

March 6, 1956

M. E. CHANDLER

2,737,168

FUEL INJECTION APPARATUS

Filed Oct. 22, 1949

2 Sheets-Sheet 1

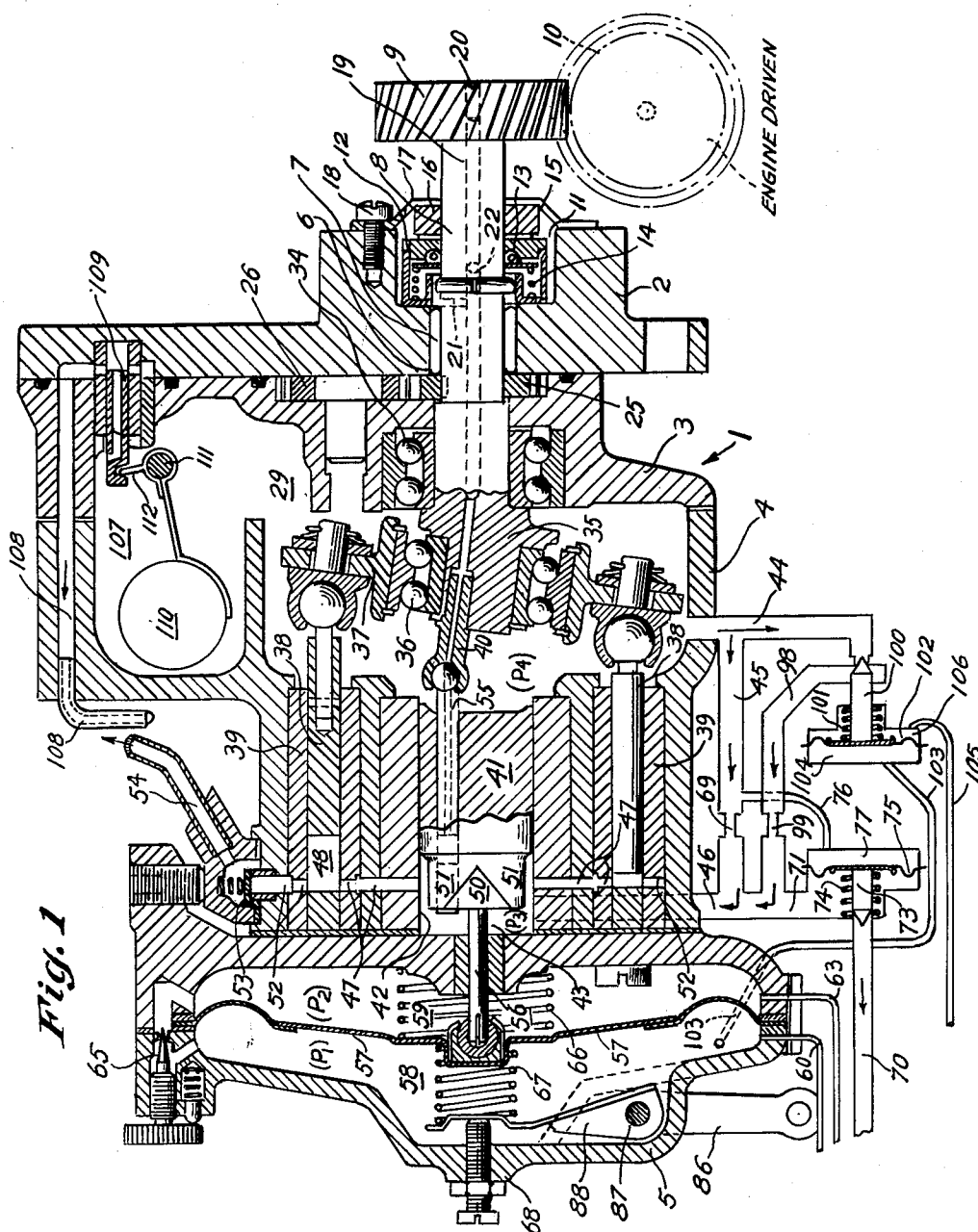


Fig. 1

INVENTOR
M. E. CHANDLER
BY *Am Prentiss*
ATTORNEY

March 6, 1956

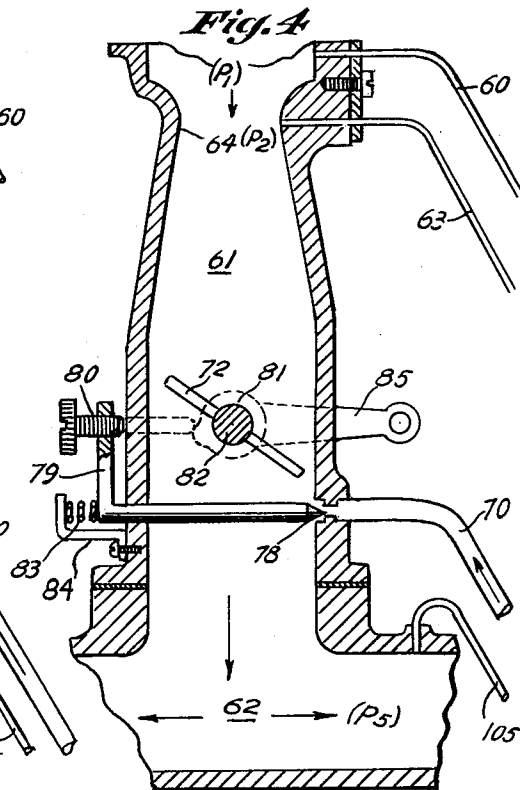
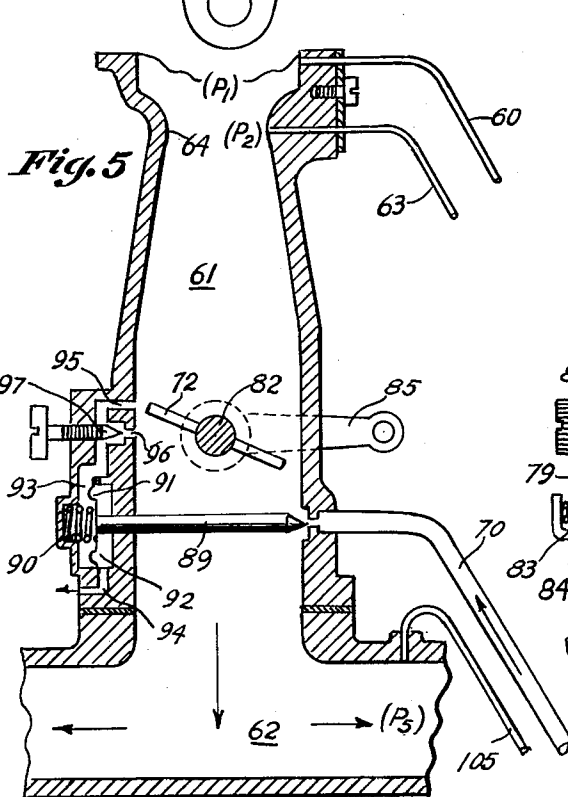
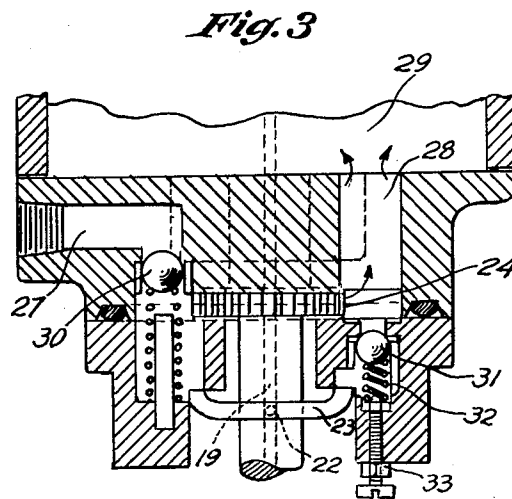
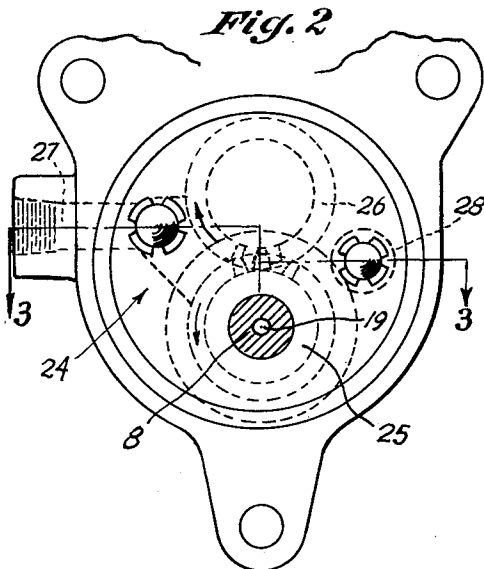
M. E. CHANDLER

2,737,168

FUEL INJECTION APPARATUS

Filed Oct. 22, 1949

2 Sheets-Sheet 2



INVENTOR
M. E. CHANDLER
BY *Am. Prentiss*
ATTORNEY

1

2,737,168

FUEL INJECTION APPARATUS

Milton E. Chandler, New Britain, Conn., assignor, by mesne assignments, to Pratt & Whitney Company, Incorporated, West Hartford, Conn., a corporation of Delaware

Application October 22, 1949, Serial No. 122,969

2 Claims. (Cl. 123—140)

This invention pertains to apparatus for controlling the supply of fuel to an internal combustion engine and more particularly has reference to fuel supply systems wherein metered charges of fuel are directly injected under pressure into the individual cylinders of an internal combustion engine of the conventional four-cycle type.

