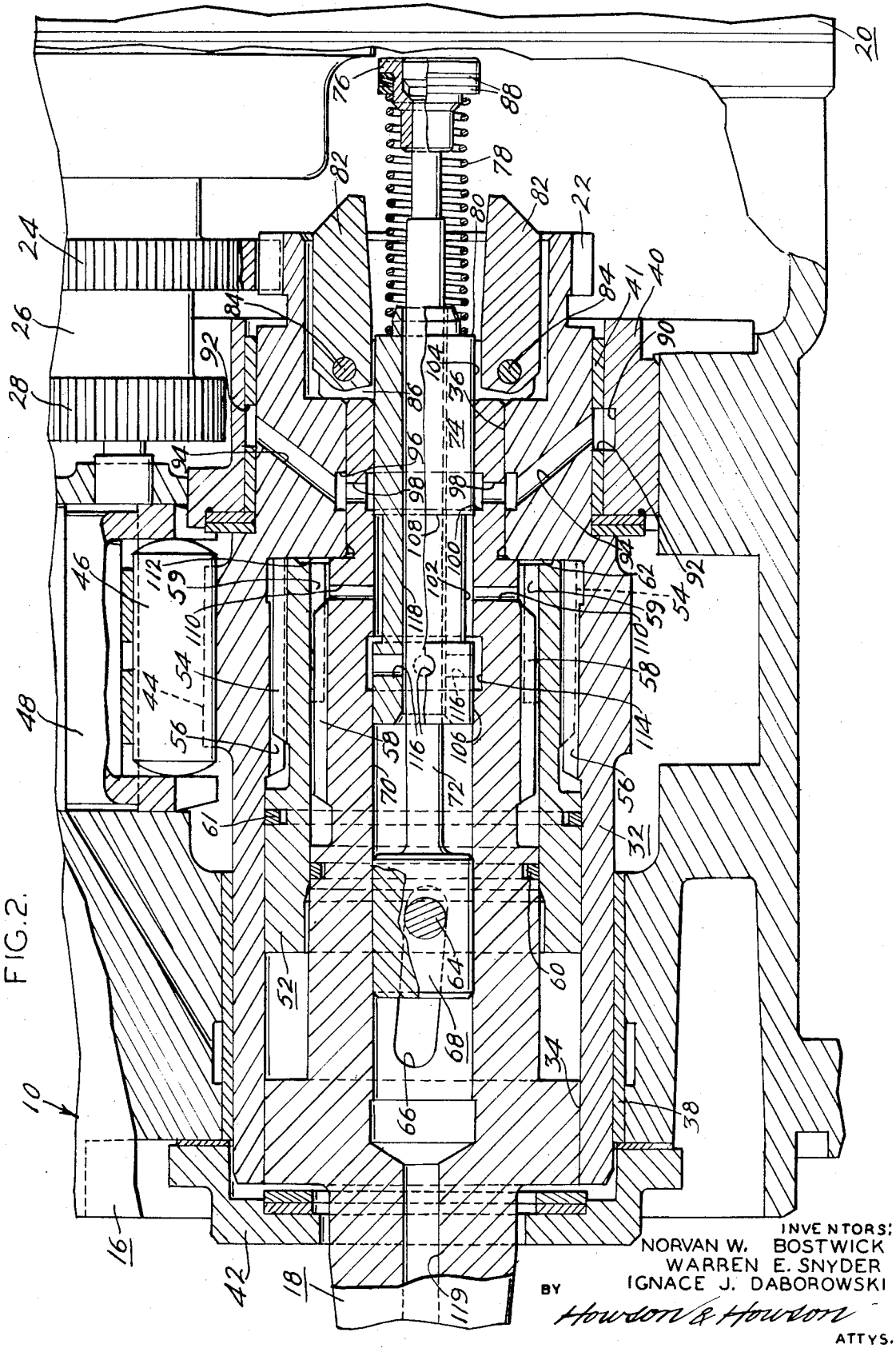


FIG. 2.

INVENTORS:
NORVAN W. BOSTWICK
WARREN E. SNYDER
IGNACE J. DABOROWSKI

BY Howson & Howson
ATTYS.

