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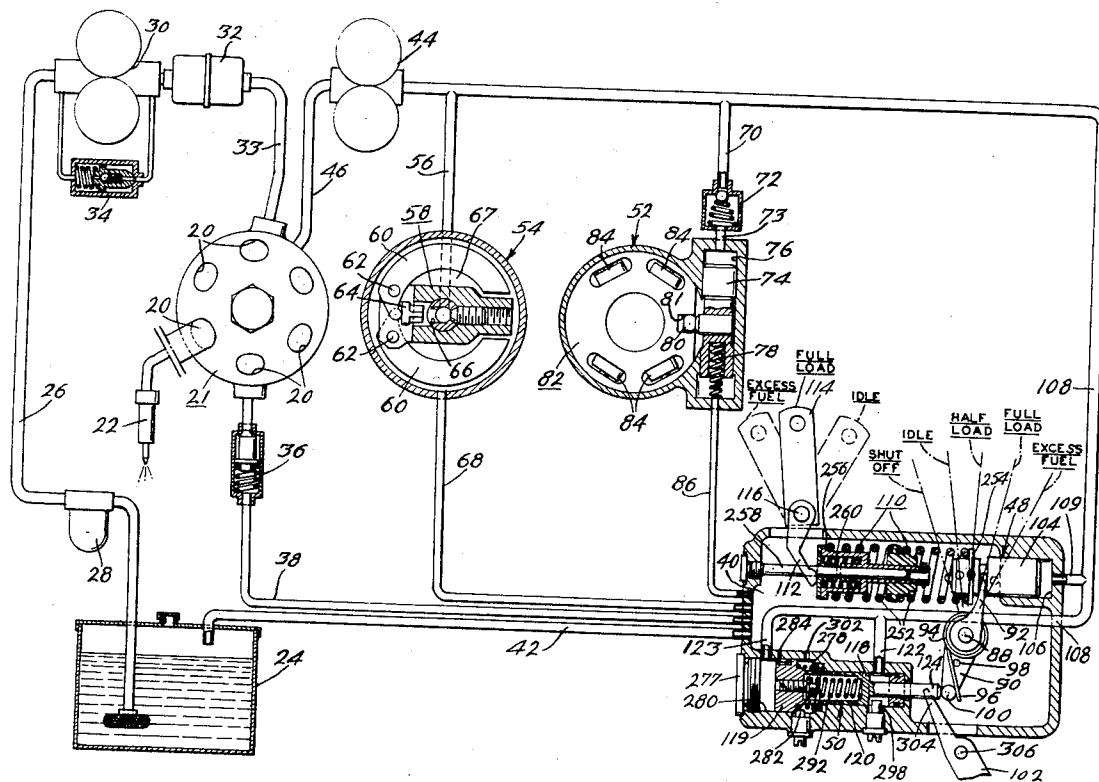
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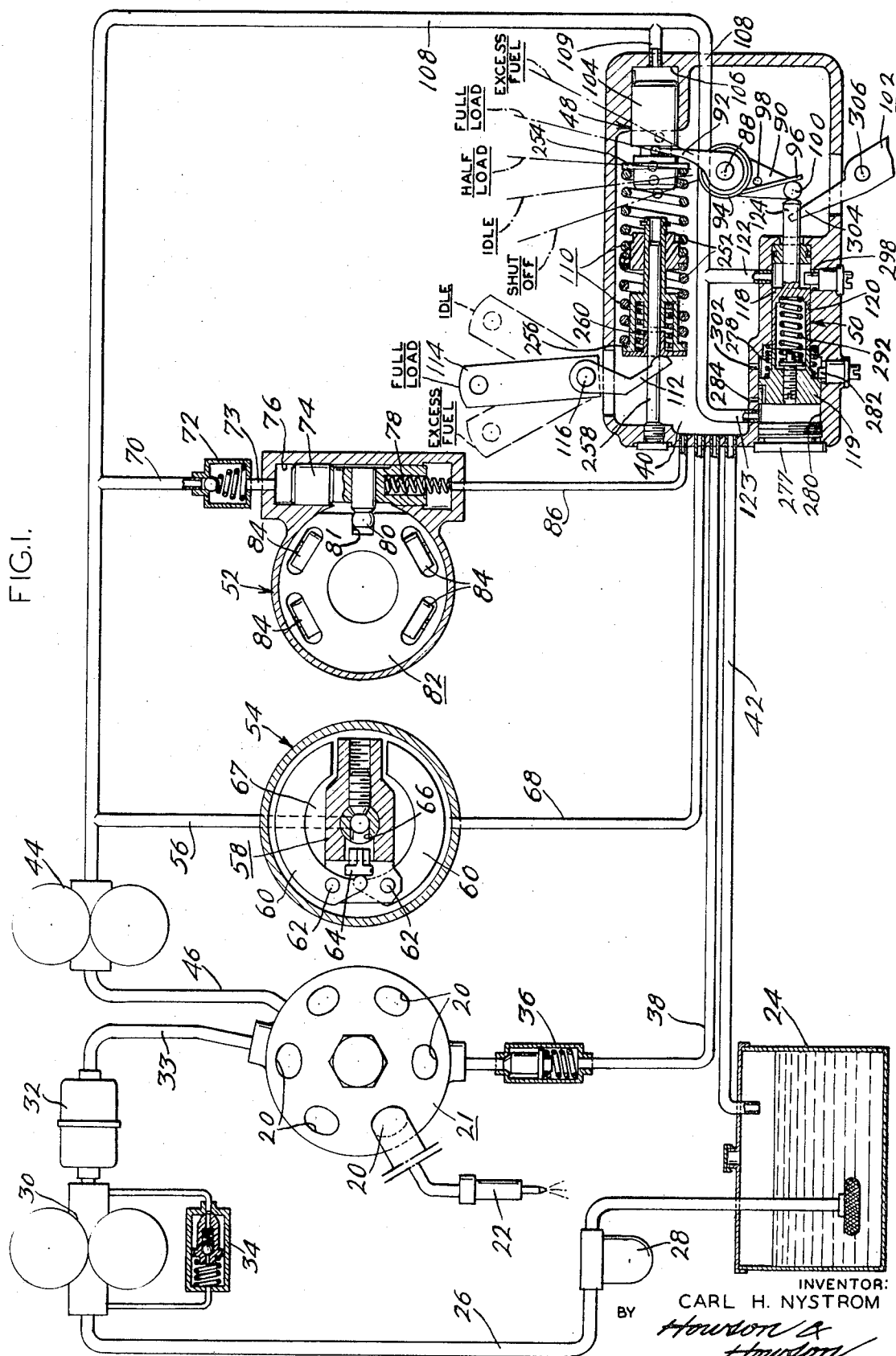
[54] **FUEL INJECTION PUMP**
12 Claims, 24 Drawing Figs.

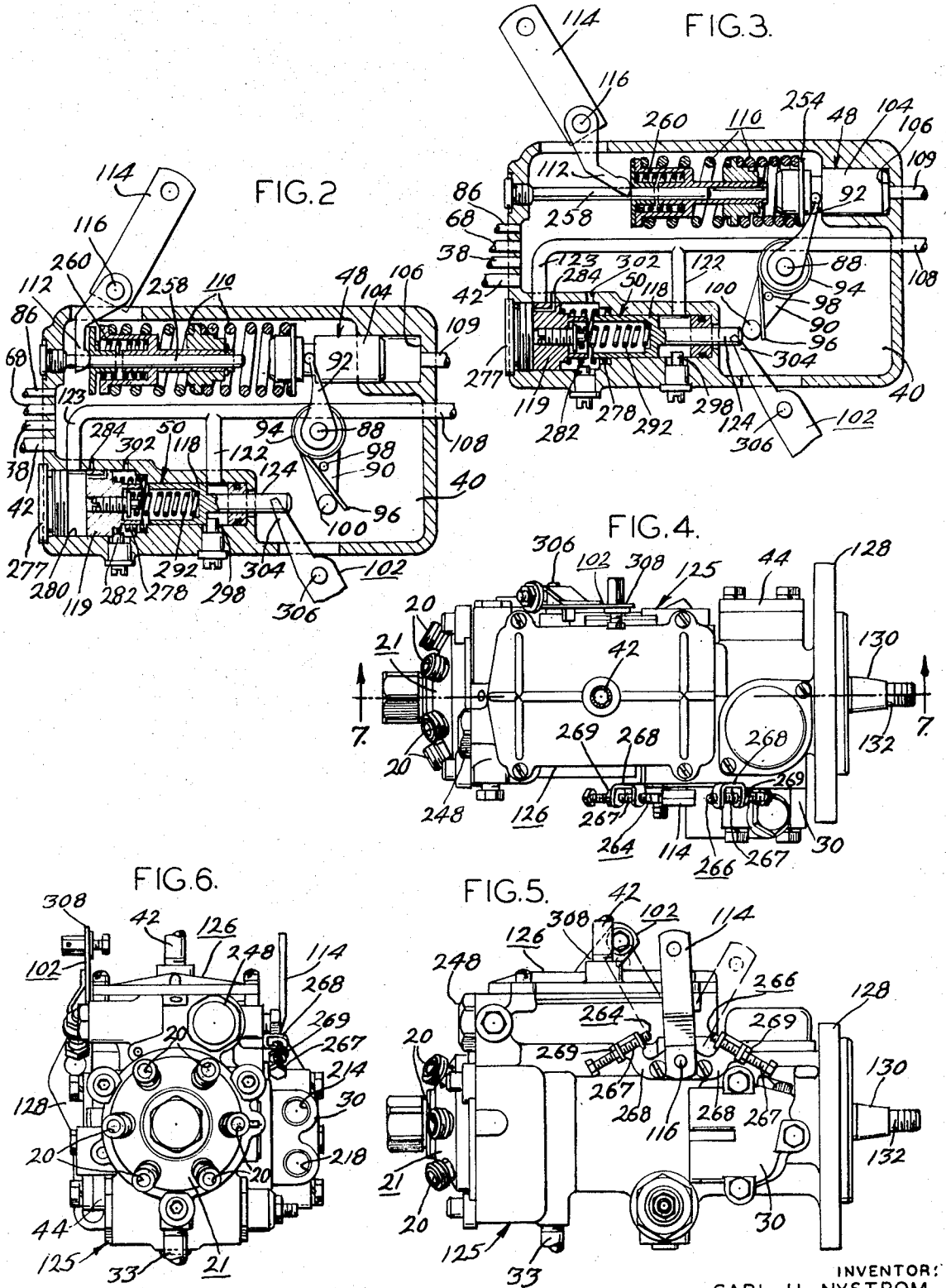
[52] U.S. Cl. **123/140,**
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 [51] Int. Cl. **F02d 1/12**
 [50] Field of Search. 123/139,
 140, 179h; 103/2.1, 37, 41S, 57

