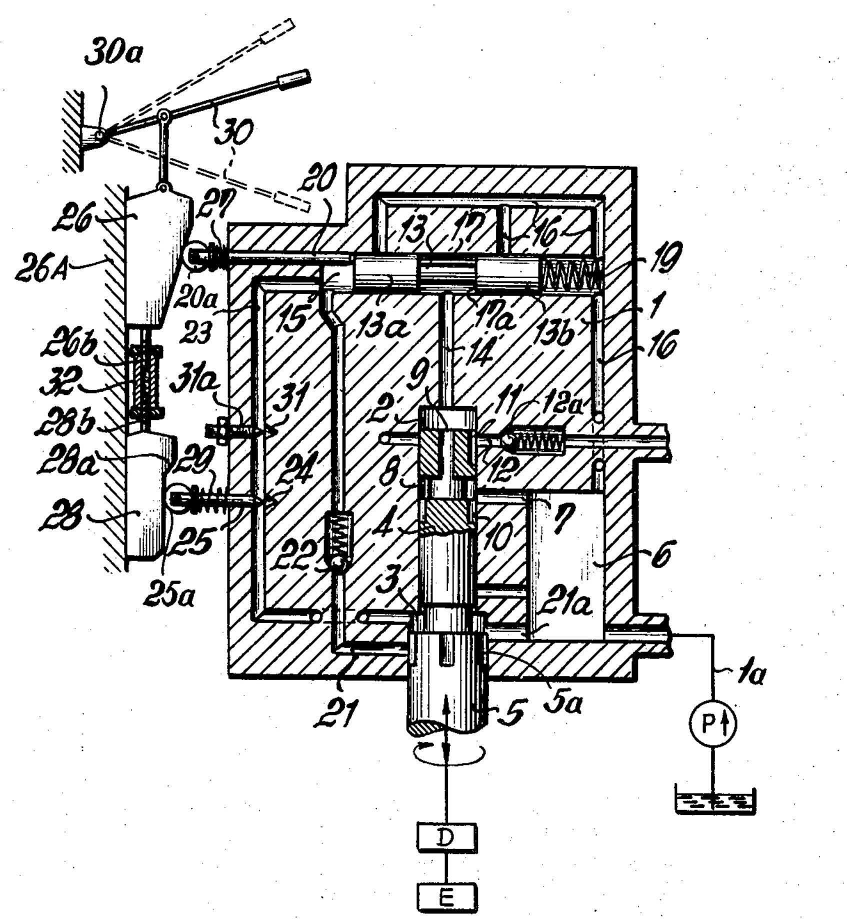
Dec. 24, 1968 | K. ECKERT ET AL | 3,417,703

FUEL INJECTION PUMP

Filed Oct. 21, 1966 | 5 Sheets-Sheet 1

FIG. 1



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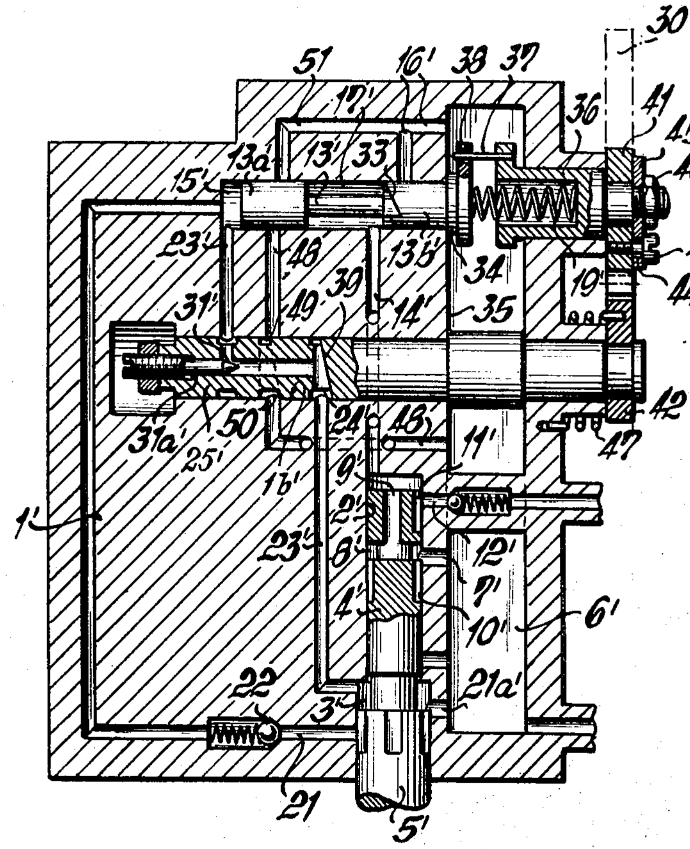
FUEL INJECTION PUMP

Filed Oct. 21, 1966 5 Sheets-Sheet 2

FIG.2



←III

FIG.3

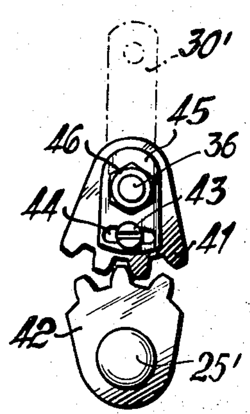


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FIG. 4

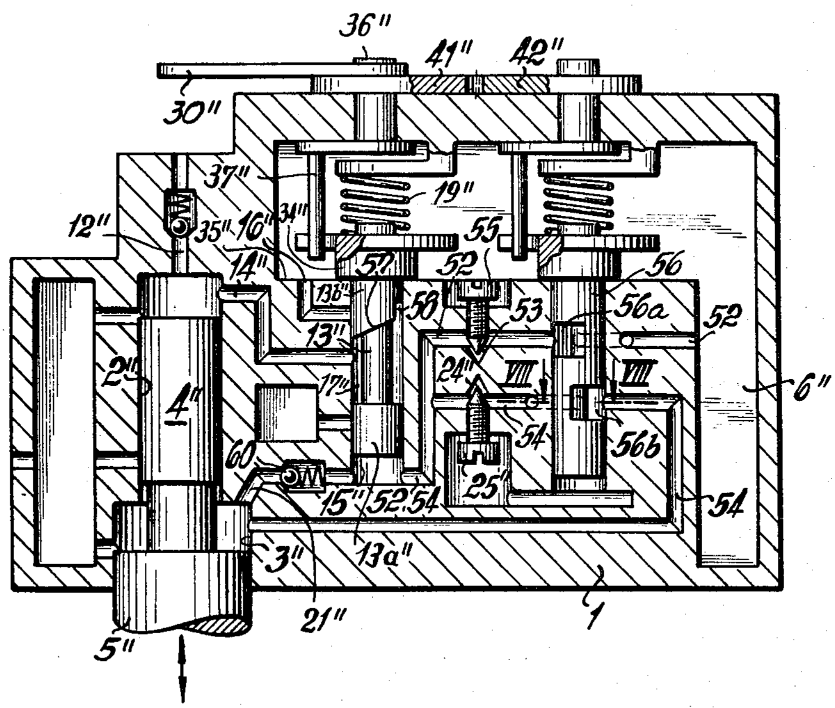
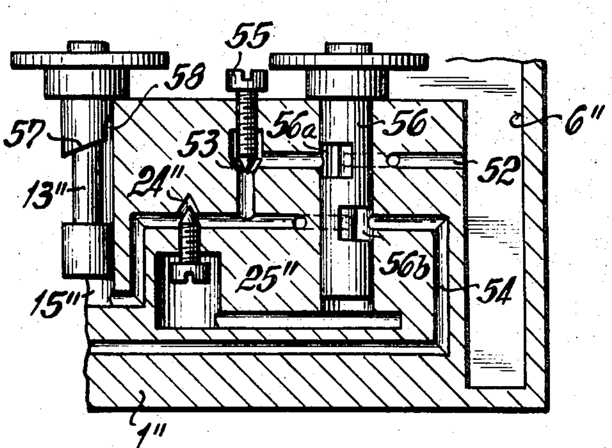