Fuel injection systems heretofore proposed for Otto cycle engines may, generally speaking, be classified as comprising either:

(1) A multiple barrel fuel pump, wherein each pump barrel supplies an individual engine cylinder, and the pump output per stroke is varied by a common, combined fuel control and distributing member connected to a separate fuel metering control means, as exemplified in the U. S. Patent to Alden, No. 2,052,549, September 1, 1936;

(2) A multiple barrel fuel pump wherein each pump barrel supplies an individual engine cylinder, and the pump output per stroke is varied by a scrolled plunger in each pump barrel that is angularly adjusted by a mechanism actuated and controlled by a common apparatus responsive to air and fuel flow pressure differentials, as exemplified in the U. S. Patent to Chandler, No. 2,410,774, November 5, 1946;

(3) A single rotary fuel pump connected to a common fuel controlling and distributing mechanism by which the fuel supply to the engine is broken up into small charges for the individual engine cylinders and conveyed thereto by suitable conduits, the distributor mechanism being rotated by the engine and axially adjusted to vary the individual fuel charges by mechanism responsive to air and fuel flow pressure differentials, as exemplified in the U. S. Patent to Lee, No. 2,445,308, November 30, 1948.

Each of these fuel injection systems is complex and expensive to manufacture, and each is attended with certain other disadvantages which it is an object of this invention to overcome by providing a relatively simple and inexpensive, single-package apparatus that combines most of the advantages without the disadvantages of the prior art systems, together with certain novel features not heretofore provided in any prior art apparatus.

Thus, the fuel injection system mentioned in (1) above requires three separate interconnected pieces of apparatus, viz.: (a) a fuel boost pump; (b) a multi-cylinder injection pump, as shown in the patent to Alden, cited; and (c) a metering control apparatus, with all the attendant complications and expense of such an arrangement.

The fuel injection system mentioned in (2) above, while combining a fuel boost pump with a metering control device, requires a separate multi-cylinder injection pump with suitable mechanism for simultaneously adjusting the angular position of each pump plunger, and in addition requires that each pump plunger be scrolled, as shown in the patent to Chandler, cited, which is an expensive precision operation.

On the other hand, the fuel injection system mentioned in (3) above, while combining a supply pump with a fuel

2

metering control and distributing device in one piece of equipment, is capable of delivering fuel only under relatively low injection pressures, such as can be developed by a rotary vane pump (as shown in the patent to Lee, cited), which is insufficient for many engine requirements. Also this type of fuel injection system is not capable of the degree of precision in measuring the individual fuel charges to the engine that is attainable with multiple barrel fuel injection pumps, wherein each pump barrel measures and delivers fuel to an individual engine cylinder.

It is accordingly an object of this invention to provide a fuel injection system wherein a relatively simple and compact, unitary apparatus combines all the required fuel supply equipment (boost pump, multi-cylinder injection pump and metering control device) in a single package.

Another object of this invention is to provide a fuel injection apparatus which combines the precision and high pressure fuel delivery capabilities of a multiple barrel pump with plain pump plungers, with a self-contained fuel boost pump and a unitary fuel control and metering device.

A further object of this invention is to provide a fuel injection system having novel means for providing supplementary fuel for idle and full power engine operation.

Still another object is to provide a novel fuel injection system comprising a high-pressure fuel injection pump having a plurality of barrel and plunger units with individual discharge lines, together with means for operating the pump units in timed sequence with the engine, and means including a novel rotary distributing valve member for controlling the delivery of fuel from the pump barrels in timed and metered quantities to their respective discharge lines.

Still other objects of the invention reside in the provision, in a fuel injection system of the foregoing character, the combination of a multiple barrel fuel injection pump, with a novel rotary distributing valve member for delivering a metered quantity of fuel from each barrel over a predetermined intermediate period in the pressure stroke of each plunger; the valve member being axially adjustable to vary the duration of the period of fuel delivery.

Another object is the provision in a fuel injection system of novel means, of relatively simple construction, for effecting control of the supply of fuel in response to both the rate of fuel flow and the rate of air flow to the engine, so that a substantially constant ratio of fuel supply may be maintained throughout the normal range of engine operation.