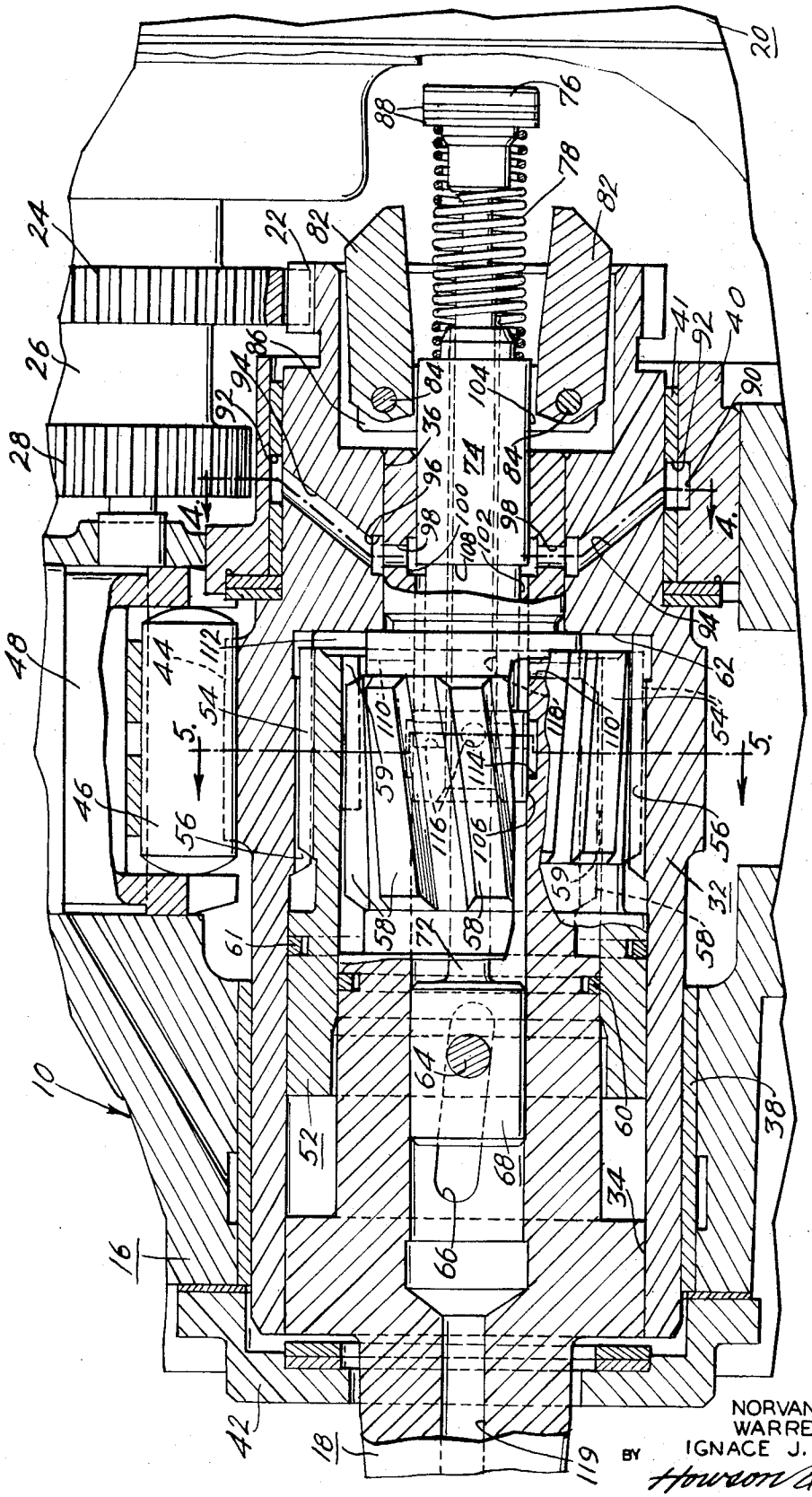


FIG. 3.

INVENTORS;
NORVAN W. BOSTWICK
WARREN E. SNYDER
IGNACE J. DABOROWSKI
BY Howson & Howson
ATTYS

FIG.5.