FIG. 5



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FUEL INJECTION PUMP

Filed Oct. 21, 1966 5 Sheets-Sheet 4

FIG. 6

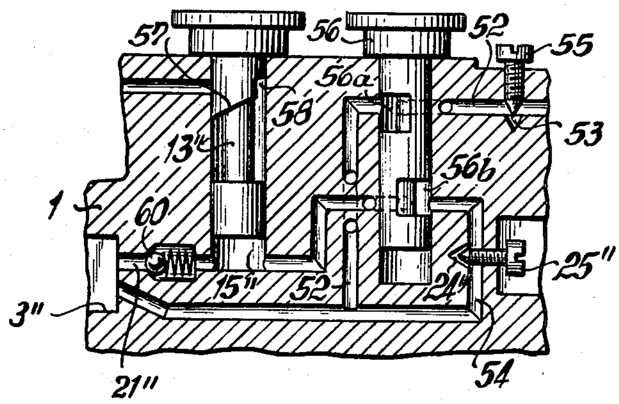

FIG. 7

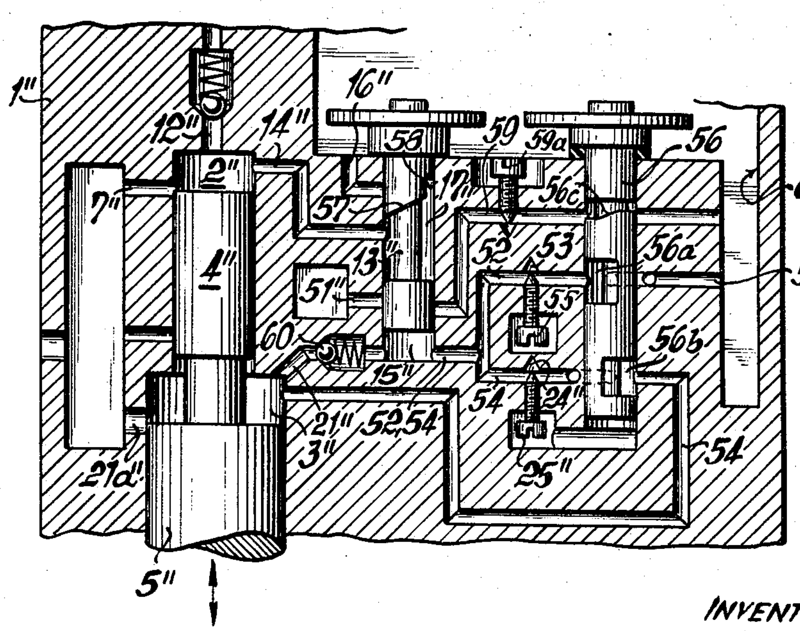

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FIG. 9

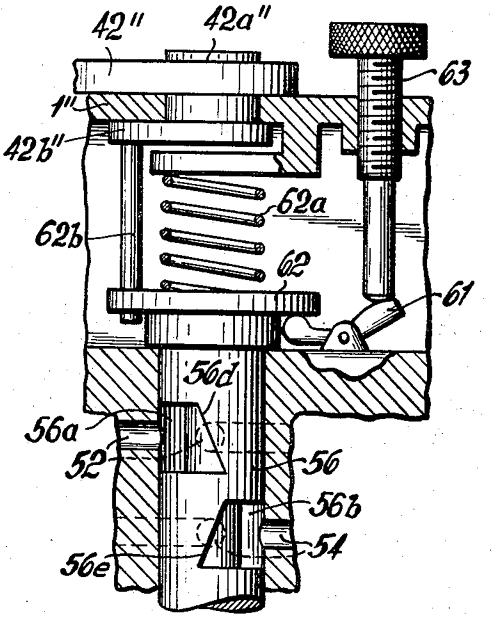


FIG. 8

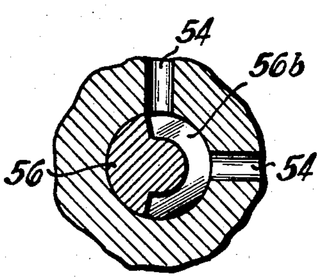


FIG. 10

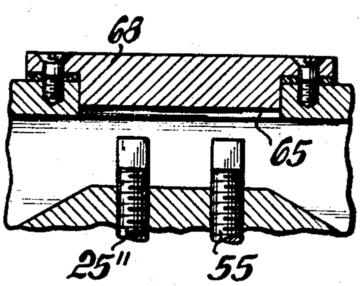
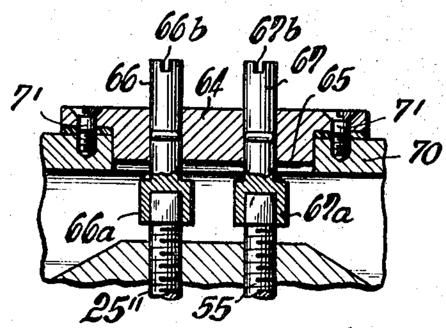


FIG. 11



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United States Patent Office

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FUEL INJECTION PUMP

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Claims priority, application France, Oct. 21, 1965, 35,803
18 Claims. (Cl. 103—41)

ABSTRACT OF THE DISCLOSURE

A fuel injection pump for internal combustion engines wherein the amounts of fuel admitted to the cylinders of the engine can be varied infinitely by means of a shuttle piston movable from a starting position to an open position in which latter position fuel pumped by the pump is discharged through an overflow channel so that injection of fuel to the cylinders of the engine is terminated and in which a return movement of the shuttle piston to the starting position under the force of a return spring is regulated by adjustable throttling means controlling outflow of fuel from the chamber in which the shuttle piston reciprocates. The pump includes further first adjusting means for regulating the distance between the starting and open position of the shuttle piston, second adjusting means for regulating the braking action of the throttling means, and common actuating means for the two adjusting means.

The present invention relates to injection pumps, and more particularly to improvements in fuel injection pumps of the type disclosed in Patent No. 3,122,100 granted to Bessiere on Feb. 25, 1964.

The patent to Bessiere discloses a self-regulating fuel injection pump wherein the outflow of fuel from a working cylinder is controlled by a regulating valve, also called shuttle piston, which is provided with an inclined face and is rotatable in its chamber to thereby establish or terminate a connection between two sections of an overflow channel. The operation of the pump is regulated automatically in each speed and load range of the engine whose cylinders receive fuel from the working cylinder. However, it is often desirable that the operator of the vehicle which is driven by an engine receiving fuel from an injection pump should be in a position to arbitrarily select the amounts of fuel which are injected into the cylinders of the engine, especially to arbitrarily select such amounts when the engine is not idling or when the engine is not driven at a maximum speed. The operation is smoother if the driver of the vehicle is in a position to determine the amounts of fuel in each but the two extreme conditions of operation.