Another feature of the invention resides in a fuel injection system wherein a rotating fuel distributing valve member is directly subjected on its opposite faces to a fuel differential pressure corresponding to the rate of fuel flow.

With these and other objects in view which may be incident to my improvements, my invention consists in the combination and arrangement of elements hereinafter described and illustrated in the accompanying drawings, in which:

Figure 1 shows a central, longitudinal section (partly diagrammatic) of my improved fuel injection metering apparatus;

Fig. 2 is a partial right end view of the apparatus shown in Fig. 1;

Fig. 3 is a partial section along the line 3—3 of Fig. 2;

Fig. 4 is a central, vertical section (partly diagrammatic) of the air supply conduit to the engine, showing one embodiment of my novel idle fuel control means; and

Fig. 5 is a view, similar to Fig. 4, showing another form of novel idle control means.

Referring to Fig. 1, the reference numeral 1 denotes the assembled housing of my fuel injection apparatus which consists of four principal parts, viz.: a right end closure plate 2, a right end body portion 3, a main body portion 4 and a left end air chamber portion 5, all held in fluid-tight connection to each other by fastening bolts (not shown) and interposed gaskets, as indicated in Fig. 1. The metering jets, idling and economizer control valves, which are built into main body portion 4, are indicated in diagrammatic outline below body portion 4 in Fig. 1, as hereinafter more specifically described.

Plate 2 has a transverse bore 6 for the reception of a needle roller bearing 7, in which is mounted a pump drive shaft 8, having on its outer end a worm gear 9 meshing with a worm wheel 10 driven by the engine at a speed of one-half the engine crank shaft speed. The right end of bore 6 is enlarged into a cylindrical recess 11 for the reception of a fluid-tight bushing 12 which surrounds shaft 8 and has a ring gasket 13 held in fluid-tight contact with shaft 8 by a spring 14 and washer 15. Bushing 12 is held in operating position by backing washer 16 and clamp 17 which latter is secured to plate 2 by a plurality of screws 18. Shaft 8 has a central passageway 19 which extends throughout the length of the shaft and is sealed at its outer end by a plug 20. A transverse passageway 21 connects passageway 19 with the interior of bushing 12, and a second transverse passageway 22, at right angles with passageway 21, connects passageway 19 with by-pass conduit 23 of a fuel boost pump 24 (see Fig. 3).

Pump 24 comprises a pair of intermeshing gears 25 and 26, located in a recess in the right end of body portion 3, and held in operating position by plate 2 (see Figs. 1 and 2). Fuel enters pump 24 from a supply tank (not shown) through an inlet 27 and is discharged through an outlet 28 into chamber 29 in body portions 3 and 4 of housing 1. A ball check valve 30 prevents back flow of fuel from pump 24 into inlet 27, and a second ball check valve 31 regulates the discharge pressure of the pump 24 by the compression of a spring 32 which is adjustably loaded by a locked screw 33 (see Fig. 3). For any given setting of screw 33 the pump discharge pressure is maintained substantially constant by shunting excess fuel from pump outlet 28 back to pump inlet 27 through by-pass 23.

The left half of shaft 8 is supported by a ball bearing 34 (Fig. 1) and terminates in angularly disposed off-set portion that is connected through a ball bearing 36 with an annular wobble plate 37, to which are attached, by ball-in-socket connections, a plurality of injection pump plungers 38. The number of plungers 38 is equal to the number of cylinders of the engine to be supplied by the fuel injection pump, and each plunger 38 is slidably mounted, with a fluid-tight fit, in a pump cylinder 39 which is secured in aligned position in the left end of body portion 4. Shaft 8 also carries at its extreme left end an angularly disposed extension 40 which is attached, by a ball-in-socket connection, to a cylindrical distributing valve member 41 that is slidably mounted, with a fluid-tight fit, in a cylinder 42, and is also rotated by shaft 8 through eccentric connection 43.