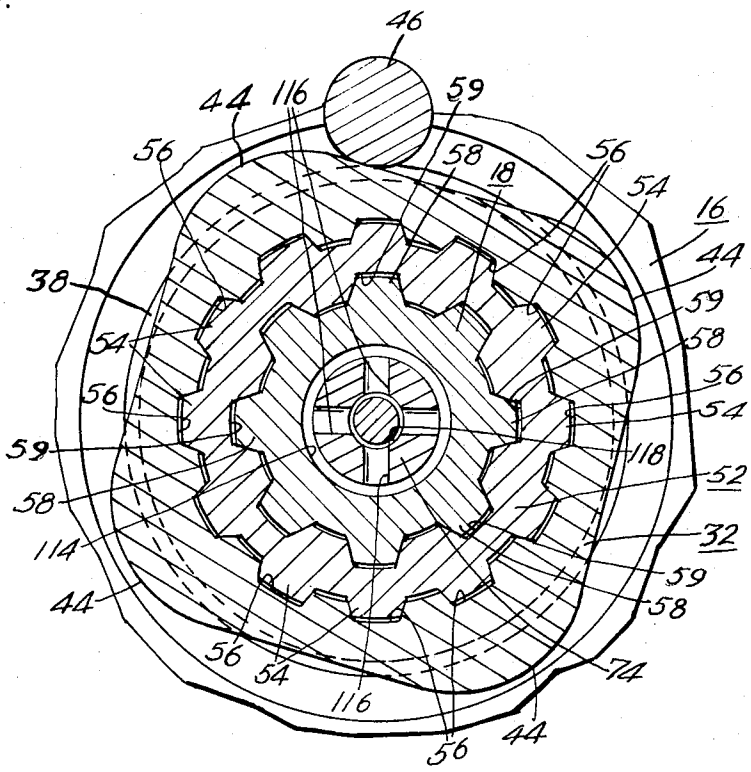
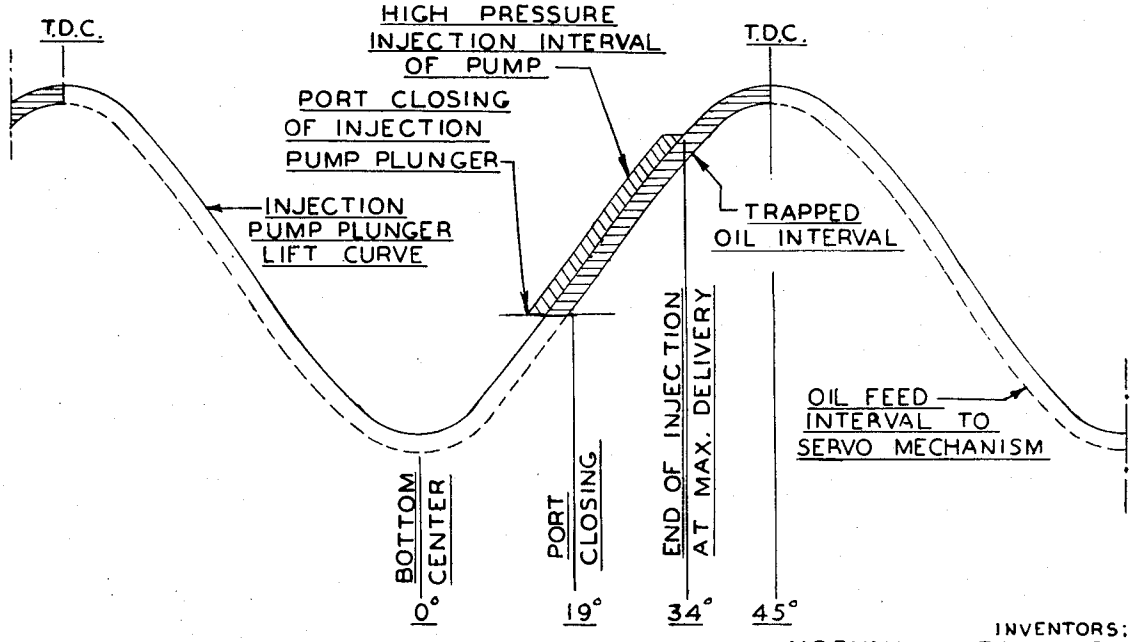


FIG.6.



INVENTORS:
NORVAN W. BOSTWICK
WARREN E. SNYDER
IGNACE J. DABOROWSKI
BY
Howson & Howson
ATTYS.

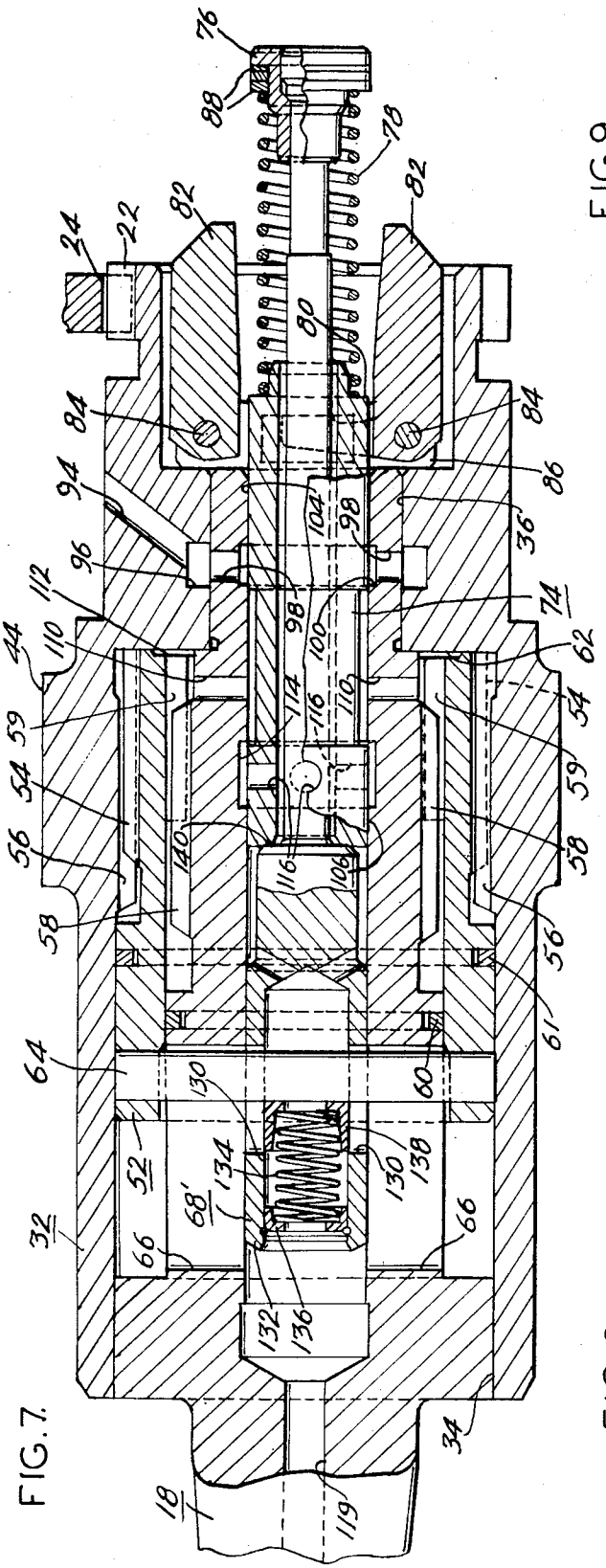


FIG. 7.