Accordingly, it is an important object of the present invention to provide a novel and improved injection pump for liquid fuel and to construct and assemble the pump in such a way that its operation will be fully automatic when the engine receiving fuel therefrom is idling or operates at a maximum permissible speed but that the operator will be in a position to determine the amounts of injected fuel in the entire intermediate range of operation.

Another object of the invention is to provide a fuel injection pump of the just outlined character wherein all such adjustments which are necessary in actual operation can be carried out by means of a single actuating member.

A further object of the invention is to provide a fuel injection pump wherein the amounts of fuel admitted to the cylinders of an internal combustion engine can be varied infinitely so that such amounts can be selected with utmost accuracy by full consideration of momentary operating conditions.

An additional object of the invention is to provide a novel system of adjustable throttling elements which can be utilized in our fuel injection pump.

Briefly stated, one feature of our present invention resides in the provision of a fuel injection pump for internal combustion engines, particularly for internal combustion engines of automotive vehicles. The pump comprises a main cylinder or working cylinder, a main piston or working piston reciprocably received in the main cylinder to respectively draw and expel fuel therefrom, drive means for reciprocating the main piston at a speed which is a function of engine speed, fuel line means arranged to receive fuel from the main cylinder for injection into the cylinder or cylinders of the engine, overflow channel means connected with the main cylinder and including a regulating chamber, an adjustable regulating valve or shuttle piston reciprocable in the regulating chamber and movable from at least one starting position to at least one open position in which latter position the overflow channel means is effective to evacuate fuel from the main cylinder instead of the fuel line means so that the injection of fuel into the cylinder or cylinders of the engine is terminated on movement of the regulating valve to open position, biasing means for urging the regulating valve to starting position, auxiliary pump means operating in synchronism with the main piston to deliver fuel to the regulating chamber for moving the regulating valve from starting position while the main piston performs a working stroke, a discharge line connected with the regulating chamber to evacuate fuel during movement of the regulating piston under the action of biasing means back toward starting position while the main piston performs a return stroke, adjustable throttling means for braking the flow of fuel through the discharge line during return movement of the regulating piston toward starting position, first adjusting means for regulating the distance between the starting and open positions of the regulating valve, second adjusting means for regulating the braking action of the throttling means, and common actuating means for the two adjusting means.

The operative connection between the two adjusting means preferably comprises an adjustable coupling which is arranged to maintain the second adjusting means in at least one first position corresponding to minimal braking action of the throttling means when the first adjusting means is set for operation of the engine at full load or higher-partial-load, and to maintain the second adjusting means in second positions corresponding to increasing braking action of the throttling means as a function of changes in load when the first adjusting means is set for operation of the engine at lower-partial-load and during idling. In the first position of the second adjusting means, the braking action of the throttling means is too low or too ineffective to prevent return movement of the regulating valve to starting position while the engine operates at less than a preselected maximum speed (i.e., while the main piston reciprocates at less than a predetermined maximum speed) so that the first actuating means can select the amounts of fuel which are supplied to the fuel line means in the entire full-load and higher-partial-load range of engine operation while the engine speed remains below the predetermined maximum speed.

The fuel injection pump preferably comprises secondary throttling means adjustable independently of the first mentioned throttling means to determine the rate of fuel flow in the discharge line in the first position of the second adjusting means by bringing about a braking action which exceeds the braking action of the first mentioned throttling means when the engine operates at full load or at higher-partial-load.

The novel features which are considered as characteristic of the invention are set forth in particular in the appended claims. The improved fuel injection pump itself, however, both as to its construction and its mode of operation, together with additional features and advantages thereof, will be best understood upon perusal of the following detailed description of certain specific embodiments with reference to the accompanying drawings, in which:

FIG. 1 is a diagrammatic sectional view of a fuel injection pump which embodies one form of our invention;

FIG. 2 is a similar diagrammatic sectional view of a second fuel injection pump;

FIG. 3 is a fragmentary front elevational view of a coupling in the second pump as viewed in the direction of arrow III in FIG. 2;

FIG. 4 is a diagrammatic sectional view of a third fuel injection pump;

FIG. 5 is a fragmentary diagrammatic sectional view of a fourth pump which constitutes a modification of the pump shown in FIG. 4;

FIG. 6 is a similar fragmentary sectional view of a fifth pump which also constitutes a slight modification of the pump shown in FIG. 4;

FIG. 7 is a fragmentary diagrammatic sectional view of a sixth fuel injection pump which is similar to the pumps of FIGS. 4 to 6;

FIG. 8 is an enlarged fragmentary horizontal section as viewed in the direction of arrows from the line VIII—VIII of FIG. 4;

FIG. 9 is a fragmentary diagrammatic sectional view of a seventh fuel injection pump which constitutes a modification of the pumps shown in FIGS. 4 to 7;

FIG. 10 is a fragmentary section through the body of a fuel injection pump and illustrates a lid serving to normally seal the opening which affords access to the throttles; and

FIG. 11 is a similar fragmentary section through the pump body and illustrates a device which can be used to preset two needle-like throttles.

Referring first to FIG. 1, there is shown a fuel injection pump which comprises a pump body or housing **1** including a working cylinder or main cylinder **2** and an auxiliary cylinder **3**. The two cylinders are coaxial and respectively receive a main piston or working piston **4** and an auxiliary piston **5**. The pistons **4**, **5** together form a multi-stage piston which is rotated and reciprocated by a drive D of the internal combustion engine E whose cylinders receive fuel from the working cylinder **2** via fuel lines **12** only one of which is actually shown in FIG. 1. The piston **5** and its cylinder **3** together constitute an auxiliary pump whose purpose is to shift a reciprocable regulating valve or shuttle piston **13** in a regulating chamber **15**. The diameter of the auxiliary piston **5** extends the diameter of the working piston **4**, and these pistons are rotated and reciprocated at a speed which is a function of the rotational speed of the engine E. The drive D which transmits motion from the engine E to the piston **5** may be constructed in a manner as disclosed, for example, in U.S. Patent No. 3,058,455 to Höfer et al., granted Oct. 16, 1962, which is assigned to the same assignee. A drive including a cam fixed to the piston **5** and rotated by the engine, stationary roller followers which engage the face of the cam, and a spring which biases the cam against the followers is preferred at this time. Reciprocatory movements of the pistons **4**, **5** are used to respectively suck and expel fuel from the cylinders **2**, **3**, and rotary movements of the piston **4** are utilized to distribute measured amounts of fuel to successive fuel lines **12**. Each such fuel line receives a measured amount of fuel once during each revolution of the piston **4**.

The pump body **1** defines a suction space **6** which is connected with a fuel pump P by a pipe **1***a*. A supply bore **7** can connect the suction space **6** with a circumferential groove **8** of the working piston **4**, and the latter comprises an axially extending bore **9** which connects the groove **8** with the interior of the working cylinder **2**. The position of the supply bore **7** is selected in such a way that its discharge end is sealed by the piston **4** when the latter performs a working (upward) stroke; however, when the piston **4** moves downwardly (suction stroke), the space **6** can communicate with the groove **8** through one of several axially parallel peripheral grooves **10** of the working piston so that the latter can suck fuel into the interior of the working cylinder **2**. The working piston **4** is provided with an axially parallel peripheral distributor groove **11** which connects the interior of the working cylinder **2** with one of the fuel lines **12** during each working stroke of the piston **4**. Each fuel line **12** contains a one-way valve **12***a* which prevents return flow of fuel. The injection nozzles (not shown) at the ends of the fuel lines **12** admit fuel into the respective cylinders of the engine E. The intake ends of all fuel lines **12** are located in a common plane which is normal to the axis of the working piston **4**, and such intake ends are equidistant from each other. As stated before, one fuel line **12** will communicate with the distributor groove **11** during each working stroke of the piston **4** and the number of such working strokes per revolution of the piston **4** corresponds to the number of cylinders in the engine E, i.e., to the number of fuel lines **12**.