Chamber 43 in the left end of cylinder 42 is supplied with fuel from chamber 29 through connecting conduits 44, 45 and 46. From chamber 43 fuel flows through a connecting passageway 47 to a chamber 48 in the left end of each pump cylinder 39, except when the inner end of passageway 47 is blocked during the middle third of each pumping stroke by a scroll land 50 on the left end of valve member 41. As each pump plunger 38 moves to the right, a scroll groove 51 on valve member 41 uncovers the inner end of passageway 47 of that cylinder 39 by virtue of the shape of groove 51 and the simultaneous rotation of valve member 41, so that fuel flows from chamber 43 into chamber 48 of that cylinder 39. When each plunger 38 moves in the reverse di-

rection (to the left) on its pumping stroke, inlet passageway 47 of that cylinder 39 is blocked during the middle third of the stroke by land 50 of valve member 41, and the fuel thus trapped in chamber 48 is expelled through an outlet passageway 52, past a check valve 53, to a conduit 54 leading to the engine cylinder connected to that pump unit. While the stroke of each plunger 38 is the same and constant, the amount of fuel injected into each engine cylinder per pump stroke depends upon the horizontal axial position of valve member 41, since the scroll contours of land 50 and groove 51 change the amount of cut-off of passageways 47 with each change in axial position of valve member 41. Thus, as valve member 41 moves to the right it increases the partial or total cut-off period of passageways 47 for each stroke of plungers 38 and thus increases the amount of fuel injected per stroke by each pump unit, and vice versa. Valve member 41 is made movable in cylinder 42 by the telescoping of the pin 55, which connects valve member 41 to extension 40 of shaft 8, within said valve member, the pin 55 being made shorter than the bore in which it fits.

Valve member 41 is reciprocated in cylinder 42 by a rod 56 connecting member 41 with a diaphragm 57 which latter is mounted in body portion 5 so as to divide said portion into two expansible chambers 58 and 59. Chamber 58 is connected by a conduit 60 to the entrance of an air supply passage 61 leading to the intake manifold 62 of the engine (Fig. 4), and chamber 59 is connected by a conduit 63 to the Venturi throat 64 of air passage 61. By these means diaphragm 57 is subjected to the differential between the air inlet pressure (P_1) and the Venturi throat pressure (P_2), which differential pushes valve member 41 to the right, in opposition to the differential between the fuel pressure (P_4) in chamber 29 and the fuel pressure (P_3) in chamber 43.

From the foregoing it is clear that so long as ($P_1 - P_2$) exceeds ($P_4 - P_3$), valve member 41 will move to the right, and vice versa, so that the axial position of valve member 41 at any one time depends upon a balance between the air flow differential ($P_1 - P_2$) and the fuel flow differential ($P_4 - P_3$). Since the axial position of valve member 41 determines the amount of fuel delivered to the engine per stroke of each pump plunger, and each pump unit delivers a fuel charge on each intake stroke of the engine, it follows that the rate of fuel supply to the engine at any engine speed is proportioned to the rate of air supply and a substantially constant mixture ratio is thus maintained under normal operating conditions. The mixture ratio can be varied by opening a manually adjustable valve 65 which permits air to bleed from chamber 58 into chamber 59. This reduces the air pressure differential ($P_1 - P_2$) acting on diaphragm 57 for any given rate of air flow through passage 61. The more valve 65 is opened, the less fuel is supplied to the engine for any given rate of air flow and hence the mixture is weakened. As applied to motor vehicle engines, which generally operate at or near sea level, valve 65 would be normally closed, and would only be opened when the motor vehicle is required to operate at altitudes materially above sea level. As applied to aircraft engines, valve 65 would normally be opened progressively with increase in flight altitude, and this is generally accomplished by actuating valve 65 by means of an air density responsive bellows, or like device.

Idling system

In order to insure that diaphragm 57 will be in a position to cause proper fuel flow, when the air flow is within the idling range of the engine, a light spring 66 is interposed between diaphragm 57 and the opposite wall of chamber 59. Spring 66 is opposed by another light spring 67 which is used primarily to momentarily increase the mixture ratio for starting the operation of the engine, as further described hereinbelow. The normal compression

of spring 67 is adjusted by lock screw 68 so that the difference in thrusts of springs 66 and 67 will just balance the air flow pressure differential (P_1-P_2) acting on diaphragm 57 when the air flow to the engine is at a maximum rate for the idling range. This insures that diaphragm 57 will be in the correct position to cause proper rate of fuel flow to the engine to maintain the desired mixture ratio in the idling range of engine operation.