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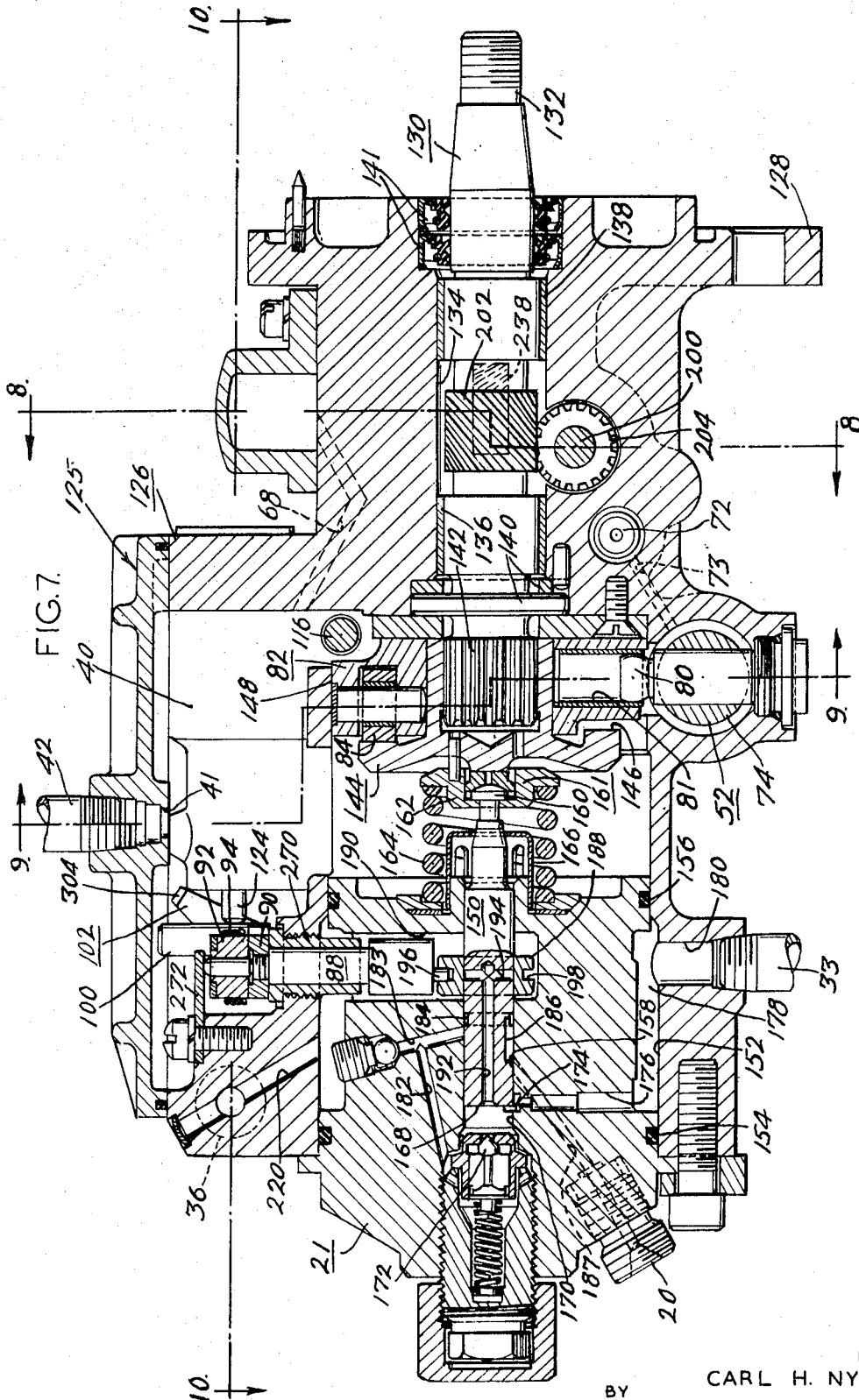
ABSTRACT: A fuel injection pump for injecting metered quantities of liquid fuel in timed sequence into the cylinders of an internal combustion engine through suitable nozzles. The pump is characterized by a single reciprocating and rotating pumping plunger, the rotation of which by means of an axial distributing slot thereon provides an accurately metered distribution of fuel to the engine nozzles. The pump includes an integral hydraulic governor which senses changes in engine speed by reference to fuel pressure. A centrifugal regulator maintains a control fuel pressure which is an accurate indication of engine speed. A hydraulic torque control is provided in association with the hydraulic governor to restrict full load fuel delivery to predetermined levels. An automatic hydraulic excess fuel device is coupled with the hydraulic torque control to permit the necessary excess starting fuel required during cranking of the engine.



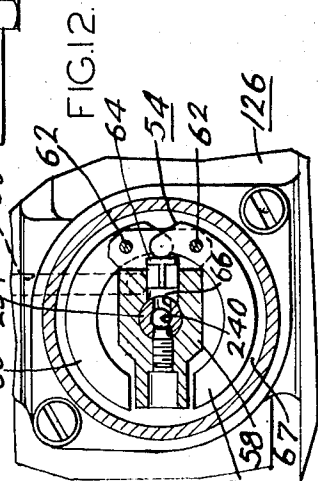
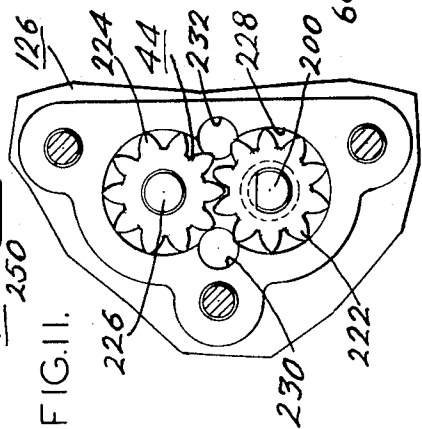
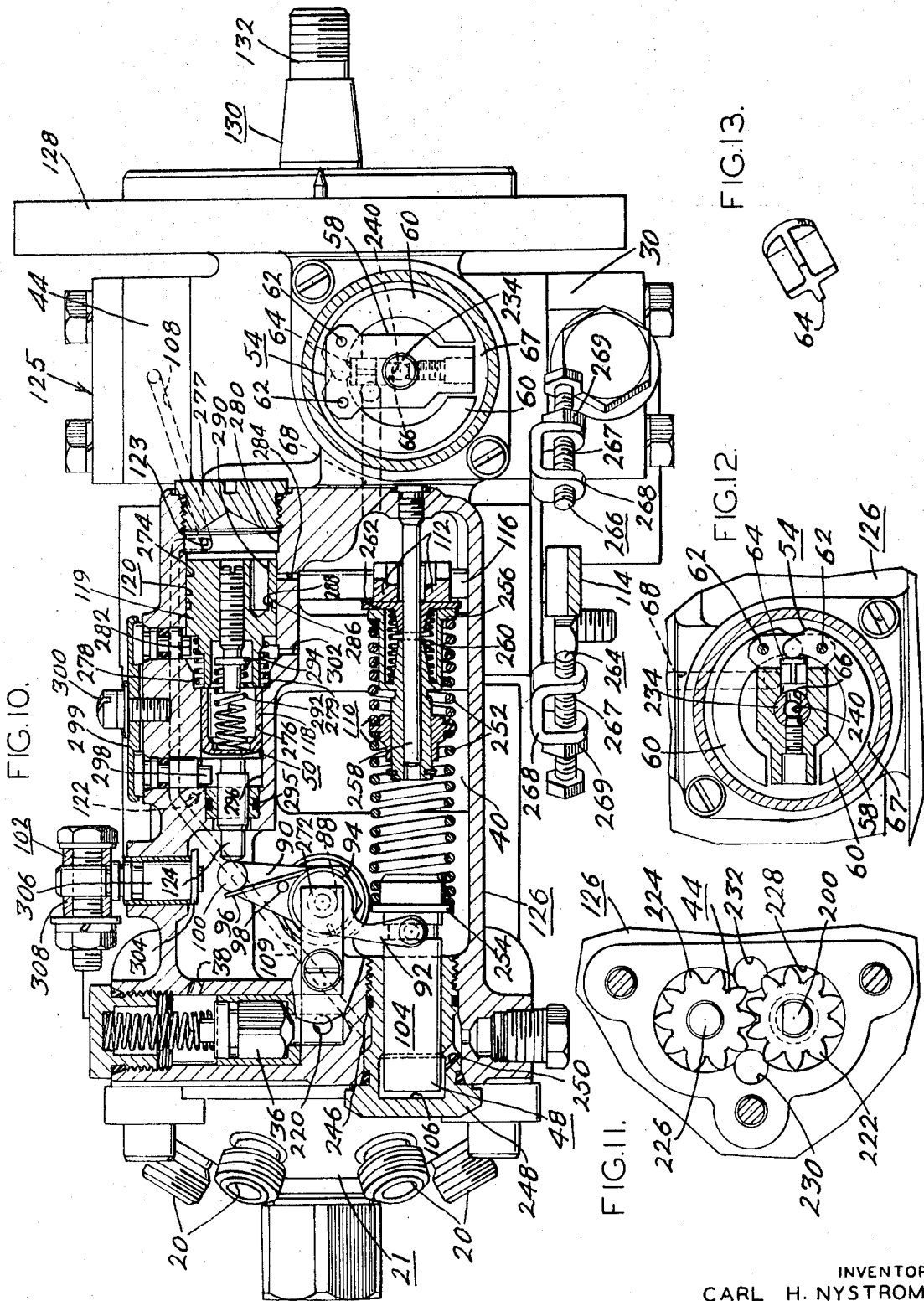




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FIG.14.