The aforementioned regulating chamber **15** forms part of an overflow channel which further comprises a bore **14** connecting the chamber **15** with the interior of the cylinder **2** and a bore **16** connecting the chamber **15** with the suction space **6**. The valve **13** comprises two cylindrical sections or plungers **13***a*, **13***b* which are slidably but sealingly fitted into the chamber **15** and define between themselves an annular groove or recess **17** adapted to establish a connection between the bores **14** and **16**. The annular end face **17***a* of the plunger **13***b* will permit partial or full communication between the chamber **15** and bore **16**, depending on the axial position of the valve **13** which is permanently biased by a helical return spring **19** tending to maintain the plunger **13***a* in abutment with an axially reciprocable adjusting pin **20**. This pin will determine the starting or initial position of the valve **13** and forms part of a first adjusting unit which further comprises a helical expansion spring **27**, a roller follower **20***a* at the outer end of the pin **20**, and a reciprocable cam **26** which is guided in stationary ways **26A** for movement in directions at right angles to the axis of the pin **20**. When the plunger **13***a* abuts against the inner end of the adjusting pin **20**, the plunger **13***b* seals the annular groove **17** from the bore **16** so that the latter is sealed from the bore **14** and from the interior of the working cylinder **2**. In order to permit escape of fuel from the cylinder **2** into the suction space **6**, the valve **13** must be shifted in a direction to the right until the end face **17**_a_ of the plunger **13**_b_ permits at least some communication between the annular groove **17** and bore **16**.

The function of the auxiliary pump including the piston **5** and cylinder **3** is to shift the valve **13** axially against the opposition of the return spring **19** so that the plunger **13**_b_ will be in a position to permit flow of fuel from the annular groove **17** into the bore **16**. This automatically interrupts the admission of fuel to that fuel line **12** which is in registry with the distributor groove **11** because the resistance offered by one-way valves **12**_a_ and the injection nozzles (not shown) to the flow of fuel through such fuel lines exceeds the resistance offered by the overflow channel **14**, **15**, **16** to the flow of fuel from the interior of the working cylinder **2** back into the suction space **6**. In other words, whenever the valve **13** establishes a path for flow of fuel via overflow channel **14–16**, the delivery of fuel to the respective fuel line **12** is terminated in a fully automatic way.

The auxiliary cylinder **3** receives fuel from the suction space **6** through a supply bore **21**_a_ and through one of several axially parallel grooves **5**_a_ machined into the periphery of the piston **5**. One of the grooves **5**_a_ will communicate with the supply bore **21**_a_ when the piston **5** completes a suction (downward) stroke but each of these grooves is invariably sealed from the bore **21**_a_ when the piston **5** performs an upward stroke whereby the fuel filling the interior of the auxiliary cylinder **3** escapes through a feed line **21** which discharges into the left-hand end of the regulating chamber **15**, i.e., at one axial end of the plunger **13**_a_. The feed line **21** contains a one-way valve **22** which prevents return flow of fuel from the chamber **15** into the interior of the auxiliary cylinder **3**. The intake end of the feed line **21** receives fuel through one of the grooves **5**_a_ when the piston **5** performs an upward stroke. Fuel entering the left-hand end of the regulating chamber **15** via feed line **21** will displace the valve **13** against the opposition of the spring **19** whereby the length of strokes necessary to be performed by valve **13** in order to establish a connection between the bores **14**, **16** of the overflow channel **14–16** depends on the axial position of the adjusting pin **20** and also on the fact whether or not the spring **19** had enough time to expel fuel from the chamber **15** and to return the valve **13** into actual abutment with the pin **20** before the auxiliary piston **5** starts to perform the next working stroke.

In normal operation of the engine, the return spring **19** will move the plunger **13**_a_ back into abutment with the adjusting pin **20** before the auxiliary piston **5** begins to perform the next working stroke. During such return movement of the valve **13** in a direction to the left, as viewed in FIG. 1, the plunger **13**_a_ expels a certain amount of fuel through a discharge line **23** which connects the left-hand end of the regulating chamber **15** with the interior of the auxiliary cylinder **3**. The discharge line **23** is provided with a throttling orifice or passage **24** whose cross-sectional area may be varied by an adjustable needle-like throttle **25**. The purpose of the throttle **25** is to brake the return flow of fuel via discharge line **23** in such a way that, starting from a predetermined speed of the engine E, the spring **19** will be unable to return the plunger **13**_a_ all the way into abutment with the adjusting pin **20** before the piston **5** starts the next working stroke. Thus, the valve **13** will have to cover a shorter than normal distance in order to permit flow of fuel from the bore **14** into the bore **16** so that the amount of fuel admitted to the fuel lines **12** is reduced accordingly, namely, as a function of the position of the valve **13** when the auxiliary piston **5** begins to perform an upward stroke. The exact engine speed at which the valve **13** cannot return into actual abutment with the adjusting pin **20** can be selected by a cam **28** which controls the axial position of the throttle **25**. The regulating action of the valve **13** upon the amounts of fuel admitted to the fuel lines **12** is called a "liquid abutment" or "liquid stop" and is fully disclosed in the aforementioned patent No. 3,122,100 to Bessiere.

As long as the speed of the engine E remains below the aforementioned predetermined speed, the amounts of fuel admitted to the fuel lines **12** will depend on the distance which the valve **13** must cover from abutment with the adjusting pin **20** to a position in which the end face **17**_a_ of the plunger **13**_b_ allows fuel to flow from the bore **14** into the bore **16**. The greater such distance, the greater is the amount of fuel which is admitted into a fuel line **12**. The just mentioned distance will depend on the axial position of the adjusting pin **20**, i.e., on the position of the aforementioned cam **26** which is shiftable by an actuating member here shown as a lever **30** rockable about a pivot pin **30**_a_ between a plurality of positions two of which are indicated by phantom lines. If the cam **26** is moved upwardly, the spring **27** can shift the pin **20** in a direction to the left so that the stroke of the valve **13** increases and each of the fuel lines **12** receives more fuel. When the engine operates at maximum load, the pin **20** will be shifted to its leftmost position which means that the actuating lever **30** will be rocked in a counterclockwise direction.

The aforementioned throttle **25** is biased by a helical expansion spring **29** which tends to maintain a roller follower **25**_a_ of the throttle in contact with the face of the cam **28**. The cams **26** and **28** are coupled to each other by an adjustable coupling device which resembles a turnbuckle and includes a pair of spindles **26**_b_, **28**_b_ respectively connected with the cams **26**, **28** and having left-hand and right-hand threads, and a spindle nut **32** which meshes with both spindles so that its rotation in opposite directions will respectively move the cams **26**, **28** nearer to or away from each other. The parts **25**_a_, **28**, **29** constitute a second adjusting unit which controls the throttle **25** and whose operation is synchronized with that of the adjusting unit **20**, **20**_a_, **26**, **27**. The cam **28** is guided in the ways **26A**.