During the normal operating range of the engine, between the idling range and the range of maximum power output, the fuel is metered by a metering restriction 69 in fuel conduit 45, and the pressure drop across this restriction creates the fuel pressure differential (P_4-P_3). For operation in the idling range an auxiliary fuel supply system is provided as follows. A fuel conduit 70 is connected to conduit 46, through an intermediate conduit 71, and leads to the main air supply passage 61 at a point below the throttle valve 72 (Fig. 4) to supply fuel to the engine when idle fuel flow cut-off valve 73 in conduit 70 is open. Valve 73 is actuated by a spring 74 which is opposed by the fuel pressure differential (P_4-P_3) acting on a diaphragm 75, so that whenever the fuel flow through metering restriction 69 falls to a point where the force of (P_4-P_3) is less than that of the spring 74, valve 73 opens and remains open as long as the fuel flow differential pressure (P_4-P_3) is less than the force of spring 74. Conversely, when the fuel flow through metering restriction increases to a point where (P_4-P_3) exceeds the force of spring 74, valve 73 closes and cuts off fuel supply through the idle fuel conduit 70 and the engine commences to operate in its normal operating range, as described hereinabove. The fuel pressure (P_4) is transmitted from conduit 45 through a pipe 76 to a sealed chamber 77 where it acts upon the diaphragm 77, urging it to the left. At the same time, fuel pressure (P_3) is applied through conduit 71 to the other side of diaphragm 75, urging it to the right. Thus the fuel pressure differential (P_4-P_3) acts on diaphragm 75 in a direction opposing spring 74, as indicated above.

The emission of fuel from conduit 70 into air passage 61 is regulated by a valve 78 (Fig. 4) mounted in the side wall of air passage 61, with an arm 79 carrying an adjusting screw 80 which bears against a cam 81 fixed on throttle shaft 82. A spring 83 interposed between a bracket 84 and the outer end of valve stem 78 insures contact between the end of screw 80 and cam 81 at all times. The shape of cam 81 is such that valve 78 is opened only when throttle 72 approaches its closed position, corresponding to the idling range of the engine. In all other throttle positions, corresponding to all other conditions of operation, valve 78 is seated and remains closed. Cam 81 is also so shaped as to cause valve 78 to close when throttle 72 is completely closed, thus preventing drainage of fuel from conduit 70 when the engine is not in operation.

Throttle shaft 82 carries a fixedly attached arm 85 by which the operator controls the position of throttle 72 and operation of the engine through a linkage connection (not shown). Another arm 86 (Fig. 1) is fixedly attached to a shaft 87 journaled in body portion 5. Shaft 86 also carries (inside chamber 58) a fixedly attached lever 88 which engages the left end of spring 67, so that when arm 86 is manually moved in a clockwise direction by the operator of the engine, through a linkage connection (not shown), spring 67 is compressed and diaphragm 57 is moved slightly to the right, thus increasing fuel flow and the mixture ratio to facilitate starting. As soon as the engine starts, arm 86 is returned by the operator to its normal position, as shown in Fig. 1. Arm 86, lever 88 and spring 67 are thus the equivalent of the conventional choke valve in the air intake passage of a carburetor which is generally employed to facilitate starting of the engine. And, like the conventional choke valve, arm 86, lever 88 and spring 67 can be operated by any of the

conventional automatic temperature-responsive devices, in lieu of manual operation, if desired.

In lieu of the manually-operated idle fuel control valve shown in Fig. 4, there may be substituted an idle fuel cut-off valve operated by differential air pressure, as illustrated in Fig. 5. This arrangement comprises a valve 89 mounted in the side wall of air passage 61 and adapted to engage the outlet port of conduit 70. Valve 89 is actuated by a spring 90, opposed by a diaphragm 91 which is mounted in a space in the wall of air passage 61 so as to divide said space into two chambers 92 and 93. Chamber 92 communicates, through a vent 94, with the outside atmosphere, so that the pressure in chamber 92 is always atmospheric. On the other hand, chamber 93 communicates through a passageway 95 with air passage 61, at a point just above the closed position of throttle 72. Chamber 93 also communicates through a passageway 96 with air passage 61 at a point just below the closed position of throttle 72. By virtue of this arrangement the pressure in chamber 93 is a maximum when the throttle is opened beyond idling range, and is a minimum when the throttle is in its minimum idling position. As throttle 72 is closed from its maximum to its minimum idling position, the pressure differential acting on diaphragm 91 decreases, but overcomes the thrust of spring 90 and opens valve 89, thus permitting fuel to flow from conduit 70 into air passage 61. When the throttle 72 is opened beyond maximum idling position the pressure differential is reduced to a value less than the thrust of spring 90, whereupon spring 90 closes valve 89. The valve of the pressure differential acting on diaphragm 91 is adjusted by means of a manually adjustable valve 97 which controls the flow of air through passageway 96, so that the idle fuel flow from conduit 70 may be adjusted to give the desired mixture ratio for idling operation of the engine.