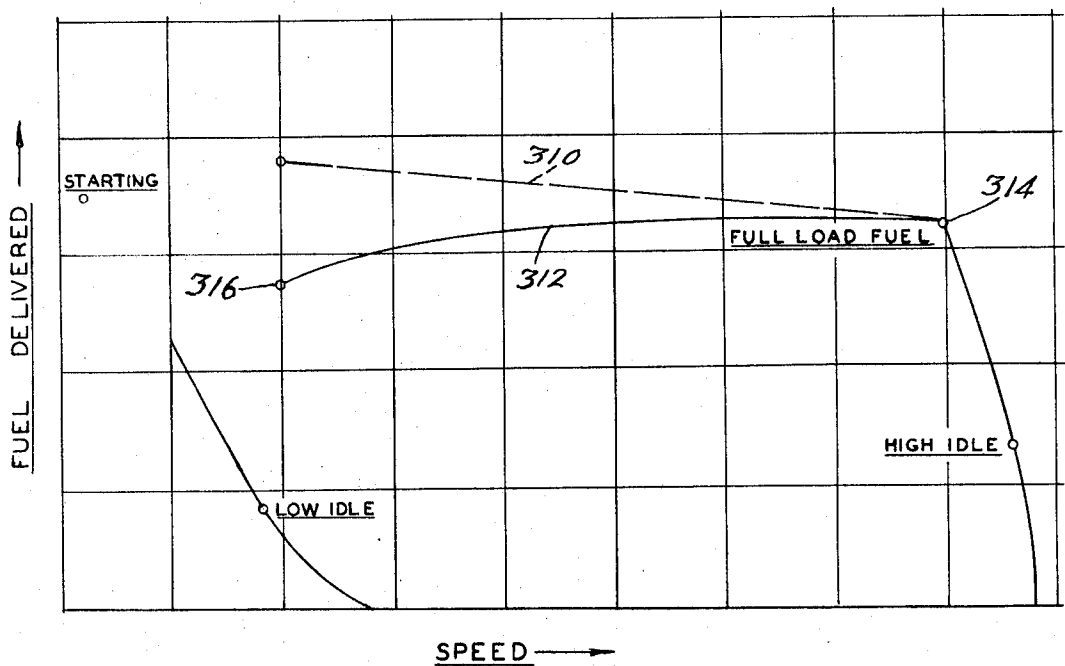
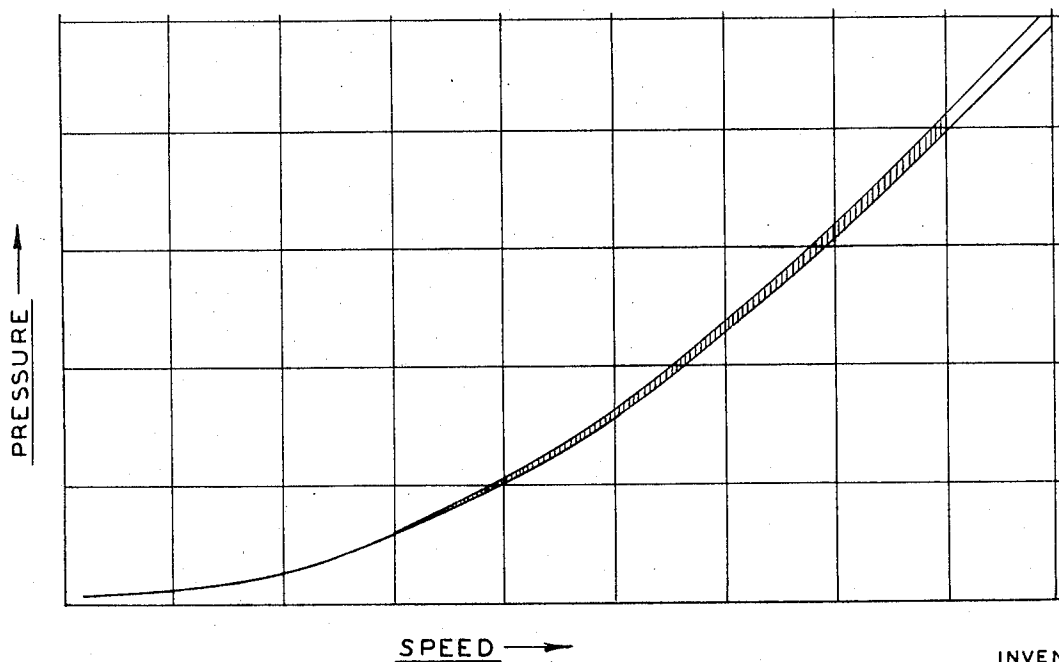


FIG.15.



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FIG. 21.

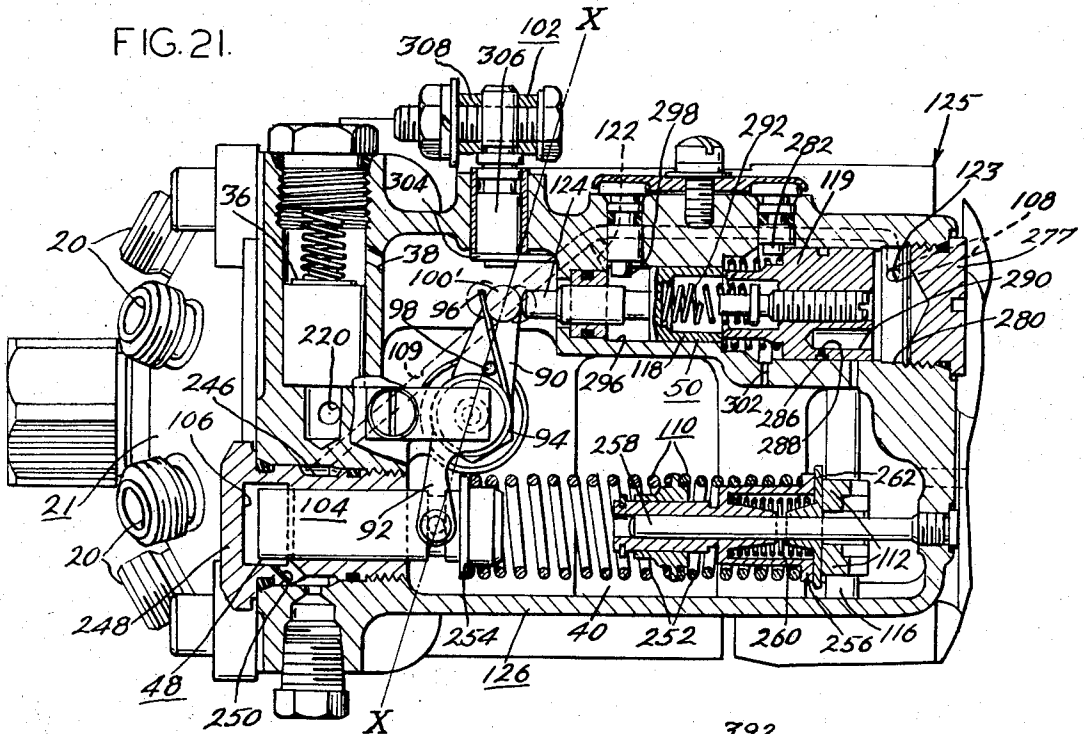


FIG. 22.

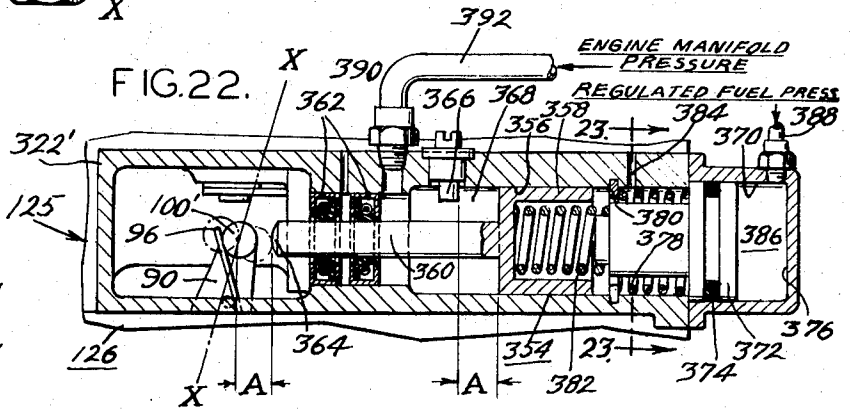


FIG. 23.

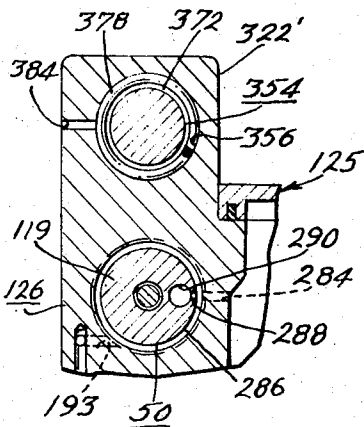
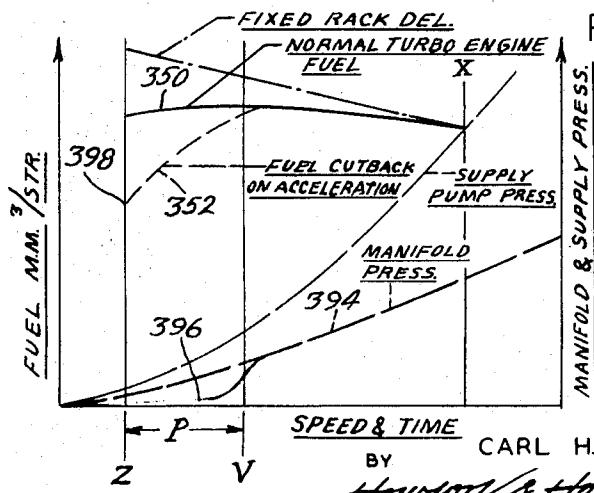


FIG. 24.



