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(71) Applicant: SIEMENS CANADA LIMITED [CA/CA]; 2185 Derry Road West, Mississauga, Ontario L5N 7A6 (CA).

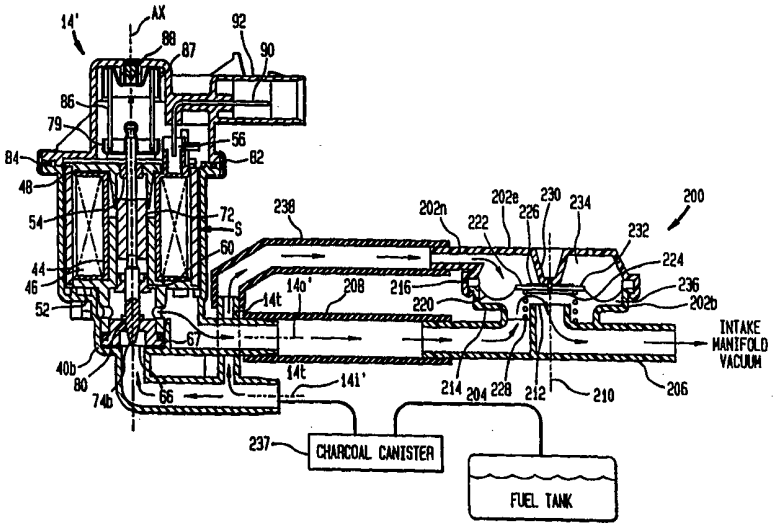
(72) Inventors: COOK, John, E.; 17 Kingsway Drive, Chatham, Ontario N7L 2S8 (CA). BUSATO, Murray, F.; 304–97 Bedford Street, Chatham, Ontario N7L 2V4 (CA).

(74) Agent: MACRAE & CO.; P.O. Box 806, Station "B", Ottawa, Ontario K1P 5T4 (CA).

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(54) Title: REGULATED LINEAR PURGE SOLENOID VALVE



(57) Abstract

A pressure regulator is associated with a solenoid–operated valve that may be operated by a pulse waveform at a fundamental frequency substantially greater than the frequency response of the valve mechanism. The pressure regulator causes a predetermined pressure differential across the valve mechanism that is independent both of pressure at the outlet of the regulator and pressure at the inlet of the valve, thereby providing improved flow control accuracy. The invention is especially advantageous for purging fuel vapor generated in a fuel tank to an intake manifold of an internal combustion engine of an automotive vehicle.

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REGULATED LINEAR PURGE SOLENOID VALVE

Field of the Invention

This invention relates generally to emission control valves for automotive vehicles. In one specific aspect, the invention relates to solenoid-operated fluid valves for purging volatile fuel vapors from fuel tanks and vapor storage canisters to internal combustion engines that power such vehicles.

Background of the Invention

A known on-board evaporative emission control system comprises a vapor collection canister that collects volatile fuel vapors generated in the headspace of the fuel tank by the volatilization of liquid fuel in the tank and a canister purge solenoid (CPS) valve for periodically purging collected vapors to an intake manifold of the engine. The CPS valve comprises a solenoid actuator that is under the control of a microprocessor-based engine management system.

During conditions conducive to purging as determined by the engine management system on the basis of various inputs to it, evaporative emission space that is cooperatively defined by the tank headspace and the canister is purged to the engine intake manifold through the CPS valve, which is fluid-connected between the canister and the engine intake manifold. The CPS valve is opened by a signal from the engine management computer in an amount that allows intake manifold vacuum to draw volatile fuel vapors from the canister for entrainment with the combustible mixture passing into the engine's combustion chamber space at a rate consistent with engine operation to provide both acceptable vehicle driveability and an acceptable level of exhaust emissions.

A known CPS valve comprises a movable valve element that is resiliently biased by a compression spring against a valve seat to close the valve to flow when no electric current is being delivered to the solenoid. As electric current begins to be increasingly applied to the solenoid, increasing electromagnetic force acts in a sense tending to unseat the valve element and thereby open the valve to fluid flow. This electromagnetic force must overcome various forces acting on the mechanical mechanism before the valve element can begin to unseat, including overcoming both whatever static friction (stiction) is present between the valve element and the seat, as well as the opposing spring bias force. Once the valve element has unseated, the valve element/valve seat geometry also plays a role in defining the functional relationship of fluid flow rate through the valve to electric current supplied to the solenoid coil. Furthermore, the extent to which a given valve possesses hysteresis will also be reflected in the functional relationship.

When the valve element comprises a tapered pintle that is selectively positioned axially within a circular orifice which is circumscribed by the valve seat, a well defined flow rate vs. pintle position characteristic can be obtained. However, certain geometric factors present at the valve element/valve seat interface may prevent this characteristic from becoming effective until the valve element has unseated a certain minimum distance from the valve seat. Accordingly, each graph plot of fluid flow rate through the valve vs. electric current supplied to the solenoid coil may be considered to comprise distinct spans: a short initial span that occurs between valve closed position and a certain minimum valve opening; and a more extensive subsequent span that occurs beyond a certain minimum valve opening.