The faces of the cams **26**, **28** are configurated in such a way that the cam **28** allows the throttle **25** to assume its outer end position (minimal throttling action) when the cam **26** maintains the adjusting pin **20** in an axial position corresponding to the operation of engine E under maximum load or higher-partial-load. Thus, and assuming that the actuating lever **30** is pivoted to its upper end position, the effective cross-sectional area of the throttling orifice **24** in the discharge line **23** will assume a maximum value and the adjusting pin **20** will be held in the left-hand end position to insure that the working piston **4** will inject into each fuel line **12** a maximum amount of fuel before the interior of the cylinder **2** begins to communicate with the suction space **6** via overflow channel **14–16**.

The liquid stop becomes effective regardless of the momentary load and insures that the amount of fuel admitted into the cylinders of the engine E is reduced automatically when the rotational speed of the engine exceeds a given value. This will be readily understood because, even when the throttle **25** is held in the fully withdrawn (outermost) position of FIG. 1, it offers a certain resistance to return flow of fuel into the interior of the auxiliary cylinder **3** so that the spring **19** has no time to return the plunger **13**_a_ into abutment with the adjusting pin **20** (in the leftmost position of this pin) when the intervals between successive upward strokes of the auxiliary piston **5** are too short. The actual amounts of fuel which are admitted into the fuel lines **12** can be regulated at the operator's discretion until the rotational speed of the engine E reaches such given value, i.e., before the liquid stop becomes effective.

When the actuating lever **30** is moved to a position corresponding to idling or lower-partial-load upon the engine E, namely, when the adjusting pin **20** is moved to a predetermined extent from its left-hand end position, the follower **25a** begins to track a lobe **28a** of the cam **28** and the braking action of the throttle **25** is intensified. This reduces the cross-sectional area of the orifice **24**. The rotational speed of the engine will decrease proportionally with rightward movement of the adjusting pin **20**. In response to a certain rightward axial displacement of the pin **20**, the speed of the engine E is reduced to idling speed, i.e., the amounts of fuel injected by the piston **4** into successive fuel lines **12** are reduced to a minimum.

In the fuel injection pump of FIG. 1, the throttle **25** can limit the maximum rotational speed at full load and at higher-partial-load. In addition, the throttle **25** will determine various idling speeds and speeds at lower-partial-load. However, it is normally preferred to provide a secondary throttle **31a** which controls the effective area of a second throttling orifice or passage **31** in the discharge line **23**. The throttle **31a** is adjustable, but not automatically, and its purpose is to take over some of the functions of the throttle **25**. Thus, the throttle **31a** will insure that the speed of the engine E at maximum load or higher-partial-load will not exceed a predetermined value, and the throttle **24** will be effective only when its follower **25a** engages the lobe **28a** of the cam **28**, namely, when the tip of the throttle **25** begins to reduce the effective cross-sectional area of the orifice **24**. In other words, the throttle **25** will control the operation of the engine E at lower-partial-load or during idling. When the secondary throttle **31a** is provided, its setting is such that the cross-sectional area of the orifice **31** does not exceed the cross-sectional area of the orifice **24** when the throttle **25** is moved to its left-hand end position. This contributes to greater versatility of the fuel injection pump because the operator can arbitrarily select the maximum rotational speed of the engine at full load or at higher-partial-load, i.e., the maximum rotational speed can be selected in accordance with the requirements of the engine.

FIG. 2 illustrates a modified fuel injection pump. Such parts of this modified pump which are analogous to those already described in connection with FIG. 1 are denoted by similar reference numerals each followed by a prime.

An important difference between the fuel injection pumps of FIGS. 1 and 2 is that the latter pump comprises a regulating valve or shuttle piston **13'** which is rotatable and reciprocable in its chamber **15'** and that the adjusting unit including the pin **20** and cam **26** of FIG. 1 is replaced by an adjusting unit which changes the angular position of the valve **13'**. The annular groove **17'** can establish a connection between the bores **14'**, **16'** of the overflow channel in response to changes in axial and/or angular position of the valve **13'**. Futhermore, the fuel injection pump of FIG. 2 is provided with a different throttle **25'** which replaces the throttle **25** of FIG. 1 and controls the effective cross-sectional area of an orifice or passage **24'**. Instead of being effective solely due to its axial displacement as the throttle **25**, the throttle **25'** of FIG. 2 is also turnable in the pump body **1'** and such turning is utilized to change the effective cross-sectional area of the orifice **24'** by means of an inclined edge face **39** of the throttle **25'**.

The plunger **13b'** of the valve **13'** has an inclined end face **33** which replaces the end face **17a** shown in FIG. 1 and controls the flow of fuel from the annular groove **17'** into the bore **16'**. As clearly shown in FIG. 2, the axial distance which must becovered by the valve **13'** from its left-hand end (starting) position to the position in which the groove **171'** can communicate with the bore **16'** will be varied in response to changes in angular position of the valve **13'**. The starting position of the valve **13'** is determined by a collar **34** which then abuts against an internal stop surface **35** of the pump body **1'**. The adjusting unit which can change the angular position of the valve **13'** comprises a shaft **36** which is coaxial with the plungers **13a'**, **13b'** and is rotatable in the pump

body **1'**. This shaft **36** is coupled with a flange **38** of the valve **13'** by an axially parallel pin **37** which permits the valve to move axially toward and away from the shaft **36** in response to or against the bias of the return spring **19'**. The spring **19'** is received in a recess provided in the left-hand end face of the shaft **36**. It will be seen that the motion transmitting connection between the shaft **36** and valve **13'** does not interfere with operation of the return spring **19'** which tends to maintain the collar **34** in abutment with the stop surface **35**.

The pump body **1'** has a bore **1b'** which receives the throttle **25'** and a portion of this bore **1b'** forms part of the discharge line **23'** which connects the interior of the auxiliary cylinder **3'** with the left-hand end of the regulating chamber **15'**. The secondary throttle **31a'** is adjustably mounted on the throttle **25'** and controls the flow of fuel toward the orifice **24'**. The orifice **31'** whose cross-sectional area is regulated by the throttle **31a'** is provided in the throttle **25'**.

The coupling device which connects the throttle **25'** with the shaft **36** comprises two intermeshing gear segments **41**, **42** best shown in FIG. 3. The function of the gear segments **41**, **42** is to change the angular position of the throttle **25'** as a function of angular displacement of the shaft **36** and regulating valve **13'**. The actuating lever **30'** is connected with the gear segment **41** or directly with the shaft **36**, and the angular position of the throttle **25'** with reference to the shaft **36** can be changed by means of a plate **45** which is releasably fixed to the shaft **36** by a nut **46** and has an arcuate slot **44** receiving the stem of the lock screw **43** which meshes with the gear segment **41**. The latter is rotatable on the shaft **36** and is held against rotation with reference to this shaft when the lock screw **43** is drawn tight to prevent rotation of the shaft **36** with reference to the gear segment **41** or vice versa. By loosening the screw **43**, the operator can change the angular position of the throttle **25'** while the angular position of the shaft **36** remains unchanged, or vice versa.