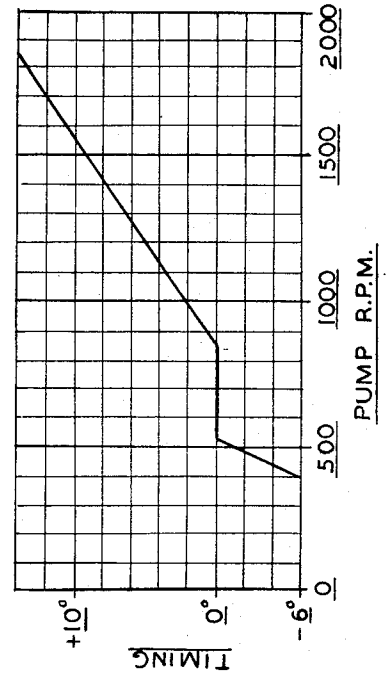


FIG. 9.

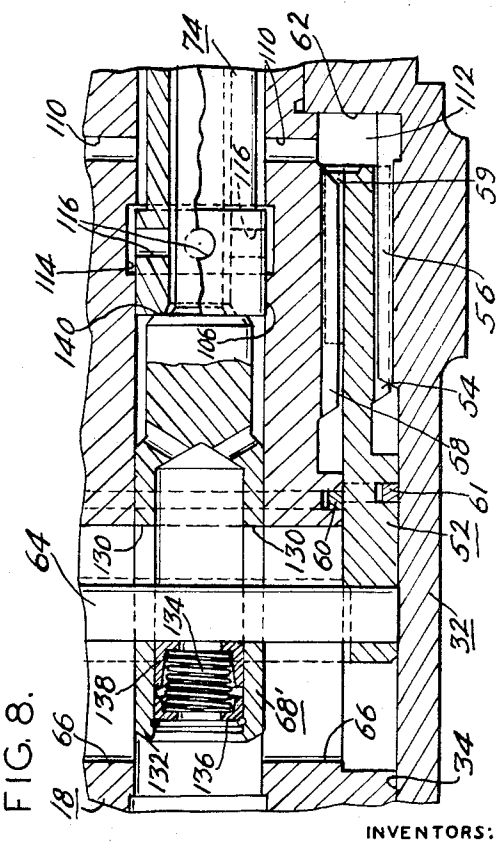


FIG. 8.

INVENTORS:
NORVAN W. BOSTWICK
WARREN E. SNYDER
IGNACE J. DABOROWSKI
BY Howson & Howson
ATTYS.

FUEL INJECTION PUMP TIMING DEVICE

The present invention relates generally to timing devices for fuel injection pumps and is particularly adapted for use as an internal timing device for single plunger, distributor type pumps.

The present invention is directed to improvements in fuel injection pump timing devices of the type shown in U. S. Pat. No. 3,050,964 (Hogemen et al.) assigned with the present application to a common assignee, which patent is hereby incorporated by reference. The present device is similar to that of the above patent and includes a centrifugal speed sensing means which by means of a servo mechanism adjusts the axial position of a splined sleeve between the pump drive shaft and plunger cam to change their angular relationship and hence the timing of the fuel injection. Important functional advantages are derived from the various structural improvements presently disclosed including the retarding of the timing during engine cranking and the prevention of cam shift during the injection interval.

To minimize the shifting of the cam during injection, the passages for the input flow of sleeve actuating fluid are formed by a plurality of slots in a non-rotating member which cooperate with an equal number of ports in a rotating member so located as to permit the passage of the high pressure actuating fluid into the sleeve chamber only between the pump injection intervals. The timing is automatically retarded for starting by providing spring means in the linkage connecting the centrifugal speed sensing means with the sleeve which will hold the sleeve in a retard position until the engine reaches a predetermined speed.

It is accordingly a first object of the present invention to provide an improved internal timing device for fuel injection pumps including means for minimizing the shifting of the pumping plunger cam during the injection interval of the plunger.

A further object of the invention is to provide a fuel injection pump timing device as described including means for automatically retarding the injection timing during cranking of the engine.

Additional objects and advantages of the invention will be more readily apparent from the following detailed description of embodiments thereof when taken together with the accompanying drawings in which:

FIG. 1 is a side elevational view of a fuel injection pump embodying the present invention with portions thereof broken away and in section to show interior details;

FIG. 2 is an enlarged sectional view of the portion of the pump of FIG. 1 shown therein in section;

FIG. 3 is a view similar to FIG. 2 but with the timing advanced in response to increased engine speed and with the helical splines of the cam and drive shaft partially shown in full;

FIG. 4 is a partial sectional view taken along line 4—4 of FIG. 3 showing the interruption of the sleeve actuating fluid passages during the injection interval;

FIG. 5 is a partial sectional view taken along line 5—5 of FIG. 3 showing the splined engagement of the sleeve with the cam and drive shaft;

FIG. 6 is a schematic phase diagram showing the timing of the servo mechanism fluid input with respect to the injection interval;

FIG. 7 is a view similar to FIG. 2 showing a modified embodiment of the invention (cam and enclosed mechanism only) including means for effecting an automatic retarding of the timing engine starting, the embodiment being shown in the starting retard position;

FIG. 8 is a partial view of the mechanism shown in FIG. 7 with the timing advanced to the 0° position; and

FIG. 9 is a graph showing the manner in which the pump timing is automatically varied by the embodiment of FIGS. 7 and 8 in accordance with pump speed to provide a timing retard during engine cranking.