Economizer system

In order to provide for an increased fuel flow to properly enrich the fuel/air mixture during the maximum power range of engine operation, there is provided a supplementary fuel supply through conduits 44, 98 and 71 (Fig. 1). A metering restriction 99 in conduit 98 regulates this supplementary fuel flow and an economizer valve 100 controls the period during which this flow takes place. Valve 100 is operated by a spring 101, opposed by a diaphragm 102 which is actuated by pressure differential (P_1-P_5); the pressure (P_1) being transmitted from chamber 58 through a pipe 103 to a chamber 104 on the left of diaphragm 102, while the pressure (P_5) is transmitted from manifold 62 through a pipe 105 (Figs. 4 and 5) to chamber 106 in the right of diaphragm 102 (Fig. 1). During the normal range of engine operation, the pressure differential (P_1-P_5) acting on diaphragm 102 is greater than the thrust of spring 101, so that valve 100 remains closed and no supplementary fuel is supplied through conduits 44, 98 and 71. However, when the engine commences to operate in its maximum power range, pressure differential (P_1-P_5) drops to a value less than the thrust of spring 101 and valve 100 opens and permits supplementary fuel flow through conduits 44, 98 and 71 to take place. Such flow is metered by metering restriction 99, in coordination with the main fuel flow through metering restriction 69, so as to maintain a desired higher mixture ratio during maximum power range.

Vapor elimination

In order to eliminate any air and/or fuel vapor from the fuel supplied by boost pump 24 before said fuel reaches metering restrictions 69 and 99, a vapor trap and vent are provided in the top chamber 29; it being understood that the apparatus shown in Fig. 1 normally operates in a horizontal position, as indicated in Fig. 1. Since any air and/or fuel vapor in chamber 29 is lighter than the liquid fuel therein, such air and/or vapor will gradu-

ally collect in the space 107 at the top of chamber 29 and displace the liquid fuel in space 107, thus lowering the liquid level in chamber 29. To eliminate this air and/or vapor from space 100, there is provided a vent passage 108 to the outside atmosphere. Flow through vent 108 is controlled by a sleeve valve 109 which is actuated by a float 110, pivotally mounted at 111 and connected to valve 109 by an arm 112. When the liquid level in chamber 29 is depressed by the accumulation of air and/or vapor in space 107, float 110 falls and opens valve 109, thus permitting said air and/or vapor to escape through vent 108. Immediately upon the release of said air and/or vapor from space 107, the fuel in chamber 29 rises to replace it, thereby causing valve 109 to close vent 108.

Distributor valve balancing

In order to counterbalance the excess thrust of the pressure differential (P_1-P_2) on diaphragm 57 and hence on valve member 41, owing to the cross-sectional area of rod 56, the fuel pressure at the inlet side of the fuel boost pump 24 is transmitted by passageways 23, 22, 21 and 19 to the left end of the bore in valve member 41 in which pin 55 slides. If fuel is fed by gravity to fuel boost pump under a pressure equal to pressure (P_1), the cross-sectional area of pin 55 would be made equal to the cross-sectional area of rod 56 in order to balance these thrusts on valve member 41. On the other hand, if boost pump 24 draws fuel from its supply tank (not shown) under a pressure less than (P_1), the cross-sectional area of pin 55 would be made correspondingly greater than that of rod 56, and vice versa. In this manner, the additional thrust of the pressure differential (P_1-P_2) due to the cross-sectional area of rod 56, is counterbalanced and valve member 41 is actuated by the air pressure differential (P_1-P_2) opposed by the fuel pressure differential (P_4-P_3) plus the net thrust of springs 66 and 67 during the normal and maximum power ranges of engine operation.

Operation

During normal operation of the engine, fuel is fed into chamber 29 by fuel boost pump 24 at a substantially constant pressure (P_4). From chamber 29 the fuel flows through conduits 44, 45 and 46 to chamber 43 and in passing through metering restriction 69 in conduit 45, the fuel pressure undergoes a drop equal to (P_4-P_3), which fuel pressure differential is proportional to the rate of fuel flow to chamber 43 and acts on opposite ends of valve member 41, always urging it to the left. At the same time, valve member 41 is constantly urged to the right by the air pressure differential (P_1-P_2) acting on diaphragm 57. Hence, the axial position of valve member 41 is at all times determined by the balancing of the air pressure differential (P_1-P_2) against the fuel pressure differential (P_4-P_3) plus the net thrust of springs 67 and 68. Thus, if the fuel flow through metering restriction 69 should tend to increase at a greater rate than the air flow through air passage 61, valve member 41 would move to the left (decreasing fuel delivery to the engine) and vice versa.