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FUEL INJECTION PUMP

The present invention relates generally to fuel injection pumps for internal combustion engines and relates more particularly to a novel single-plunger distributor-type fuel injection pump having a built in variable speed hydraulic governor including hydraulic torque control and excess fuel devices.

Diesel engines, due to the nature of their operation, require a governor to regulate the injected fuel to prevent the engine from stalling or overspeeding. The governor, which is associated with the fuel injection pump, senses changes in engine speed and regulates the output of the fuel injection pump accordingly, increasing fuel delivery with increased load and decreasing fuel delivery with a decreased load to maintain a relatively constant speed at a given throttle setting.

There are essentially three types of governors; constant speed, minimum-maximum speed, and variable speed governors. The constant speed governor maintains the engine at a fixed speed regardless of load and is generally employed in stationary engine installations such as air compressors and the like. The minimum-maximum speed governors are utilized essentially for vehicle operation to prevent stalling at low speeds and racing at high speeds, the intermediate speeds being controlled only by the vehicle operator. The variable speed-type of governor, to which the present invention is directed, provides automatic fuel control over the entire engine speed range. This type of governor is customarily used with locomotives, tractors, marine engines, etc. and is also often used in vehicles such as trucks. The most common type of variable speed governor is the mechanical type employing centrifugal weights acting against one or more springs which is coupled to the fuel control mechanism of the pump. Pneumatic, hydraulic, and mechanical-hydraulic and electrical governors have also been employed for the same purpose.

The present invention includes a hydraulic governor which is integrally built into the fuel injection pump. Since the pressure output of the engine-driven fuel pump is generally proportional to the square of the engine speed, the pump pressure can be utilized as an indication of engine speed to control a hydraulic governor. The pump pressure is, however, dependent on the fuel viscosity as well as the engine speed and hence the fuel temperature will have an effect on the governor operation. In addition, the wear of the pump with age will result in a gradual dropping of the pressure produced at a given speed, adding another factor to the inaccuracy of the governor operation.

In the present invention a centrifugal regulator is employed to provide a control fuel pressure which varies only with changes in engine speed, being unaffected by oil viscosity change or supply pump wear. The invention further includes a hydraulic torque control associated with the governor and operated by the same regulated control fuel pressure for limiting fuel delivery at any given speed to a predetermined maximum amount. Also operated by the regulated control fuel pressure in the present invention is a timing device for advancing or retarding the fuel injection with respect to the engine cycle in accordance with the engine speed. An automatic hydraulic excess fuel device is coupled with the hydraulic torque control and provides the necessary excess starting fuel required during engine cranking.

It is accordingly a primary object of the present invention to provide a fuel injection pump having a built-in hydraulic governor adapted to sense engine speed by reference to a control fuel pressure.

A further object of the invention is to provide a fuel injection pump having a hydraulic governor as described and including a centrifugal regulator for regulating the fuel pressure from an engine-driven supply pump to provide a control fuel pressure which is an accurate indication of engine speed.

Another object of the invention is to provide a fuel injection pump having a hydraulic governor as described and including an associated hydraulic torque control operated by the regulated fuel pressure to control maximum fuel delivery throughout the operating speed range of the engine.

Still another object of the invention is to provide a fuel injection pump as described including an excess fuel starting device coupled with the torque control and hydraulically actuated to provide the necessary excess fuel required during cranking of the engine.

A still further object of the invention is to provide a fuel injection pump as described having a fuel injection timing device which is hydraulically actuated by the regulated control fuel pressure.

Another object of the invention is to provide a fuel injection pump as described of a compact construction wherein the governor, fuel supply pump, torque control, excess starting device and injection timing device are built integrally into the pump housing.

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

FIG. 1 is a schematic view showing a fuel injection pump in accordance with the present invention with the hydraulic head, pumps and control components separated to clearly show the fuel flow paths, the governor components being shown in the full load rated speed operation positions;

FIG. 2 is a view showing the schematic governor and torque control assembly of the pump as shown in FIG. 1 but in the no load, low idle position;

FIG. 3 is a view similar to FIG. 2 showing the schematic hydraulic governor and torque control in the excess fuel cranking position;

FIG. 4 is a plan view of the fuel injection pump embodiment schematically illustrated in FIG. 1;

FIG. 5 is a side elevational view of the pump shown in FIG. 4;

FIG. 6 is an end elevational view of the pump shown in FIGS. 4 and 5;

FIG. 7 is a sectional view taken along line 7-7 of FIG. 4 showing details of the interior of the pump including the hydraulic head;

FIG. 8 is a sectional view taken along line 8-8 of FIG. 7 showing details of the centrifugal regulator, supply pump and auxiliary pump;

FIG. 9 is a sectional view taken along line 9-9 of FIG. 7 showing details of the pump timing device;

FIG. 10 is a sectional view taken along line 10-10 of FIG. 7 showing details of the regulator, governor and torque control with the engine in a full load condition;

FIG. 11 is a sectional view taken along line 11-11 of FIG. 8 showing details of the auxiliary fuel pump;

FIG. 12 is a sectional view taken along line 12-12 of FIG. 8 showing details of the centrifugal regulator;

FIG. 13 is a perspective view showing the regulating valve element of the centrifugal regulator;

FIG. 14 is a graph showing fuel delivered versus engine speed over the operating speed range of the engine at full load illustrating the regulating effect of the torque control device;

FIG. 16 is an end view of a slightly modified form of fuel injection pump in accordance with the invention;

FIG. 17 is a sectional view taken along line 17-17 of FIG. 16;

FIG. 18 is a sectional view taken along line 18-18 of FIG. 16;

FIG. 19 is a sectional view taken along line 19-19 of FIG. 18;

FIG. 20 is a graph showing the full load fuel delivery curve obtainable with the modified embodiment of FIG. 16-19;

FIG. 21 is a view similar to FIG. 17 showing a further modified form of the invention;

FIG. 22 is a view similar to FIG. 18 of the further modified form of the invention;

FIG. 23 is a sectional view similar to FIG. 19 taken along line 23-23 of FIG. 22; and

FIG. 24 is a graph showing the fuel delivery characteristics obtained with the modified embodiment of FIGS. 21-23.

Referring to the drawings and particularly FIG. 1 thereof wherein the automatic control components of a fuel injection pump embodying the present invention are schematically illustrated, the fuel discharge outlets 20 of the hydraulic head 21 of the single-plunger distributor-type pump are connected with a plurality of fuel injection nozzles 22 of an internal combustion engine in the conventional manner. Fuel flows to the pump hydraulic head from tank 24 through conduit 26, passing through a primary filter 28, supply pump 30, final filter 32 and conduit 33. A double-acting check valve 34 permits a priming flow of fuel around the pump 30 and also permits a recycling of fuel around the pump in the event of clogging of the fuel line or filter downstream of the pump. The fuel pressure within the hydraulic head fuel sump is regulated by the pressure regulating valve 36 which spills excess fuel through conduit 38 into the drain sump 40 from which it returns through conduit 42 into tank 24.

An engine-driven auxiliary pump 44 of relatively low capacity is connected by conduit 46 to the hydraulic head fuel sump and serves to supply a control fuel pressure to a hydraulic governor 48, torque control 50, and timing device 52, the control pressure being regulated to vary only with change in engine speed by the centrifugal regulator 54. The regulator 54 receives fuel from the pump 44 through conduit 56 and directs the fuel into the center of a rotating valve assembly 58 which rotates at a speed corresponding to the engine speed. The valve assembly 58 includes a pair of weights 60 pivotally mounted thereto at 62 and adapted to actuate a valve element 64 axially with respect to valve passage 66 from which the fuel passes into the regulator chamber 67. At any given speed, the weights 60 exert a fixed force urging the valve element 64 toward the pressurized fuel flow exiting through valve passage 66. Since the valve element will move toward or away from the passage 66 until the weight force is balanced by the fuel pressure, the fuel pressure at a given speed will always be constant regardless of changes in viscosity of the fuel and regardless of the gradual wear and decreased capacity of the pump 44. The fuel passing into the regulator chamber 67 is drained through conduit 68 into the drain sump 40 and thence into the tank 24 through conduit 42.

The graph of FIG. 15 shows the regulated control pressure as a function of engine speed. The shaded area of the graph shows the slight variation which may occur in the control pressure due to the effect of changes of viscosity of the fuel which causes a shifting of the center of gravity of the weights. Thus at a given speed with a change of fuel viscosity the position of the weights will necessarily change and in a relatively small regulator as is required for internal mounting in a fuel injection pump, the change of weight position will affect the pressure regulation of the regulator to a slight degree insufficient to affect the operation of the several hydraulically actuated control components.

The fuel at the regulated control pressure enters the timing device 52 through conduit 70, check valve 72 and conduit 73 and acts against piston 74 in the chamber 76 against the force of compression springs 78 to position the piston in accordance with the engine speed. Arm 80 extending from the piston engages a slot 81 in the roller support member 82 carrying the plunger actuating rollers 84 the function of which will be presented in detail below. In brief, the rotation of the support member 82 by the movement of the piston 74 advances or retards the fuel injection in accordance with the engine speed. To permit the piston 74 to move against the fluid pressure under the influence of spring 78, leakage is permitted around the piston and drains through conduit 86 into the drain sump 40 and thence into tank 24.

The hydraulic governor 48 and the torque control 50 having the excess starting fuel device coupled therewith are each actuated by the regulated control pressure from the pump 44 and are each adapted to control the fuel quantity delivered by the pump hydraulic head 21 by controlling the rotational position of the fuel control shaft 88 which as described in detail below has an eccentric pin on one end thereof adapted to posi-

tion the fuel control sleeve on the pump plunger. As shown in FIG. 1, the control shaft 88 includes a control arm 90 secured to the shaft extending toward the torque control unit 50 and also a lever 92 rotatably mounted on the shaft 88 connected with the governor 48 and rotationally influencing the shaft 88 by means of the attached coil spring 94, the outer end 96 of which extends between the spring stop 98 and the control arm post 100. With this arrangement, which will be more clearly understood with reference to FIG. 7 and the discussion thereof presented herebelow, the control shaft 88 will turn in response to movement of the lever 92 connected with the governor mechanism, but the lever 92 can be overridden by movement of the control arm 90 either by means of the torque control 50 or the shutoff lever 102 each of which are adapted to engage the end post 100 of the control arm 90.