Referring to the drawings and specifically FIG. 1 thereof, a fuel injection pump generally designated 10 embodying the present invention is generally similar to that disclosed in the above mentioned Hogeman patent and hence the details of the pump except for those portions with which the present invention is directly involved need not be described. The pump in brief includes a hydraulic head 12 within which a pump plunger is driven in reciprocation and rotation to sequentially inject fuel in metered amounts into the fuel outlets 14 which are connected by appropriate conduits to nozzles mounted in the engine.

The hydraulic head 12 is mounted in the pump housing 16 with the plunger thereof being driven by the pump drive shaft 18 as described below. The drive shaft 18 is coupled directly with the engine and may be driven at engine speed or a speed proportional to engine speed. A governor container within the governor housing 20 connected at one end to the pump housing 16 is driven indirectly as later described by the drive shaft 18 by means of the gear 22 which drives a similar gear 24 secured to the governor shaft 26. The engine speed is controlled by means of a linkage connected to the operating lever 23 in a conventional manner. Since the construction of the governor is conventional and nor material to the present invention, the interior details are not shown.

The rotation of the pump plunger is effected by the gear 28 on the governor shaft 26 which meshes with the face gear 30 which is rotationally secured but axially slidable on the pump plunger.

Referring to FIG. 2, the pump drive shaft 18 rotates within the cam 32, the drive shaft being supportd by the cam at bearing surfaces 34 and 36. The cam is rotatably supported within the housing 16 by bearings 38 and 40. A bearing liner 41 within the bearing 40 serves an important function to be presently described. The drive shaft 18 and cam 32 are secured and sealed within the housing by the retainer plate 42 which is secured to the housing 16 and includes appropriate seal means.

It will be noted that the gear 22 is an integral part of the cam. As described below, this feature insures a constant phase relation between the timing advance and the plunger rotation. The Hogeman structure does not coordinate the plunger rotation with timing advance, with a possible danger of hydraulic lock should be distribution channels fail to register properly during the injection interval.

The cam 32 includes a plurality of cam lobes 44 corresponding in number to the number of cylinders of the engine. The cam lobes 44 are adapted to engage the roller 46 of the tappet 48 which actuates the pump plunger in reciprocation. A return spring 50 (FIG. 1) maintains the contact of the roller 46 with the cam

lobes to insure the desired reciprocatory movement of the plunger.

The automatic timing device comprises means for varying the angular relationship of the drive shaft **18** and cam **32** automatically in accordance with the speed at which the pump is driven. For the most effective engine performance, it is necessary to advance the injection timing at higher speeds and to retard the timing during starting of the engine. This is accomplished in the present invention as in the above mentioned Hogeman patent by a centrifugal speed sensing device which controls a hydraulic servo mechanism. The servo mechanism actuates a sleeve **52** disposed between the cam **32** and drive shaft **18**. The sleeve includes external helical splines **54** for cooperative engagement with similar interior splines **56** of the cam, and internal helical splines **59** which cooperate with the splines **58** of the drive shaft. Seal rings **60** and **61** axially spaced from the respective splines prevent servo fluid passage between the sleeve and the cam, or the sleeve and the drive shaft. A predetermined axial movement of the sleeve **52** will produce a change in the rotational angular relationship of the cam and the drive shaft. In the position shown in FIG. **2**, the sleeve **52** is in the fully retarded position abutting the shoulder **62** of the cam. Since the range of timing adjustment might typically be 20°, this position is termed the position of 0° advance.

A pin **64** is radially secured to the sleeve **52** and passes through a helical slot **66** in the drive shaft **18** into the rod **68** which is slidably disposed in the axial bore **70** of the drive shaft **18**. The position of the rod **68** will accordingly correspond to the position of the sleeve **52**. A stem portion **72** of the rod **68** extends axially through a hollow valve member **74**, also slidably disposed in the bore **70**, and terminates in a spring seat **76** at its opposite end. A coil spring **78** extends in compression between the spring seat **76** and the end **80** of the valve member **74**. A plurality of weights **82** are pivotally mounted at **84** to a spider **86** extending from the valve element **74** in such a manner as to cause a movement of the valve element toward the right as viewed in FIG. **2** upon the centrifugally effected outward movement of the weights. Such weight induced movement of the valve **74** is opposed by the compression spring **78**, the spring rate of which is appropriately chosen to provide the required movement of the valve member with increasing speed of the pump. Shims **88** are applied to the spring seat **76** to create an initial spring force which must be overcome by the weight force before radial movement of the weights and hence movement of the valve member can occur.