The angular position of the regulating valve **13'** with reference to the throttle **25'** and the inclination of the end face **33** on the plunger **13b'** are selected in such a way that the angular positions of the valve **13'** requiring long travel of the end face **33** into a position to establish a connection between the bores **14'** and **16'** (when the engine operates at full load or higher-partial-load) correspond to such positions of the throttle **25'** in which the effective cross-sectional area of the orifice **24'** is at a maximum value. In such positions of the throttle **25'** only the secondary throttle **31a'** is effective to interfere with return flow of fuel through the discharge line **23'** and to thus determine the maximum rotational speed of the engine at full load or higher-partial-load. When the angular position of the regulating valve **13'** is such that a short axial movement of the valve sufficies to establish a connection between the bores **14'** and **16'** (i.e., when the engine is idling or operates under lower-partial-load), the edge face **39** of the throttle **25'** reduces the effective cross-sectional area of the orifice **24'** so that fuel flowing back through the discharge line **23'** meets a greater resistance. In such ranges of operation of the engine, the secondary throttle **31a'** is ineffective because the cross-sectional area of the orifice **24'** is then smaller than that of the orifice **31'**.

A torsion spring **47** is provided to eliminate backlash between the teeth of the gear segments **41**, **42**. One end of this spring is anchored in the pump body **1'** and its other end is fixed to the gear segment **42**.

The pump body **1'** is further provided with a bypass channel **48** which connects the suction space **6'** with the regulating chamber **15'** and one end of which is normally sealed by the plunger **13a'**. A portion of the throttle **25'** regulates the flow of fuel between the two branches of the bypass channel **48**. The effective cross-sectional area of the throttling orifice **49** in the channel **48** can be varied by an inclinded edge face **50** of the throttle **25'** in such a way that the area of the orifice **49** increases when the area of the orifice **24'** decreases and vice versa. The plunger **13a'** will permit flow of fuel from the chamber **15'** into the bypass channel **48** before the chamber **15'** is permitted to communicate with a relief channel **51** discharging into the bore **16'** of the overflow channel. The relief channel **51** limits the strokes of the valve **13'** by permitting fuel delivered by the auxiliary pump **3'**, **5'** to escape into the bore **16'**. The function of the bypass channel **48** is to vary the proportional band of regulation of the fuel injection pump.

In the fuel injection pumps of FIGS. 1 and 2, regulation during idling and during lower-partial-load upon the engine is effected by varying the effective cross-sectional area of the throttling orifice **24** or **24'** and by simultaneously varying the distance between the starting position of the valve **13** or **13'** and that position in which the plunger **13b** or **13b'** connects the bores **14**, **16** or **14'**, **16'** of the respective overflow channel. FIG. 4 illustrates a third fuel injection pump which differs from the previously described pumps in that during starting, idling and operation at lower-partial-load, the regulation is effected by means of fuel flowing through a spill passage or spill conduit **52**. This spill passage **52** connects the suction space **6''** with the regulating chamber **15''** at the free end of the plunger **13a''**, i.e., with that region of the chamber **15''** from which the valve **13''** expels fuel in response to expansion of the return spring **19''** and which receives fuel from the auxiliary pump **3''**, **5''**. At very low engine speeds (during starting), at least a portion of fuel delivered by the auxiliary piston **55''** can escape through the spill passage **52** so that the strokes of valve **13''** from its starting position are short and the working piston **4''** delivers to fuel lines **12''** maximum amounts of fuel. The spill passage **52** has a throttling orifice **53** whose cross-sectional area may be regulated by a needle-like throttle **55**. This throttle **55** insures that, once the engine is started and its r.p.m. rises to a certain value, the fuel pressure suffices to open the overflow channel **14''–16''**.

The interior of the auxiliary cylinder **3''** is further connected with a discharge line **54** corresponding to the discharge line **23** of FIG. 1 and containing an orifice **24''** adjustable by a needle-like throttle **25''** which brakes the return movement of valve **13''** to starting position. The throttle **25''** insures that the liquid stop develops when the engine speed rises to a predetermined value. The throttles **25''**, **55** are adjustable by hand and their position then remains unchanged.

Such parts of the pump shown in FIG. 4 which are clearly analogous to those described in connection with FIGS. 1 and 2 are denoted by similar reference numerals each of which is followed by two primes.

The flow of fuel through the discharge line **54** and spill passage **52** is further controllable by a rotary overriding throttle **56** which is coupled with the regulating valve **13''** substantially in the same way as described in connection with the valve **13'** and throttle **25'** of FIG. 2. The throttle **56** can seal the line **54** and passage **52** or it can reduce the flow of fuel more than the throttles **55** and **25''**. An inclined end face **57** of the regulating valve **13''** performs the same function as the end face **33** shown in FIG. 2, i.e., the distance which the plunger **13b''** must cover by moving from its starting position in order to establish a connection between the bores **14''** and **16''** can be varied by changing the angular position of the valve **13''** through the intermediary of actuating lever **30''**, shaft **36''** and coupling pin **37''**. The starting position of the valve **13''** is determined by the collar **34''** and stop surface **35''**.

The coupling between the shaft **36''** and overriding throttle **56** includes gear segments **41''**, **42''**. The actuating lever **30''** is connected with the gear segment **41''** and can be pivoted to an infinite number of positions each of which corresponds to a different setting of the valve **13''** and throttle **56**. The backlash eliminating spring **47** is not shown in FIG. 4 but the coupling **41''**, **42''** of FIG. 4 can be provided with such a spring or with an analogous device which eliminates play between the intermeshing teeth.

The overriding throttle **56** is formed with arcuate cutouts **56a**, **56b** which can be moved into partial or full registry with the spill passage **52** and discharge line **54**. The configuration of the cutout **56b** is shown in FIG. 8 and the cutout **56a** is preferably of similar configuration.

The angular position of the throttle **56** with reference to the regulating valve **13''** is selected in such a way that the spill passage **52** is sealed when the engine operates at full load or higher-partial-load. The discharge line **54** is then fully open, i.e., the overriding throttle **56** does not interfere with flow of fuel through the line **54** and such flow is then controlled solely by the throttle **25''**. This means that the selected cross-sectional area of the orifice **24''** determines the maximum rotational speed of the engine. Such maximum rotational speed is selected in advance by setting of the throttle **25''** but the operator is free to vary the amounts of fuel delivered to fuel lines **12''** in the speed range below the preselected maximum speed by changing the angular position of the regulating valve **13''**. When the valve **13''** is moved to an angular position which corresponds to idling of the engine, the overriding throttle **56** seals the discharge line **54** or throttles the flow of fuel through the line **54** more than the throttle **25''**. At the same time, the throttle **56** places its cutout **56a** into full or substantially full registry with the spill passage **52** so that the flow of fuel through the passage **52** is controlled exclusively by the throttle **55**.

Closing of the discharge line **54** and opening of the spill passage **52** will be effected gradually with a plurality of intermediate positions of the overriding throttle **56**. For example, the throttle **56** will start to open the spill passage **52** before it begins to close the discharge line **54**. Also, the line **54** is sealed completely before the passage **52** is fully open.

Angular displacements of the actuating lever **30''** from the position corresponding to full load to the position corresponding to idling of the engine will bring about the following results:

At the outset, i.e., when the engine operates at full load or at higher-than-average load (nearly full load), angular displacement of the lever **30''** will cause turning of the valve **13''** in order to reduce the amounts of fuel delivered to the fuel lines **12''** in response to successive working strokes of the piston **4''**. The discharge line **54** remains fully open and the spill passage **52** is sealed, i.e., the cutout **56a** is out of registry with the passage **52** and the flow of fuel through the line **54** is determined solely by the setting of the throttle **25''**.