In view of the small scale and partly schematic showing of the governor 48 and fuel control 50 with its associated excess fuel device, these mechanisms will not be discussed in detail at present. However, to briefly summarize their structure and operation, it will be noted that the hydraulic governor includes a governor piston 104 slidable in chamber 106 into which fuel at the regulated control pressure passes through conduits 108 and 109. The lever 92 is connected to the piston 104 for actuation thereby. A governor spring assembly 110 bears against the piston 104 at one end and against an arm 112 of the operating lever 114 at the other end, the operating lever being pivoted at 116. With this arrangement, the position of the lever 92 and hence the fuel quantity injected by the fuel pump is determined by the position of the operating lever and additionally by the speed at which the engine is running, the latter factor changing the control pressure in the chamber 106 and hence the position of the piston 104. As indicated by the alternate broken line positions of the operating lever and the lever 92, the control shaft 88 and lever 92 move counterclockwise to a decreased fuel delivery position, and clockwise toward a maximum fuel delivery position.

The hydraulic torque control 50 which also incorporates the excess fuel device consists of torque control piston 118 and excess fuel piston 119 slidable in stepped bore 120, into separate chambers of which the regulated control fuel pressure is introduced through conduits 122 and 123. Since the details of construction and operation of the torque control and associated excess fuel device are more suitably presented below in connection with the detailed description of the invention, it will at present suffice to indicate the position of pistons 118 and 119 are dependent on the regulated fuel pressure and that the extending nose portion 124 of piston 118 will during starting of the engine and full load operation thereof engage the end post 100 of control arm 90 to limit clockwise rotation of the arm and hence limit the maximum fuel delivery of the pump.

Summarizing the schematic presentation of the invention, an auxiliary supply pump of relatively low capacity provided with a centrifugal regulator delivers a control fuel pressure which varies only in response to variation in engine speed. This pressure is utilized to control a hydraulic governor and an associated hydraulic torque control device which are effective over the operating range of the engine to control fuel delivery. In addition, the regulated control pressure is utilized to control the fuel injection timing by means of a hydraulic timing control. In the detailed embodiment of the invention described below, all of the recited control components are integrally incorporated into the fuel injection pump housing in an extremely compact arrangement.

Considering the specific embodiment of the invention shown in FIGS. 4-13, the pump 125 is enclosed within an irregularly shaped housing 126 having a mounting flange 128 at one end thereof permitting the bolting of the pump to an internal combustion engine (not shown). Pump drive shaft 130 extends perpendicularly to the mounting flange 128, having a projecting end 132 adapted for gear drive attachment to the engine and extending interiorly into the pump housing within bore 134 thereof. The drive shaft is journaled within the hous-

ing by radial bearings 136 and 138 and thrust bearing assembly 140 and is sealed adjacent the mounting flange by seals 141 as shown most clearly in FIG. 7. The inner end 142 of the drive shaft is splined to receive the face cam 144 thereon in axially reciprocable relation. The cam surface 146 of the face cam 144 rides on the four rollers 84 journaled on the rotatably adjustable roller support member 82 which is radially supported by the circumferential bearing surface 148 of the housing. Rotation of the drive shaft can accordingly be understood to provide both a rotation and, due to the cam surface contact with the rollers 84, a reciprocation of the face cam 144. The face cam includes six equally spaced cam lobes which engage the rollers arranged in pairs 60° apart to provide six reciprocating strokes per revolution as required for a six-cylinder pump.

The rotating and reciprocating face cam 144 is employed to drive the pumping and distributing plunger 150 axially aligned therewith in the hydraulic head 20 of the pump. The hydraulic head 20 is fitted within the stepped bore 152 at the opposite end of the housing 126 from the housing flange 128, being sealed at each end thereof by seal rings 154 and 156. The plunger 150 is disposed for rotating and sliding movement within an axial bore 158 in the hydraulic head. The inner end 160 of the plunger is connected to the face cam 144 by a coupling assembly 161 which includes spring guide 162 against which is seated the compression spring 164. The other end of the compression spring bears against the hydraulic head, passing over the spring clip 166 thereon. The spring 164 serves to hold the face cam 144 against the rollers 84 and thus effects the return stroke of the pumping and distributing plunger as will be described herebelow.

The outer or pumping end 168 of the plunger 150 forms in the bore 158 a fuel distribution chamber 170 in conjunction with a check valve 172 mounted in the outer end of the bore. When the plunger is retracted, fuel is admitted to the fuel distribution chamber 170 through a fuel port 174 communicating with fuel passage 176 leading to the peripheral fuel sump 178 created by a reduced diameter portion of the hydraulic head and extending substantially between the seal rings 154 and 156. Fuel at the supply pressure is introduced to the sump 178 through fuel port 180 from the supply pump 30 which will be presently described.

Upon rotation and reciprocation of the plunger 150, fuel introduced into the fuel distribution chamber from the sump 178 upon closing of the port 174 by the plunger is compressed and forced through the check valve 172 into passages 182 and 183 of the hydraulic head and into the annulus 184 of the plunger which connects with the plunger distributor slot 186. The distributor slot during each pumping stroke of the plunger indexes with one of the plurality of outlet passages 187 in the hydraulic head leading to one of the fuel discharge outlets 20. As indicated schematically in FIG. 1, suitable conduits connected with the discharge outlets 20 connect with the fuel injection nozzles in the engine.

Control of the amount of fuel delivered per pumping stroke is effected by the fuel control sleeve 188 slidably mounted on the plunger in the spill sump 190 which opens into the sump 178. An axial bore 192 in the plunger opening into the fuel distribution chamber 170 communicates with a cross bore 194 which is covered by the sleeve 188 during the pumping phase of the plunger stroke. When the cross bore 194 moves beyond the sleeve, the pressure in the fuel distribution chamber 170 is reduced to the sump pressure permitting the closing of the valve 172 to end the fuel injection. The sleeve 188 is axially positioned in the sump 190 as indicated above by eccentric pin 196 on the lower end of the control shaft 88 which extends into the sump 190. The pin 196 is engaged in a peripheral slot 198 in the sleeve 188. Rotation of the control shaft 88 in the manner indicated above and as will be described hereinafter thus regulates the fuel quantity delivered per pumping stroke by the fuel injection pump.

The fuel pumps 30 and 44 shown in the schematic view of FIG. 1 are built integrally into the fuel injection pump as

shown in FIG. 8. A pump drive shaft 200 transversely journaled within the housing 126 beneath the drive shaft 130 is rotationally driven thereby through spiral gears 202 and 204 on the respective shafts. At one end of the shaft 200 is located the fuel supply pump 30 in pump chamber 206 of the housing, pump 30 being a gear-type pump and comprising pump gear 208 driven by the shaft 200 and gear 210 meshing therewith rotating on stub shaft 212. Fuel input to the fuel injection pump from the tank 24 enters the housing 126 from conduit 26 through port 214 shown in FIGS. 6 and 8 and is led into the pump 30 through fuel passage 216. The output of the pump 30 passes through a passage 217 and exits from the housing through port 218 from which it passes into the final filter 32 as shown in the schematic drawing of FIG. 1. The double-acting check valve 34 as shown in FIG. 8 communicates at one end with the port 214 and at the other end with the port 218 and permits a priming flow of fuel around the pump 30 while also permitting a safety return of fuel pumped in the event of downstream blockage of the fuel channels. Fuel from the filter 32 passes through conduit 33 into the fuel inlet port 180 to direct fuel into the sump 178 of the hydraulic head. The sump fuel pressure is regulated by the pressure regulating valve 36 which is connected with the sump 178 by the fluid passage 220 as shown in FIGS. 7 and 10. The fuel spilled by the pressure regulating valve 36 passes through conduit 38 into the drain sump 40 in the housing interior.

The auxiliary fuel pump 44 as shown in FIG. 8 is located at the opposite end of the shaft 200 from the pump 30 and is of an almost identical construction although of a smaller capacity. As shown in FIG. 8 and in FIG. 11, the pump 44 includes a pump gear 222 secured to the shaft 200 and a meshing pump gear 224 rotatably mounted on stub shaft 226. The pump gears 222 and 224 rotate in the pump chamber 228 in the housing 126. Fuel is led into the pump 44 through inlet port 230 shown in FIG. 11 from the hydraulic head sump 178 through conduit 46 shown only in the schematic FIG. 1. The output of pump 44 passes through outlet port 232 with which communicates the conduit 56 leading to the centrifugal regulator 54.

The regulator 54 as shown most clearly in FIGS. 8, 10 and 12, includes a vertical shaft 234 rotatably mounted in bearing assembly 236 in the pump housing. At its lower end, the shaft 234 includes a spiral gear 238 meshing with the gear 202 on the pump drive shaft 130. At its upper end, the shaft 234 carries a rotor 58 to which the weights 60 are rotatably secured at 62 as described above with reference to FIG. 1. The shaft 234 includes an axial bore 240 which provides a fluid flow path from radial ports 242 to the radial valve passage 66 at its upper end. A port arrangement 244 in the bearing assembly 236 permits fuel flow from the conduit 56 into ports 244 and central bore 240 of the shaft 234. The fuel passes through the valve passage 66 and is directed against the valve element 64 bearing against the arms of the weights 60. The fuel spilled by the regulator passes into regulator chamber 67 and thence through the conduit 68 into the drain sump 40 as shown most clearly in FIG. 7. In a well-known manner, the weights 60 upon rotation of the shaft 234 under the influence of centrifugal force provide a force opposing the flow of pumped fuel which varies only with changes in the speed of the engine, thereby positioning the valve element so as to regulate the pump output pressure to correspond with engine speed. The regulator is not affected by changes in the viscosity of the fuel since it is responsive only to the fuel pressure.