The position of the valve member **74** controls the flow of high pressure sleeve actuating fluid, which in the present instance comprises oil from a source such as the engine lubrication system. The high pressure oil enters through ports **89** in the bearing **40** into an annulus **90** which distributes oil to the slots **92** in the bearing liner **41**. A plurality of ports **94** in the cam connect the slots **92** with an annulus **96** in the bearing surface **36** adjacent the drive shaft **18**. A plurality of equally spaced ports **98** in the drive shaft of a number equal to the number of lobes on the cam **52** extend radially to an annulus **100** opening into bore **70**. As discussed in further detail below, the slots **92** in the bearing liner **41** and the ports **94** in the cam are cooperatively arranged so that the slots and ports are not in communication during the injection interval of the pump plunger thereby preventing any shifting of the cam with respect to the drive shaft.

The valve member **74** is characterized by a region **102** of reduced diameter which divides the member into two spaced lands **104** and **106**. The position of shoulder **108** of the land **104** with respect to the annulus **100** of the drive shaft controls the flow of high pressure actuating oil into the region **102** of the valve. A plurality of radial ports **110** in the drive shaft connect the chamber formed by the drive shaft bore **70** and region **102** of the valve with the chamber **112** within which the splined sleeve **52** is disposed. Accordingly, when the land **104** of the valve member **74** is so positioned as to permit communication of the annulus **100** with the region **102** of the valve member, high pressure fluid will flow through the region **102** and ports **110** into the chamber **112** whereupon the sleeve **52** will be moved to the left as viewed in FIG. **2** as the fluid pressure overcomes the force of the spring **78**.

An annulus **114** in the drive shaft is spaced from the annulus **100** exactly the same distance as the spacing of the valve member lands **104** and **106**. A plurality of radial ports **116** in the land **106** permit oil to pass from the annulus **114** through the valve member into the hollow bore **118** thereof which it will be noted is of a diameter substantially larger than the diameter of the stem portion **72** of the rod **68**, thus permitting the exhausting of the oil into the sump formed by the pump and governor housings. Oil may also drain past the rod **68** into drain bore **119** in the driven end of the drive shaft.

In operation, the timing device functions in a manner similar to that described in the above-cited Hogeman patent. With the pump drive shaft **18** connected with the engine to rotate at a speed equal or proportional to engine speed, the pump plunger will deliver metered quantities of fuel to the injection nozzles in accordance with the setting of the operating lever **23** which, acting through the governor, regulates the engine speed. At idling and relatively low speeds, the phase relationship of the drive shaft **18** with the cam **32** is in the normal 0°0 advance position shown in FIG. **2** wherein the sleeve **52** abuts the shoulder **62** of the cam. At such idling or low speeds, the weights **82** do not develop sufficient centrifugal force to overcome the opposing force of the spring **78** and the valve member **74** is accordingly disposed at its extreme leftward or closed position as in FIG. **2**. In this position, the land **104** covers the annulus **100** thereby preventing high pressure fluid from reaching the sleeve chamber **112**. At the same time, the land **106** is so positioned as to permit communication between the region **102** and the annulus **114** thereby permitting fluid to drain from the sleeve chamber **112** through ports **116**.

As the speed of the engine and hence the pump is increased, the weights **82** overcome the force of spring **78** and in swinging outwardly move the valve member **74** to the right closing off the annulus **114** and at the same time opening the annulus **100** permitting a flow of high pressure fluid to enter the sleeve chamber **112** through the region **102** and ports **110**. The pressurization of the chamber **112** moves the sleeve **52** to the left as shown in FIG. **3**. The leftward movement of the

sleeve is substantially stabilized at a position dependent on the speed of operation of the engine by the linkage of the sleeve, pin **64** and the rod **68**. Thus as the sleeve moves to the left, the rod **68** also moves to the left, further compressing the spring **78** and counteracting the centrifugally developed force of the weights **82**. This movement of the rod **68** to the left serves to move the valve member **74** to the left thereby closing the annulus **100**. For a given increased speed, the sleeve **52** will thus move to a predetermined position and the valve member **74** will become relatively stabilized as the spring force **78** and centrifugal force of the weights become balanced. By virtue of the helical splines on the sleeve **52** and the corresponding splines of the drive shaft and cam, the change in the timing relationship of the drive shaft and cam is automatically regulated in accordance with the speed of rotation of the pump.

Upon a further increase in the speed of engine operation, the weights will again move the valve member to admit fluid to the sleeve chamber **112** and move the sleeve to the left to establish a new advanced timing relation of the drive shaft and cam. Conversely, should engine speed decrease, the inwardly collapsing weights will allow the valve member **74** to move to the left, opening the main annulus **114** and draining the fluid in the chamber **112** to the sump, thus permitting movement of the sleeve to the right under the influence of the spring **78** acting through the rod **68** and pin **64**. When the spring and weight forces are balanced, the valve member **74** will again reach a relatively stabilized position although a slight oscillation of the valve takes place even during constant speed operation.