When the lever **30''** is moved to a position in which the amounts of fuel delivered to the fuel lines **12''** correspond to between 75 and 80 percent of the amount at full load, the cutout **56a** of the overriding throttle **56** begins to move into partial registry with the spill passage **52** while the cutout **56b** still remains in full registry with the discharge line **54**. Up to such position of the lever **30''**, the maximum rotational speed remains substantially constant and, as long as the actual rotational speed is below such maximum speed, the fuel injection pump operates in a range without automatic regulation. In other words, the amounts of fuel which will enter the cylinders of the engine can be selected by the operator, not by the automatic regulating system of the pump.

If the operator continues to turn the lever **30''** in a direction away from the position corresponding to full load, namely, toward the position corresponding to idling of the engine, the throttle **56** begins to close the discharge line **54** by moving the cutout **56b** out of full registry with the adjoining sections of the line **54**. The throttling effect of the throttle **25''** is then greater than that of the throttle **25''**. This starts the regulation at lower-than-average load which is automatic and determines the rotational speed of the engine. If the lever **30''** continues to reduce the amounts of fuel which enter the fuel lines **12''**, the cutout

56*a* reaches an angular position in which the throttle **56** cannot interfere with the flow of fuel through the leak passage **52** while the throttle **56** simultaneously reduces considerably the flow of fuel in the discharge line **54**. The spill passage **52** is fully open when the valve **13″** assumes an angular position in which the fuel lines **12″** receive just as much as or a little more fuel than necessary for idling of the engine. In the idling position of the lever **30″**, the throttle **56** completely seals the discharge line **54** or leaves only a very narrow path for the flow of fuel through the cutout **56*b*** which means that the fuel lines **12″** will receive no fuel as soon as the rotational speed of the engine exceeds a predetermined idling speed. If the lever **30″** moves beyond such idling position, a peripheral connecting groove **58** of the plunger **13*b*″** establishes a direct connection between the annular groove **17″** and the bore **16″** so that the interior of the working cylinder **2″** is in continuous communication with the suction space **6″** and the engine cylinders receive no fuel at all. This brings the engine and hence the pistons **4″**, **5″** to a halt.

FIG. 7 shows a fourth fuel injection pump which is very similar to the pump of FIG. 4. Therefore, the reference numerals used in FIGS. 4 and 7 are the same. In addition to the structure of FIG. 4, the pump body **1″** of this fourth pump is provided with a bypass channel **59** corresponding to the channel **48** of FIG. 2. The flow of fuel in the channel **59** is controlled by a preset needle-like throttle **59*a*** and also by an inclined edge face bounding one side of a cutout **56*c*** in the overriding throttle **56**. The latter seals the channel **59** when the engine operates at full load or higher-partial-load. The channel **59** will convey fuel in that range of operations when the rotational speed of the engine is determined automatically, not at the will of the operator. The rate of fuel flow in the channel **59** will determine the proportional band of the regulation.

It will be noted that the throttles **25″** and **55** in the pumps of FIGS. 4 and 7 are disposed between the regulating chamber **15″** and the cutouts **56*b***, **56*a*** of the overriding throttle **56**. These throttles **25″**, **55** are installed in parallel. The passage **52** and discharge line **54** communicate with a portion of the regulating chamber **15″** which is located downstream of a one-way valve **60** provided in a feed line **21″** corresponding to the feed line **21** of FIG. 1.

FIG. 5 shows a portion of a fifth fuel injection pump wherein the throttles **25″**, **55** are installed in series. When the spill passage **52** is open, fuel flowing therethrough must pass first through the orifice **24″**, thereupon through the orifice **53**, and finally through the cutout **56*a*** of the overriding throttle **56**. Such arrangement renders it possible to provide the body **1″** with an orifice **53** of relatively large cross-sectional area. When the throttle **56** assumes an angular position in which the cutouts **56*a***, **56*b*** respectively open the passage **52** and discharge line **54**, a first portion of fuel displaced by the regulating valve **13″** from the chamber **15″** flows through the orifices **24″**, **53**, cutout **56*a*** and back to the suction space **6″**. A second portion of such fuel flows from the orifice **24″** to cutout **56*b*** and via discharge line **54** back into the auxiliary cylinder **3″** which is not shown in FIG. 5.

The reference numerals employed in FIG. 5 are the same as those in FIGS. 4 and 7. The same holds true for the fuel injection pump of FIG. 6.

In the pump of FIG. 6, the throttles **25″**, **55** are installed in the spill passage **52** and discharge line **54** downstream of the cutouts **56*a***, **56*b*** in the throttle **56**. The passage **52** branches from the downstream portion of the discharge line **54** so that it bypasses the one-way valve **60** in the feed line **21″**. Consequently, the throttle **55** which controls the orifice **53** is effective only during forward movement of regulating valve **13″** (against the opposition of return spring **19″** which is not shown in FIG. 6) and only during idling and starting of the engine. When the valve **13″** moves under the bias of the return spring **19″**, fuel which fills the regulating chamber **15″** below the valve **13″** will escape only through the cutout **56*b***, orifice **24″** and through the discharge line **54** back into the interior of the auxiliary cylinder **3″** (not shown).

In this embodiment of our fuel injection pump, the throttle **56** will not completely seal the discharge line **54** when the engine is idling. Automatic selection of the maximum speed with application of the liquid stop will be made by means of the throttle **25″**. The throttle **55** retards the forward stroke of the valve **13″** and thus increases the amounts of fuel which are supplied to the fuel lines **12″**.

In the embodiments which are illustrated in FIGS. 4 to 7, the flow of fuel through the discharge line **54** and spill passage **52** can be regulated by a single control element, namely, by the overriding throttle **56** having the aforementioned cutouts **56*a***, **56*b***. In order to render the regulating system even more versatile, one can vary the relationship of positions in which the throttle **56** permits, throttles or prevents the flow of fuel through the passage **52** and line **54**. Such a modified fuel injection pump is shown in FIG. 9 wherein the overriding throttle **56** comprises two inclined camming edges **56*d***, **56*e*** which respectively bound portions of the cutouts **56*a***, **56*b***. The edges **56*d***, **56*e*** are located in mutually inclined planes making an acute angle with the axis of the throttle **56** and the latter is movable axially without interrupting the connection established by the coupling **41″**, **42″** (only the gear segment **42″** is actually shown in FIG. 9). The means for adjusting the throttle **56** axially comprises a threaded adjusting member **63** which can transmit motion to a flange **62** of the throttle **56** by means of a two-armed lever **61**. A spring **62*a*** operates between the flange **62** and the pump body **1″** to maintain the lever **61** in permanent abutment with the flange **62** and adjusting member **63**. The motion transmitting connection between the gear segment **42″** and throttle **56** comprises an axially parallel coupling pin **62*b*** which is slidable in the flange **62** and/or in a flange **42*b*″** of the shaft **42*a*″** for the gear segment **42″**.

Many reference numerals employed in FIG. 9 are the same as those in FIGS. 4–8 because the structure for adjusting the axial position of the throttle **56** can be readily incorporated in the pumps of FIGS. 4–8.

As stated before, the throttles **25″**, **55** of FIGS. 4 to 7 are adjusted when the respective fuel injection pump is idle and thereupon remain in such preselected positions. FIGS. 10 and 11 illustrate the manner in which the throttles **25″**, **55** may be adjusted. The adjusting arrangement comprises a cover plate **64** which can be fitted into an opening **65** provided in a wall **70** bounding the suction space. The cover plate **64** is then sealingly affixed to the wall **70** by screws **71**.