The regulated pressure from the pump 44 as shown in FIG. 8 passes from the pump outlet port 232 through passage 70 into check valve 72 from which it passes through conduit 73 into chamber 76 of the timing device 52. The timing device, which is seen most clearly in FIGS. 7 and 9, as described above with respect to the schematic showing of FIG. 1, includes the piston 74 slidably disposed in the piston chamber 76 and adapted to be axially positioned by the fuel pressure force from the pump 44 and by the springs 78 acting in opposition to the fuel pressure at the opposite end of the chamber.

The piston by means of the arm 80 extending into slot 81 of the roller support member 82 provides a rotational adjustment of the roller support member position in accordance with the regulated control pressure and hence the speed of the engine, thereby establishing a timing of the plunger reciprocation which varies in accordance with the engine speed. Ample fuel leakage is permitted around the piston 74 to permit a prompt return of the piston under the influence of the spring 78, the leakage being bled from the spring end of the piston chamber through the passage 86 into the low fuel drain sump 40. As indicated in the schematic view of FIG. 1, the fuel in the housing drain sump 40 is returned therefrom to the fuel tank through overflow conduit 42 connected to port 41 in the top of the housing.

The pressure regulated fuel from the auxiliary pump 44 also passes from the outlet port 232 of the pump through fuel passage 108 and 109 to the hydraulic governor 48 which is located in the upper part of the housing 126. The conduit 109 opens into peripheral chamber 246 in the governor plug 248 threaded into the housing and thence through port 250 into the piston chamber 106 therein. The regulated fuel pressure acts in the piston chamber against the piston 104 slidable within the plug 248, the extending end of the piston being connected as described above to the lever 92 of the fuel control unit.

The governor spring assembly 110 acting axially against the piston 104 in opposition to the fuel pressure force in chamber 106 includes a main governor spring 252 extending between shoulder 254 of the piston and the spring guide assembly 256 slidable on the support rod 258 secured to the housing. An idling spring 260 on the support rod bears against the guide assembly 256 and bears at its opposite end against the spring guide 262 slidable on the support rod 258. The spring assembly 110 is actuated by the operating lever 114 pivotally mounted on shaft 116 journaled in the housing 126. The forked arm 112 of the operating lever extending from the shaft 116 engages the spring guide 262 and by adjusting the axial position of the guide on support rod 258 determines the base position from which the springs 252 and 260 act against the piston 104. The operating lever 114 is, aside from the shutoff lever 102, the only external control element governing the fuel pump operation and is connected in the usual manner to the engine throttle control. The arc of travel of the operating lever 114 is limited by the adjustable excess fuel stop 264 at one extreme and the idling stop 266 at the other extreme, said stops respectively being secured to the pump housing and comprising bolts 267 extending through brackets 268 and having lock nuts 269 thereon.

The hydraulic governor as indicated above adjusts the position of the governor piston 104 in accordance with the speed of the engine and the setting of the operating lever 114 and accordingly through the lever 92 connected with the fuel control shaft 88 adjusts the position of the fuel control sleeve 188. The fuel control shaft 88 which has been described in some detail above, is journaled in the bushing 270 in the housing 126 with the control arm 90 seated on the flanged upper end of the bushing. The shaft 88 is held axially in position by the stop plate 272 which is bolted to the housing.

The torque control 50 which is shown most clearly in FIG. 10, is as indicated above coupled with the excess fuel device both of which are located in the upper portion of the housing within the stepped bore 120. The bore 120 includes a relatively large diameter portion 274 which necks down to a smaller diameter portion 276 and is closed at one end by the plug 277. The excess fuel piston 119 is slidably disposed in the larger diameter portion 274 of the bore and is biased by spring 278 bearing against shoulder 279 of the bore toward the plug 277. The piston 119 is urged in the opposite direction by the regulated fuel pressure from the auxiliary pump 44 which is directed into the chamber 280 between the piston 119 and plug 277 through conduit 123. Since the spring 278 is relatively weak, the pressure developed in the chamber 280 when the engine is running at even idling speed is sufficient to urge the

piston against an adjustable eccentric stop 282 extending through the sidewall of the housing. When the engine is stopped, the force of the spring 278 moves the piston 119 against the falling pressure in the chamber 280 against the plug 277 at which time the chamber 280 is opened to the housing drain sump 40 by means of port 284 which aligns with circumferential groove 286 in piston 119 which in turn through restricted port 288 and bore 290 opens into chamber 280. Movement of the piston toward the stop 282 is prevented during the cranking of the engine since the pressure developed in chamber 280 during engine cranking is insufficient to move the piston against spring 278 because of the leakage through the restricted port 288. The manner in which piston 119 serves as an excess fuel device will be presently clear from the further description of the torque control unit 50 and the operation of the pump.

The torque control piston 118 is slidably disposed in the smaller portion 276 of the bore 120 and is biased by an internal spring 292 away from the piston 119. The spring 292 is seated at one end against the piston 118 and at the other end against the adjustable spring stop 294 threaded within a coaxial bore of the excess fuel piston 119. The piston 118 includes an axially extending nose portion 124 which extends through the annular seal 295 in the housing. An annular chamber 296 is formed between the seal 295 and the major diameter of the piston into which pressurized fuel from the pump 44 at the speed regulated pressure is introduced through passage 122. The fuel pressure in chamber 296 acts against the force of spring 292 to urge the torque control piston 118 toward the piston 119. An adjustable eccentric stop 298 extending into the chamber 296 provides a limit for movement of the piston 118 toward the seal ring 295. The eccentric adjustable stops 282 and 298 are locked and covered by the cap plate 299 secured to the outside of the housing by screw 300.

The chamber surrounding the spring 278 as well as the spring 292 are vented to the drain sump 40 by the vent port 302 in the housing. The nose portion 124 of the piston 118 is so aligned as to contact the end post 100 of the control arm 90 and serves as a stop to limit rotation of the arm in the direction of increased fuel delivery during certain phases of the engine operation as will be presently described. The end post 100 of control arm 92 is also engageable by the arm 304 of the shutoff lever 102 as may be seen in FIG. 7, the arm extending from the shutoff lever shaft 306 extending through the side of the housing and connected with the exterior lever arm 308 as seen in FIG. 8.

For operation, the pump 125 is mounted on an internal combustion engine by means of the flange 128 and the shaft 130 is appropriately connected to the engine drive shaft to rotate at one-half engine speed for a four-stroke cycle, six-cylinder engine. The fuel discharge outlets 20 are connected with the engine fuel injection nozzles in the conventional manner. To start the engine, the shutoff lever 102 is set at the on position permitting movement of the control arm 90 in response to the governor as limited by the torque control device. With the pump primed by a suitable priming device (not shown) to fill the sump 178, cranking of the engine is begun with the operating lever set at the excess fuel position shown in broken lines in FIG. 1, and also in FIG. 3. The cranking of the engine provides insufficient pressure in the chamber 280 to move the excess fuel piston 119 against the force of spring 278 and the piston 119 accordingly remains during cranking in the position shown in FIG. 3 against the plug 277. This permits the torque control piston 118 and its nose portion 124 to move back to the retracted excess fuel position illustrated to allow the control arm 90 to move to an excess fuel delivery position. The excess starting fuel delivered at the cranking speed is shown in the graph of FIG. 14.

Upon starting of the engine, the regulated fuel pressure from the pump 44 to chamber 280 is sufficient to move the excess fuel piston 119 against the stop 282 as shown in FIG. 2. Prior to the application of load to the engine, the operating lever is moved to the low idle position illustrated which allows

the piston 104 and the control shaft 88 to move to a low fuel delivery position. Because of the relatively low pressure produced by the pump 44 at low engine speed, the torque control piston 118 at low idle is advanced under the influence of the spring 292 against the stop 298 in view of the low fuel pressure in the chamber 296. The end post 100 of the control arm 90 is in the low idle position substantially spaced from the nose portion 124 of piston 118 since the torque control piston serves as a limit to the fuel delivery position of the control arm 90 only at the full load and excess fuel positions of the operating lever. The torque control piston 118 will remain in the position against the eccentric stop 282 until the engine is stopped and thus serves as a base from which the torque control piston 118 is positioned by the spring 292 against the force of the fuel in chamber 296.

The present fuel injection pump with the exception of the fuel governing controls operates in a manner similar to that of the conventional single-plunger distributor-type pumps. The rotation of the drive shaft 130 by the engine, for example at one-half engine speed for a four-cycle engine, provides a rotation and a reciprocation of the plunger 150 as the face cam 144 moves across the rollers 84. Fuel pumped from the pump 30 into sump 178 enters the fuel distribution chamber 170 when the fuel port 174 is open and is pumped from the fuel distribution chamber through check valve 172, passages 182 and 183 into annulus 184 of the plunger. The plunger distributor slot 186 connects the annulus 184 with one of the outlet passages 187 during each pumping stroke of the plunger and fuel is thus pumped to the discharge outlets 20 in a sequential manner.

The timing of the fuel injection is automatically controlled by the hydraulic timing device 52. As viewed in FIG. 9, the shaft 130 and plunger 150 rotate in a clockwise direction and, upon an increase in engine speed producing a corresponding increase in the regulated fuel pressure of the pump 44, the piston 74 is moved to the right, causing a counterclockwise rotation of the roller support member 82 thereby advancing the timing of the fuel injection with respect to the engine cycle. Upon lowering of the engine speed, the output pressure of the pump 44 is correspondingly lowered and the piston 74 moves to the left under the influence of the springs 78 to move the roller support member 82 in a counterclockwise direction. The face cam 144 then reaches the rollers at a slightly later point in the engine cycle thereby retarding the injection of the fuel.

The manner in which the quantity of fuel is regulated by the fuel control sleeve 188 as positioned by the control shaft 88 has been described above. Since a certain amount of excess capacity is built into the supply pump 30, there will during pump operation be a constant flow of fuel from the sump 178 through the pressure regulating valve 36, the fuel flowing into the drain sump 40 and back to the tank through conduit 42. When the shutoff lever 102 is moved to the off position, the sleeve 188 as viewed in FIG. 7 is moved to the extreme right permitting the cross bore 194 to be exposed to the sump 190 throughout the pumping stroke of the plunger, thereby preventing any pumping of fuel from taking place and causing the engine to stop.