Because of the greatly increased force transmitted through the cam during the plunger injection interval, there is a tendency for the cam to shift on the helical splines with respect to the drive shaft during the injection interval, especially during periods when the annuli **100** and **114** are in communication with the chamber **112**. The tendency is for the sleeve **52** to be shifted to the right under the increased resistance during the injection interval and accordingly to produce a less advanced timing relationship than is required for optimum efficiency at a particular speed. Such a shifting is not as detrimental during the opening of the drain annulus **114** since this annulus is opened only during a retarding phase of operation which is hastened by the shifting tendency. However, during the opening of the annulus **100**, such a shifting tendency runs counter to the timing advance being carried out.

By the present invention, such a shifting tendency is minimized during the time the valve member **74** is positioned to permit flow from the annulus **100**. The structure provided to minimize the shifting comprises the slots **92** in the bearing liner **41** and their cooperative relationship with ports **94** of the cam. With reference to FIGS. **4** and **6**, it will be observed that the ports **94** do not communicate with the slots **92** during the high pressure injection interval of the pump due to the selection of the length and angular relation of the slots with respect to the ports **94** and the cam lobes **44**. There is accordingly a positive trapping of the sleeve actuating fluid in the sleeve chamber **112** during the injection interval. The fluid cannot escape past the sleeve due to the seal rings **60** and **61**. In the view of FIG. **4**, the ports **94** have just rotated out of communication with the slots **92** as the plunger has advanced to approximately 20° from bottom center. As shown in FIG. **6**, the port closing occurs approximately 19° after bottom center at the same time as the port closing of the injection pump plunger. As indicated, the end of injection at maximum delivery occurs at approximately 34° and the ports **94** again communicate with the slots **92** at approximately top center which is 45° of shaft rotation from bottom center with a four lobe cam configuration.

The provision of the gear **22** on the cam rather than on the drive shaft as in the Hogeman U.S. Pat. No. 3,050,964 maintains a constant phase relation of the plunger reciprocation and rotation. Thus, as the timing is advanced or retarded by cam rotation, there is a corresponding rotation shift of the plunger. This allows at least theoretically an unlimited timing advance without danger of hydraulic lock.

A modified embodiment of the invention is shown in FIGS. **7** and **8** which includes essentially the same timing device described above with the addition of an arrangement for permitting a retarding of the timing below the 0° timing position during starting and low idling of the engine. Since the structure providing this function is contained wholly within the cam, only the cam and interior details of the modified embodiment are shown in FIGS. **7** and **8**. The cam, drive shaft, sleeve, valve member and flyweight assembly are identical with the counterparts of the timing device described above and accordingly bear the same identifying numbers as do other identical elements.

Referring to FIG. **7**, the rod **68'** is at its left end of a different configuration than the rod **68** of the above described embodiment. The pin **64** secured to the sleeve **52** passes through an axial slot **130** in the hollow end portion **132** of the rod **68'**. A light spring **134** is mounted in compression in the hollow end **132** of the rod between a spring seat **136** and a piston **138** slidably disposed within the hollow bore of the rod and bearing against the pin **64**. The spring **134** will accordingly urge the rod **64** and hence the sleeve **52** toward a retard position. With the engine stopped as shown in FIG. **7**, the spring **134** positions the sleeve **52** against the shoulder **62** of the cam, which in this embodiment is a timing position of −6° below the normal 0° advance position as illustrated in the graph of FIG. **9**. A shoulder portion **140** of the rod **68'** engages the left hand end of the valve member **74** in the stopped or idling position of the pump, and the spring **78** and shims **88** are selected so that the rod **68'** remains with the shoulder **140** abutting the valve member until the pump reaches a speed of approximately 850 rpm, at which point the centrifugal force of the weights is sufficient to overcome the force of the compressed spring and to move the valve member out of engagement with the rod shoulder.

At speeds below 400 rpm, the force of spring **134** is greater than the centrifugal force of the weights **82** and the rod **64** and sleeve **52** will accordingly move to the fully retarded position shown in FIG. **7**. At speeds between approximately 400 and 520 rpm, the centrifugal force of the weights is sufficient to compress the spring **134** and move the rod **68'** and the valve member **74** to the right to open the annulus **100** and permit a fluid flow into the sleeve chamber **112**, thereby advancing the timing as high as the 0° advance position illus-

trated in FIG. 8. At speeds between approximately 520 and 850 rpm, the sleeve remains in the position shown in FIG. 8 with the spring 134 fully compressed. Since the valve member 74 and rod 68' remain in abutting relationship in this speed range, there is no change in the timing relationship of the drive shaft and cam which remains at 0°. At speeds above 850 rpm, the timing advance is in direct proportion to increase speed as shown in FIG. 9.

While the several embodiments of the invention have been described in the setting of a single plunger distributor type pump, it will be apparent that the invention may also be used with other types of pumps wherein the angular relation of a plunger cam with respect to the cam drive shaft is selectively controlled.

Manifestly, changes in details of construction can be effected by those skilled in the art without departing from the spirit and the scope of the invention.