Two motion transmitting bolts **66**, **67** are rotatably mounted in the cover plate **64** and have slotted outer ends **66*b***, **67*b*** so that each thereof can be rotated by a screwdriver or another rudimentary tool. The inner end portions **66*a***, **67*a*** of the bolts **66**, **67** constitute sockets which can receive non-circular heads of the throttles **25″**, **55**. The sockets **66*a***, **67*a*** are in engagement with the respective throttles when the cover plate **64** is affixed to the wall **70**.

When the adjustment is completed, the cover plate **64** is removed and the opening **65** is sealed by a lid **68**. This lid **68** remains attached to the wall **70** when the fuel injection pump is in actual use.

Of course, and in order to be able to utilize the cover plate **64**, the designer of the fuel injection pump must install the throttles **25″**, **55** sufficiently close to and in parallelism with each other, otherwise, the body of the pump will be provided with several openings so that the operator can gain access to all such throttles which are not adjustable by the lever **30**, **30′** or **30″**.

The cover plate **64** can remain attached to the wall **70** when the pump is in actual use, at least during testing of the engine, so that the operator can make eventual adjustments if the engine fails to run properly. The lid **68** will be applied when the operator has found a satisfactory angular position for each of the throttles **25''**, **55**.

Without further analysis, the foregoing will so fully reveal the gist of the present invention that others can, by applying current knowledge, readily adapt it for various applications without omitting features which fairly constitute essential characteristics of the generic and specific aspects of our contribution to the art and, therefore, such adaptations should and are intended to be comprehended within the meaning and range of equivalence of the following claims.

What is claimed as new and desired to be protected by Letters Patent is:

1. A fuel injection pump for internal combustion engines comprising a main cylinder; a main piston reciprocable in said main cylinder to respectively draw and expel fuel therefrom; drive means for reciprocating said piston at a speed which is a function of engine speed; fuel line means arranged to receive fuel from said cylinder for injection into the engine; overflow channel means connected with said cylinder and including a regulating chamber; an adjustable regulating valve reciprocable in said chamber and movable from at least one starting position to at least one open position in which said channel means is effective to evacuate fuel from said cylinder instead of said fuel line so that the injection of fuel into the engine is terminated on movement of said valve to said open position; biasing means for urging said valve to said starting position; auxiliary pump means operating in synchronism with said main piston to deliver fuel to said chamber for moving said valve from said starting position; a discharge line connected with said chamber to evacuate fuel during movement of said valve to said starting position; adjustable throttling means in said discharge line for restricting the cross section thereof and for thus braking the flow of fuel through said discharge line; first adjusting means for regulating the distance between starting and open positions of said valve; second adjusting means for adjusting said throttling means to regulate the throttling action thereof; coupling means between said adjusting means and arranged to maintain said second adjusting means in at least one first position when said first adjusting means is set for maximum or near maximum of fuel delivery per stroke and to maintain said second adjusting means in second positions corresponding to increasing braking action of said throttling means when said first adjusting means is set for minimum or near minimum of fuel delivery per stroke; and common actuating means for said first and second adjusting means.

2. A fuel injection pump as defined in claim **1**, further comprising secondary throttling means adjustable independently of said first mentioned throttling means to determine the rate of fuel flow in said discharge line in said first position of said second adjusting means by bringing about a braking action which exceeds the braking action of said first mentioned throttling means during maximum or near maximum fuel delivery per stroke.

3. A fuel injection pump as defined in claim **1**, wherein each of said adjusting means comprises a cam and said first adjusting means further comprises an adjusting member shiftable by the respective cam and having a portion extending into said regulating chamber to determine the starting position of said regulating valve, said throttling means comprising a throttle reciprocable by the respective cam and having a portion extendable into said discharge line to obstruct the flow of fuel from said chamber.

4. A fuel injection pump as defined in claim **3**, wherein said coupling means comprises a turnbuckle adjustably connecting said cams to each other and wherein said actuating means comprises an actuating member connected with one of said cams.

5. A fuel injection pump as defined in claim **1**, further comprising spill conduit means connected with said regulating chamber to evacuate some of the fluid from said chamber independently of the position of said regulating valve when said actuating means is set for minimum or near minimum delivery per stroke, said throttling means comprising means for stopping the flow of fuel through said conduit means when said actuating means is set for maximum or near maximum delivery of fuel per stroke.

6. A fuel injection pump as defined in claim **5**, wherein said throttle means comprises a rotary overriding throttle a portion of which constitutes said means for stopping the flow of fuel through said conduit means.

7. A fuel injection pump as defined in claim **6**, wherein said overriding throttle comprises a second portion rigid with said first mentioned portion and arranged to regulate the flow of fuel in said discharge line.

8. A fuel injection pump as defined in claim **7**, further comprising third adjusting means for moving said overriding throttle axially, said portions of the overriding throttle having mutually inclined edges which change the rate of fuel flow in said discharge line and said conduit means in response to axial adjustment of the overriding throttle.

9. A fuel injection pump as defined in claim **5**, wherein said coupling means is arranged to effect sealing of said conduit means by said throttling means in the first position of said second adjusting means and to fully open said conduit means when said actuating means is set for idling of the engine.

10. A fuel injection pump as defined in claim **5**, further comprising bypass channel means having an end connected with said regulating chamber, said end of said bypass channel means being sealed by said regulating valve in the starting position and during movement of said valve to a position at a predetermined distance from such starting position, said throttling means comprising a portion arranged to regulate the flow of fuel in said bypass channel means.

11. A fuel injection pump as defined in claim **5**, further comprising second throttling means for regulating the flow of fuel in said spill conduit means independently of said first mentioned throttling means.

12. A fuel injection pump as defined in claim **11**, further comprising third throttling means for regulating the flow of fuel in said discharge line independently of said first mentioned throttling means.

13. A fuel injection pump as defined in claim **12**, wherein said second and third throttling means are installed in series so that said third throttling means regulates the flow of fuel in said discharge line and in said spill conduit means while said second throttle means regulates the flow of fuel only in said conduit means.

14. A fuel injection pump as defined in claim **1**, further comprising a feed line for supplying fuel from said auxiliary pump means to said regulating chamber, one-way valve means provided in said feed line, and spill conduit means directly connected with said auxiliary pump means and adapted to receive fuel therefrom, said throttling means comprising means for stopping the flow of fuel through said conduit means when said actuating means is set for maximum and near maximum fuel delivery per stroke.

15. A fuel injection pump as defined in claim **1**, wherein said first adjusting means comprises means for changing the angular position of said regulating valve and wherein said valve is arranged to perform strokes of different length between said starting and open positions thereof in response to angular displacement by said first adjusting means.

16. A fuel injection pump as defined in claim **15**, wherein said throttling means is rotatable by said second adjusting means and wherein said coupling means comprises intermeshing gear members.

17. A fuel injection pump as defined in claim **1**, wherein said auxiliary pump means comprises an auxiliary cylinder coaxial with said main cylinder and an auxiliary piston rigid with said main piston, said pistons being arranged to draw fuel from a common source and said overflow channel means being arranged to discharge into said common source.

18. A fuel injection pump as defined in claim **17**, wherein said drive means comprises means for rotating said pistons at the speed of the engine and said fuel line means comprises a plurality of fuel lines each arranged to normally receive fuel from said main cylinder during each revolution of said main piston.

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