During the loaded operation of the engine provided with the present fuel injection pump, the hydraulic governor 48 functions to maintain a substantially uniform engine speed for a given operating lever setting. As the engine speed falls off at a given operating lever setting due to an increase in load, the pressure in chamber 106 drops off permitting the piston 104 under the influence of spring assembly 109 to move the lever 92 to increase the fuel delivered by the pump. On the other hand, should the load decrease permitting the engine speed to increase, the pressure in chamber 106 will increase moving the piston 104 against the spring assembly 109 and, through the lever 92, decreasing the fuel delivered to maintain a relatively constant speed.

In the absence of the torque control device 50, the fuel injection pump would characteristically provide a full load fuel

delivery as indicated by the broken line 310 in the graph of FIG. 14. Because of the high fuel requirements necessary at cranking speeds, the full load fuel delivery at normal operating speeds is substantially in excess of that required by the engine for efficient operation. In the absence of a torque control device, the excess fuel would appear as smoke in the engine exhaust and would represent a substantial waste in fuel consumed. The torque control device 50 eliminates this waste by limiting the full load fuel delivery as illustrated in the curve 312 of the graph of FIG. 14 which should approximate the fuel required by the engine for most efficient operation at full load over its operating speed range.

To set the torque control 50 to the desired full load fuel curve 312, the eccentric stop 282 is adjusted to provide a stop position for the excess fuel piston 119 which, with the torque control piston 118 bearing thereagainst under the influence of pressure in chamber 296, will position the piston nose 124 to stop the control arm 90 allowing a maximum fuel delivery at rated engine speed as indicated by the point 314 in the graph of FIG. 14. The desired fuel delivery at the low end of the operating range of the engine as indicated by point 316 on the full load fuel curve is obtained by adjusting the eccentric stop 298 against which the torque piston 118 bears when fully extended under the influence of the spring 292. The pressure in the chamber 296 will fall as the engine speed drops and the tension of spring 292 should be adjusted by means of the adjustable spring stop 294 so that the piston 118 will engage the stop 298 at the desired minimum operating speed. With this arrangement, by virtue of the spring characteristic, the full load fuel delivered will gradually decrease with a decrease in engine speed to closely approximate the most efficient fuel consumption at full load over the engine operating range. As indicated above, when the torque control device overrides the governor in establishing the rotational position of the control shaft 88, the coil spring 96 is compressed permitting the control arm 90 and control shaft 88 to stop rotating when the post 100 of the control arm engages the nose 124 of the torque control piston. The lever 92 continues to rotate to the position dictated by movement of governor piston 104 although without any effect on the fuel delivery.

Referring to FIGS. 16—19, a slightly modified fuel injection pump embodiment is illustrated. The pump of this embodiment is identical with that previously described with the exception of the torque control device which is modified to provide a full load fuel delivery curve of the type shown in FIG. 20. This curve from the lowest practical engine operating speed Z rises to a midspeed Y and then drops to the maximum rated speed X. With the torque control device previously described, due to the fact that the overriding effect of the torque control is operative only to decrease the fuel delivery otherwise effected by the hydraulic governor, a full load fuel curve of the type shown in FIG. 20 could not be provided.

To obtain the rising and falling maximum fuel delivery curve of FIG. 20, which for example might be required for a constant horsepower engine, the modified embodiment of FIGS. 16—19 includes in addition to the torque control device 50 previously described, a second spring-loaded piston assembly 320 parallel to and positioned directly above the device 50. The assembly 320 is actuated by the regulated fuel pressure and engages an elongated end post 100' of the control arm 90 to effect the desired full load fuel control. As shown particularly in FIGS. 18 and 19, the assembly 320 includes a cylinder 321 formed within the housing extension 322 above the device 50 previously described. Within the cylinder 321 a piston 324 is slidably disposed and provided with a plunger 326 extending through a bore 328 in the end of the cylinder. The plunger 326 is aligned with the elongated post 100' of the control arm to permit engagement of the plunger tip 330 therewith under certain operating conditions as described below. A spring 332 in the cylinder 320 biases the piston 324 and plunger 326 away from the post 100'. The movement of the piston under the influence of the spring is limited by the screw stop 333 which is threadably engaged in

the end 335 of the cylinder. Fuel at the regulated control pressure is introduced into the chamber 334 of cylinder 320 adjacent the cylinder end 335 by conduit 336 extending from the conduit 123.

The operation of the modified torque control shown in the embodiment of FIGS. 16—19 is best explained with reference to the desired full load fuel delivery curve of FIG. 20. Between the lowest normal operating speed Z and a midrange speed Y, the desired curve 338 shows an increase in fuel delivery with increasing speed. This portion 338 of the full load delivery curve is controlled by the torque control device 50 described in conjunction with the preferred embodiment of the invention. As shown in FIG. 17, the device 50 is operable to decrease the fuel delivery upon a decrease of speed from the speed Y down to the speed Z, the piston 118 having a range of axial movement R sufficient to effect such a decrease. As the speed regulated pressure falls in the chamber 296, the spring 292 advances the piston 118 against the post 100' of control arm 90 as described above to effect a reduction in the delivery fuel, overriding the hydraulic governor. At speed Y and higher speeds, the pressure in the chamber 296 is sufficient to hold the piston 118 against the excess fuel piston 119 as shown in FIG. 17.

The piston assembly 320 serves to control the full load fuel delivery between the speeds X and Y and is thus adapted to increase the fuel delivery upon decrease of speed in this range as indicated by portion 340 of the full load curve. The spring 332 of the assembly 320 is chosen so as to equal the force developed by the supply pump pressure in the chamber 334 at speed Y when the piston is against the screw stop 333. As the engine speed increases above speed Y, as shown in FIG. 20, the supply pump pressure also increases causing the piston 324 to move away from the stop 333 and causing the plunger 326 to move the post 100' toward a lower fuel delivery position. At the rated speed X, the maximum fuel delivery is reduced by the assembly 320 to the amount 342.

From the above it can be understood that the torque control device 50 in conjunction with the assembly 320 serve over the entire operating speed range Z to X of the engine to control the full load fuel delivery of the engine. From speed Z to Y the control 50 by engaging the post 100' of the control arm establishes the maximum fuel delivered, while between the speeds Y to X, the assembly 320 controls the full load fuel by engagement with the post 100'. The control 50 and assembly 320 override the control arm setting established by the hydraulic governor 48 only at full load. Under operating conditions other than full load, the post 100' of the control arm 90 is not engaged by the plunger of either the control 50 or assembly 320.

A further modified embodiment of the invention is shown in FIGS. 21—23. This modification is similar to that shown in FIGS. 16—20 in that it is directed to an arrangement supplementing the torque control device 50 to regulate the pump full load fuel delivery. The modification of FIGS. 21—23 is specifically adapted for use with a so-called "turbo" engine wherein the engine air charging blower or supercharger is driven by an exhaust gas turbine. Specifically, the modification is directed to the problem of excess smoking of the turbo engine upon rapid acceleration at low speeds. The smoking is caused by the momentary lack of excess air provided during rapid acceleration due to the low speed of the exhaust driven blower.

In order to reduce smoking upon rapid low speed acceleration, the fuel delivery is momentarily cut back by the modification of FIGS. 21—23. As shown in FIG. 24, the normal turbo engine full load fuel delivery curve 350 controlled by the torque control device 50 as described above is a substantially horizontal line on the fuel-speed graph. To reduce smoking, the maximum fuel delivery on rapid acceleration of a turbo engine is desirably cut back below that permitted by torque control device 50 between the lowest normal operating speed Z and a relatively low speed V at which the exhaust turbine turns at an adequate speed to supply the required air to

the engine. A suitable maximum fuel delivery curve for rapid acceleration between speeds Z and V is indicated by the broken line 352 of the graph of FIG. 24.

The modification of FIGS. 21—23 comprises essentially the pump and control elements shown in the preferred embodiment of FIGS. 1—15 with the addition of an engine manifold pressure operated override device 354. The override device 354 is mounted in an extension 322' of the pump housing above the torque control device 50 and includes a cylindrical bore 356 in the housing extension 322' in which is slidably disposed a piston 358 having a plunger 360 extending axially therefrom. The plunger 360 extends through seals 362 at one end of the bore 356 and terminates in a tip 364 proximate the post 100' of control arm 90 for engagement therewith under certain operating conditions of the engine. An adjustable eccentric stop 366 extends into the chamber 368 formed between the piston 358 and the seals 362 in the bore and limits the travel of the piston toward the post 100'.

An auxiliary cylinder 370 is mounted on the housing extension 322' opening coaxially into the bore 356 and having an internal diameter somewhat larger than that of the bore 356. Within the auxiliary cylinder 370, a piston 372 having a circumferential resilient seal ring 374 is slidably disposed and is biased towards the closed end 376 of the auxiliary cylinder by spring 378 seated against the snap ring 380 in the bore 356. A spring 382 extending between piston 358 and piston 372 urges the pistons in opposed directions. A vent 384 in the housing vents the portion of the bore 356 between the two cylinders to the atmosphere.

The chamber 386 formed by the auxiliary cylinder 370 and the piston 372 is connected by conduit 388 to the speed-regulated fuel pressure. The spring 378 is chosen so as to permit a compression thereof during operation of the engine to permit the piston 372 to move into the position illustrated in FIG. 22 against the end of the housing 322'. When the engine is stopped, the piston 372 will, under the influence of the spring 378, move to the right against the end 376 of the auxiliary cylinder. In this respect, the piston 372 functions in the same manner as and for the same purpose as the piston 119 of the torque control device 50 in permitting an excess fuel position of the fuel delivery system for starting of the engine.

The chamber 386 is connected by housing port 390 and conduit 392 to the engine air intake manifold. The manifold pressure acting against the piston 358 thus opposes the force of the spring 382 in positioning the piston in the bore 356. As shown by curve 394 of FIG. 24, the manifold pressure is a function of engine speed during most phases of engine operation including acceleration and deceleration. However, during rapid acceleration at low speeds, as shown by curve 396, the manifold pressure remains briefly at a low level due to the delayed response of the exhaust turbine mentioned above.