Since the width of land 50 on valve member 41 is a maximum at its left end and a minimum at its right end, any movement of valve member 41 to the left decreases the angular period during which land 50 partially or completely cuts off communication between chamber 43 and passageway 47. And since fuel is ejected from each pump chamber 48 only during such period as such cut-off occurs, it follows that the shorter the period of cut-off, the less fuel is ejected from each pump chamber 48 per pump stroke. Conversely, when valve member 41 moves to the right, it correspondingly increases the amount of fuel ejected from each pump chamber 48 per pump stroke.

Furthermore, since the valve member 41 and pump plungers 38 are connected to the engine so that the former makes one complete revolution, and each of the latter one complete stroke, for each two revolutions of the engine,

the amount of fuel ejected from each pump chamber 48 per pump stroke determines the total rate of fuel delivery to the engine. It is also clear from the foregoing that during normal engine operation the rate of fuel flow to the engine will be maintained by the action of valve member 41 at a substantially constant ratio to the air flow and hence a substantial mixture ratio will be secured.

When the engine enters its idling range of operation, valve member 41 is in its extreme left position wherein land 50 no longer blocks communication between chamber 43 and passageways 47 in any degree at any time during the rotation of member 41; hence no fuel is ejected from pump chambers 48 to the engine cylinders. On the contrary, check valves 53 remain closed and all the fuel in chambers 48 is returned to chamber 43 on each stroke of plungers 38. Therefore, no fuel is supplied to the engine by the injection pump. At the same time, the closing of the throttle to the idling position shown in Figs. 4 and 5, causes the idle fuel cut-off valve 73 to open and permit fuel to flow from conduit 70 into air passage 61. The rate of flow of fuel through conduit 70 is regulated by the valve 78 or 89 in accordance with the degree of throttle opening in the idling range, and the proper idling mixture is thus maintained. When the engine speed increases, the air pressure differential (P_1-P_2) correspondingly increases, and upon reaching a value greater than the force of spring 66 plus the fuel pressure differential (P_4-P_3), pushes valve member 41 to the right, so that land 50 commences to partially block outlet ports 47 on each revolution of valve member 41 and fuel commences to flow to the engine through conduits 54. At the same time the flow of fuel through conduit 45 increases the pressure drop across restriction 69 so that the force of (P_4-P_3) acting on diaphragm 75 overcomes the force of spring 74 and closes valve 73. This cuts off idle fuel flow through conduit 70.

Similarly, when the engine speed increases to a point corresponding to the lower limit of the maximum power output range, the force of (P_1-P_2) acting on diaphragm 102 has increased to a value sufficient to overcome the force of spring 101 and open valve 100, whereupon supplementary fuel flows to chamber 43 (and thence to the engine) to properly enrich the mixture for maximum power operation. Metering restriction 99 is so related to metering restriction 69 that the fuel flows through conduits 45 and 98 are properly coordinated to give the desired mixture ratio during maximum power operation.

While I have shown and described the preferred embodiment of my invention I desire it to be understood that I do not limit myself to the details of construction disclosed by way of illustration, as these may be changed and modified by those skilled in the art without departing from the spirit of my invention or exceeding the scope of the appended claims.

I claim:

1. A fuel supply system for an internal combustion engine having a conduit for supplying air to said engine for combustion purposes and a throttle for controlling the flow of air through said conduit, comprising: an injector pump for delivering fuel directly to said engine, means including a rotary fuel distributing valve member reciprocable directly by the difference between the quantity of air flowing through said conduit and to the quantity of fuel flowing through said pump, to control the delivery of said injector pump, means operative as an incident to a decrease in said quantities below a predetermined value of each to permit an additional flow of fuel to said conduit at a point between said throttle and said engine, and means responsive to the position of said throttle for controlling said additional flow.

2. A fuel supply system for an internal combustion engine having a conduit for supplying combustion air to said engine and a throttle for controlling the flow of air through said conduit, comprising: an injector pump for delivering fuel directly to said engine, means including a

rotary fuel distributing valve member reciprocally responsive to the rate of air flow through said conduit and to the rate of fuel flow through said pump to control the delivery of said injector pump, and means operative as an incident to a decrease in said rates below a predetermined value of each to stop fuel delivery from said pump and produce a flow of fluid to said conduit at a point between said throttle and said engine.

2,384,282
2,402,332
2,447,265
2,453,329

577,007
594,336

References Cited in the file of this patent

10

UNITED STATES PATENTS

2,156,933 Alden ----- May 2, 1939

Chandler ----- Sept. 4, 1945
Lee, 2nd ----- June 18, 1946
Beardsley, Jr. ----- Aug. 17, 1948
Lee, II ----- Nov. 9, 1948

FOREIGN PATENTS

Great Britain ----- May 1, 1946
Germany ----- Mar. 15, 1934