We claim:

1. In a fuel injection pump including a drive shaft, a pump plunger, a pump plunger actuating cam rotatably driven by said drive shaft, said cam driving said plunger radially away from said drive shaft during a portion of the cam rotation including the plunger injection interval, timing means for varying the relative angular relation of said drive shaft and cam, said timing means including a sleeve disposed between and operatively connected to said cam and said drive shaft, and hydraulic means for actuating said sleeve to vary the relation of said drive shaft and cam, said hydraulic means including an input conduit for sleeve actuating fluid, the improvement comprising means for preventing the shifting of the cam with respect to the shaft during the plunger injection interval, said latter means comprising means for closing the sleeve actuating fluid input conduit during the injection interval.

2. The invention as claimed in claim 1 wherein said means for preventing the shifting of the cam comprising seal means associated with said sleeve for preventing escape of hydraulic fluid along said sleeve during the injection interval.

3. The invention as claimed in claim 2 wherein said sleeve actuating fluid input flow path passes through adjacent non-rotating and rotating elements of said pump, said means for closing the sleeve actuating fluid input flow path comprising slots in said non-rotating element communicating with a source of pressurized fluid and ports in said rotating element adapted to communicate with said slots, the circumferential length and position of said slots and the position of said ports being chosen so as to prevent communication of said slots and ports during the plunger injection interval.

4. In a fuel injection pump including a drive shaft, a pump plunger, a pump plunger actuating cam rotatably driven by said drive shaft, said cam being concentrically disposed in overlying rotatable relation with said drive shaft, said cam driving said plunger radially away from said drive shaft during a portion of the cam rotation including the plunger injection interval, timing means for automatically varying the relative angular relation of said drive shaft and cam, said timing means including a sleeve disposed in a chamber between said cam and drive shaft and operatively connected by helical splines to said cam and drive shaft, hydraulic means including conduit means for introducing pressurized fluid into said sleeve chamber to effect axial movement of said sleeve and a resultant variation of the timing relation of said drive shaft and cam, and speed sensing means operatively connected with said hydraulic means to effect automatic control of said hydraulic means, the improvement comprising means for preventing the shifting of the cam with respect to the drive shaft during the plunger injection interval, said latter means comprising means for closing said sleeve actuating fluid conduit means during the injection interval, said closing means comprising slots in a non-rotating element of said pump and ports in an adjacent rotating element, said slots and ports formed a part of said conduit means, said slots and ports being in communication during part of each revolution of the rotating member, said slots and ports being out of communication during the plunger injection interval.

5. The invention as claimed in claim 4 wherein said slots are disposed in a bearing member rotatably supporting said cam, and wherein said ports are disposed in said cam.

6. The invention as claimed in claim 4 wherein said means for preventing the shifting of the cam comprises seal rings associated with said sleeve for preventing escape of sleeve actuating fluid from said sleeve chamber during the injection interval.

7. A fuel injection pump timing device comprising a drive shaft, a pump plunger actuating cam rotatably driven by said drive shaft, said cam being concentrically disposed in overlying rotatable relation with said drive shaft and operatively connected by helical splines to said cam and drive shaft, hydraulic means including conduit means for introducing pressurized fluid into said sleeve chamber to effect axial movement of said sleeve and a resultant variation of the timing relation of said drive shaft and cam, said hydraulic means including a valve slidably disposed in a coaxial bore within said drive shaft, centrifugal speed sensing means operatively connected with said valve to effect automatic control of said hydraulic means, a rod coaxial with said drive shaft and disposed within a concentric bore in said drive shaft and passing through a concentric bore of said valve, a first spring means extending between said valve and said rod opposing the centrifugal movement of said speed sensing means, and a second spring means associated with said rod adapted to urge said sleeve toward a timing retard position when said pump speed falls below a predetermined level.

8. The invention claimed in claim 7 including a radial pin secured to said sleeve and passing through an axial slot in said rod, said second spring means being connected with said rod to urge said pin and hence said sleeve against the force of fluid pressure in said sleeve chamber.

9. The invention claimed in claim 8 including means for preventing relative movement of said rod and valve during pump operation below a predetermined speed.

10. The invention claimed in claim 7 including means for closing said conduit means for introducing pressurized fluid into said sleeve chamber during the injection of the pump plunger.

11. The invention as claimed in claim 10 including seal means associated with said sleeve for preventing escape of pressurized fluid along said sleeve during the injection interval.

* * * * *

UNITED STATES PATENT OFFICE

CERTIFICATE OF CORRECTION

Patent No. 3,726,608 Dated April 10, 1973

Inventor(s) Norvan W. Bostwick et al

It is certified that error appears in the above-identified patent and that said Letters Patent are hereby corrected as shown below:

Column 2, line 30, change "container" to --contained--;

line 38, change "nor" to --not--;

line 59, after "be" insert --the--.

Claim 7, line 31 after "drive shaft" insert -- , a sleeve disposed in a chamber between said cam and drive shaft--.

Claim 10, line 61 after "injection" insert --interval--.

Signed and sealed this 20th day of November 1973.

(SEAL)
Attest:

EDWARD M.FLETCHER,JR.
Attesting Officer

RENE D. TEGTMEYER
Acting Commissioner of Patents

FORM PO-1050 (10-69)

USCOMM-DC 60376-P69
* U.S. GOVERNMENT PRINTING OFFICE : 1969 O—366-334.