In the operation of the embodiment of FIGS. 21—23, the torque control device 50 operates in the manner described above to override the hydraulic governor 48 to limit the full load fuel delivery of the pump as indicated by the curve 350 on the graph of FIG. 24. During rapid acceleration of the engine from speed Z to speed V, the override device 354 in effect provides an override of the torque control 50 in cutting the fuel back further to the curve 352. When the engine is operating, the fuel pressure in chamber 386 is at all times sufficient to move the piston 372 into the position illustrated in FIG. 22. The piston 358 is positioned with respect to the piston 372 by the manifold pressure in chamber 368 acting against the spring 382.

Upon rapid acceleration of the engine across the speed range P between speeds Z and V, due to the low manifold pressure 396, the spring 382 moves the piston 358 and plunger 360 to the left to cut back the fuel in accordance with curve 352. At speed Z, this movement is shown by increment A in FIG. 22 at which the piston engages the stop 366. The fuel delivery at this position is cut back to 398 as shown in FIG. 24. As speed V is reached, the speed is sufficient to bring the

manifold pressure up to the normal level at which the maximum fuel delivery is determined by the torque control 50 as indicated by curve 350.

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 as defined in and limited solely by the appended claims.

I claim:

1. In a fuel injection pump including a hydraulic head having a bore therein, a pumping and distributing plunger slidably and rotatably disposed within said bore, pump drive means for driving said plunger in rotation and reciprocation, a fuel control sleeve on said plunger axially adjustable to control the end of injection by opening a spill port in the plunger, and a fuel supply pump, the improvement comprising hydraulic governing means and torque control means for said pump, said governing means including a governor piston and cylinder, means for delivering fuel into said cylinder at a pressure proportional to the pump speed, means linking said piston with said fuel control sleeve, a spring acting on said governor piston opposed to the fuel pressure, a throttle linkage connected with said spring for selectively varying the force exerted on said governor piston by said spring, said torque control means comprising a torque control piston slidably disposed in a cylinder, means for delivering fuel under pressure into said cylinder at a pressure proportional to the pump speed, a spring acting on said torque control piston to oppose said fuel pressure, means operatively connecting said torque control piston with said fuel control sleeve during full load fuel delivery of said pump, and means permitting an overriding of said governor means by said torque control means, said means for delivering fuel at a pressure proportional to pump speed to said governor and torque control cylinders comprising an auxiliary pump fed by said fuel supply pump and coupled to said pump drive means, conduit means connecting said auxiliary pump with said governor and torque control cylinders, and a centrifugal fuel pressure regulator driven by said pump drive means connected with said fuel conduit means for regulating the pressure of fuel therein in accordance with the speed of said pump.

2. A fuel injection pump as claimed in claim 1 wherein said means linking said governor piston with said fuel control sleeve comprises a control shaft rotatably mounted in said hydraulic head perpendicularly to said plunger, an eccentric pin on the inner end of said control shaft operatively connected to said control sleeve, and a lever on the outer end of said control shaft extending radially therefrom and operatively connected with said governor piston for movement thereby.

3. A fuel injection pump as claimed in claim 2 wherein said means operatively connecting said torque control piston with said fuel control sleeve comprises a fuel control arm extending from said control shaft, and an extending portion of said torque control piston aligned with and adapted to engage said fuel control arm.

4. A fuel injection pump as claimed in claim 3 wherein said means permitting an overriding of said governor means by said torque control means comprises a spring connection of said lever to said control shaft, said control arm engageable by said torque control device being secured to said control shaft.

5. A fuel injection pump as claimed in claim 4 including excess fuel means actuated by said speed-regulated fuel pressure for reducing the spring force applied to said torque control piston during engine cranking to permit an excess fuel setting of said fuel control arm by said throttle linkage.

6. A torque control device for controlling the maximum fuel output of a fuel injection pump comprising a torque control piston slidably disposed in a cylinder, means for delivering a fluid under pressure into said cylinder on one side of said piston at a pressure varying proportionately with pump speed, said means including an auxiliary fuel pump driven by said fuel injection pump means and fed by the fuel supply pump, conduit means connecting said auxiliary pump with said cylinder, and a centrifugal fuel pressure regulator driven by the pump

drive means connected with said fuel conduit means, spring means acting on said piston to oppose said fluid pressure, means operatively connecting said piston with the pump fuel control during full load fuel delivery of the pump over at least a portion of the operating speed range of the pump, the piston movement under the influence of said spring means serving to decrease pump fuel delivery, excess fuel means comprising an excess fuel piston and cylinder axially aligned with said torque control piston and cylinder, means for delivering the speed regulated fluid pressure to said excess fuel cylinder to urge said piston toward said torque control piston, said spring means extending between said torque control and said excess fuel piston, and supplemental spring means urging said excess fuel piston against said fluid pressure.

7. A torque control device for controlling the maximum fuel output of a fuel injection pump comprising a torque control piston slidably disposed in a cylinder, means for delivering a fluid under pressure into said cylinder on one side of said piston at a pressure varying proportionately with pump speed, spring means acting on said piston to oppose said fluid pressure and means operatively connecting said piston with the pump fuel control during full load fuel delivery of said pump over a portion of the operating speed range of said pump, piston movement under the influence of said spring means serving to decrease pump fuel delivery, a second piston and cylinder, means for delivering fluid at a pressure proportional to the pump speed into said second cylinder, spring means acting against said second piston in opposition to said fluid pressure, and means operatively connecting said second piston with the pump fuel control during pump full load fuel delivery over a second portion of the pump operating speed range, movement of said second piston under the influence of said fuel pressure serving to increase fuel delivery.

8. A torque control device for controlling the maximum fuel output of a fuel injection pump comprising a torque control piston slidably disposed in a cylinder, means for delivering a fluid under pressure into said cylinder on one side of said piston at a pressure varying proportionately with pump speed, spring means acting on said piston to oppose said fluid pressure, means operatively connecting said piston with the pump fuel control during full load fuel delivery of said pump over at least a portion of the operating speed range of said pump, the piston movement under the influence of said spring means serving to decrease pump fuel delivery, an override device for providing fuel cutback upon rapid acceleration of the pump engine, said override device comprising a piston and cylinder adjacent and aligned with the pump fuel control, said latter piston including a plunger extending therefrom, spring means urging said piston and plunger toward engagement with the pump fuel control, and conduit means connected with the engine manifold for delivering engine manifold pressure into said latter cylinder to urge said piston in opposition to said latter spring means.

9. In a fuel injection pump including a hydraulic head having a bore therein, a pumping and distributing plunger slidably and rotatably disposed within said bore, pump drive means for driving said plunger in rotation and reciprocation, a fuel control sleeve on said plunger axially adjustable to control the end of injection by opening a spill port in the plunger, and a fuel supply pump, the improvement comprising hydraulic governing means for said pump, said governing means including a piston and cylinder, means for delivering a fluid into said cylinder at a pressure proportional to the pump speed, said latter means including an auxiliary pump fed by said fuel supply pump and coupled to said pump drive means, conduit means connecting said auxiliary pump with said cylinder, and a centrifugal fluid pressure regulator driven by said pump drive means connected with said conduit means, means linking said piston with said fuel control sleeve to control the movement of said sleeve, a spring means acting on said piston opposed to the pressure of said fluid, and throttle linkage means associated with said spring means for selectively varying the force exerted on said piston by said spring means.

10. In a fuel injection pump including a hydraulic head having a bore therein, a pumping and distributing plunger slidably and rotatably disposed within said bore, pump drive means for driving said plunger in rotation and reciprocation, and a fuel control sleeve on said plunger axially adjustable to control the end of injection by opening a spill port in the plunger, the improvement comprising hydraulic governing means for said pump, said governing means including a piston and cylinder, means for delivering a fluid into said cylinder at a pressure proportional to the pump speed, means linking said piston with said fuel control sleeve to control the movement of said sleeve, spring means acting on said piston opposed to the pressure of said fluid, and throttle linkage means associated with said spring means for selectively varying the force exerted on said piston by said spring means, said means linking said piston with said fuel control sleeve comprising a control shaft rotatably mounted in said hydraulic head perpendicularly to said plunger, an eccentric pin on the inner end of said control shaft operatively connected to said control sleeve, and a lever on the outer end of said control shaft extending radially therefrom and operatively connected with said piston for movement thereby.

11. In a fuel injection pump including a hydraulic head having a bore therein, a pumping and distributing plunger rotatably and slidably disposed within said bore, drive means for providing a reciprocating and rotating motion of the plunger, a closed end of the bore defining a fuel distribution chamber in conjunction with one end of the plunger, means for introducing fuel into the fuel distribution chamber, means in the head and the plunger for distributing fuel from the fuel distribution chamber sequentially to a plurality of outlet ports in the head, passage means in said plunger extending from said

fuel distribution chamber to a spill port opening into a fuel sump in the hydraulic head, and a fuel control sleeve slidably disposed on said plunger in said sump for controlling the injection cutoff by varying the plunger position at which said spill port opens, the improvement comprising a hydraulic governor for controlling the fuel delivery of said pump in conjunction with the pump throttle, said governor including a governor piston slidable within a cylinder, means for delivering fuel into said cylinder at a pressure proportional to the pump speed, said latter means comprising a fuel pump coupled to the plunger drive means, conduit means connecting said fuel pump with said cylinder, and a centrifugal fuel pressure regulator driven by said pump drive means connected with said fuel conduit, means mechanically linking said governor piston to said fuel control sleeve to control the movement of said sleeve, a compression spring connected to said piston urging said piston against the fluid in said cylinder, said spring being operatively connected with the pump throttle operating lever whereby the force exerted on said piston by said spring may be selectively varied in accordance with the desired fuel output of the pump.

12. A fuel injection pump as claimed in claim 11 wherein said means mechanically linking said governor piston with said fuel control sleeve comprises a control shaft rotatably mounted in said hydraulic head perpendicularly to said plunger, an eccentric pin on the inner end of said control shaft operatively connected to said control sleeve, and a lever on the outer end of said control shaft extending radially therefrom and operatively connected with said governor piston for movement thereby.

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