One specific type of CPS valve comprises a linear solenoid and a linear compression spring that is increasingly compressed as the valve increasingly opens. It is sometimes referred to as a linear solenoid purge valve, or LSPV for short. Such a valve can provide certain desirable characteristics for flow control. By itself, a linear solenoid possesses a force vs. electric current characteristic that is basically linear over a certain range of current. When a linear solenoid is incorporated in an electromechanical device, such as a valve, the overall electromechanical mechanism possesses an output vs. electric current characteristic that is a function of not just the solenoid, but also the mechanical mechanism, such as a valve mechanism, to which the solenoid force is applied. As a consequence then, the output vs. electric current characteristic of the overall device is somewhat modified from that of the linear solenoid alone.

While a CPS valve that incorporates both a linear solenoid and a tapered pintle valve element which is selectively positionable axially within a circular orifice that is circumscribed by the valve seat can exhibit a desired fluid flow rate vs. pintle position characteristic, such characteristic may not become effective until after the pintle has opened a certain minimum amount because of geometric factors at the pintle/seat interface, as noted earlier. Accordingly, each graph plot of fluid flow rate through the valve vs. electric current applied to the solenoid coil may be considered to comprise the spans referred to above, namely, a short initial span that occurs between valve closed position and a certain minimum valve opening, and a more extensive subsequent span that occurs beyond a certain minimum valve opening.

Generally speaking, a linear solenoid purge valve may be graphically characterized by a series of graph plots of fluid flow rate vs. electric current, each of which is correlated to a particular pressure differential across the valve. Each graph plot may be characterized by the aforementioned short initial span and the more extensive subsequent span. Within the latter span of each graph plot, one especially desirable attribute is that a substantially constant relationship between incremental

change in an electric control current applied to the solenoid and incremental change in fluid flow rate through the valve may be obtained by appropriate design of the valve element/valve seat interface geometry. Within the former span, incremental change in fluid flow rate through the valve may however bear a substantially different relationship to incremental change in an electric control current applied to the solenoid.

In one such linear solenoid purge valve, a certain minimum electric current is required before the valve begins to open. For a given pressure differential across the valve, a corresponding graph plot of fluid flow rate vs. electric current may be described as comprising a relatively short initial span where a small incremental change in electric current will result in an incremental change in flow that is much different from the incremental change that occurs over an ensuing span where the valve has opened beyond a certain minimum opening and incremental change in flow through the valve bears a substantially constant relationship to incremental change in electric current.

Electric current to the solenoid coil of any solenoid-operated device can be delivered in various ways. One known way is by applying a pulse width modulated D.C. voltage across the solenoid coil. In choosing the pulse frequency of the applied voltage, consideration may be given to the frequency response characteristic of the combined solenoid and mechanical mechanism operated by the solenoid. If a pulse frequency that is well within the frequency response range of the combined solenoid and mechanism is used, the mechanism will faithfully track the pulse width signal. On the other hand, if a pulse frequency that is well beyond the frequency response range of the combined solenoid and mechanical mechanism is used, the mechanism will be positioned according to the time average of the applied voltage pulses. The latter technique may be preferred over the former because the mechanical mechanism will not reciprocate at the higher frequency pulse width modulated waveform, but rather will assume a position corresponding to the time averaged current flow in the solenoid coil. Under the former technique, the mechanism could, by contrast, experience significant reciprocation as it tracks the lower frequency waveform, and that might create unacceptable characteristics. In the case of a CPS valve, such characteristics may include undesirable pulsations in the purge flow and objectionable noise caused by repeated impacting of the valve element with the valve seat and/or a limit stop that limits maximum valve travel. Such a valve may experience unacceptable variation in the start-to-flow duty cycle.

In order to address the pulsation issue, it is known to associate a mechanical pressure regulator with a CPS valve. The pressure regulator mechanically damps the purge flow pulses, but does not address the root cause, which is due to the pulsating solenoid.

A known mechanical pressure regulator comprises a flow path having an entrance through which fluid flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit from which fluid flow that has entered the pressure regulator flow path exits the pressure regulator flow path. The pressure regulator comprises a pressure regulating mechanism that regulates the pressure at the entrance of the pressure regulator flow path to a pressure that is essentially independent of pressure at the exit of the pressure regulator flow path. The pressure regulating mechanism comprises a fluid-impermeable movable wall that divides an interior space of the pressure regulator into two variable volume chamber spaces. A first of these chamber spaces is communicated to atmosphere so that the pressure therein is maintained at atmospheric pressure. A second of these chamber spaces forms a portion of the flow path that extends between the pressure regulator entrance and the pressure regulator exit. The movable wall carries an element that selectively restricts the flow path such that the pressure in the second chamber space is regulated to a predetermined magnitude that is substantially independent of the pressure at the pressure regulator exit. In this way, the vacuum in the first chamber space is regulated relative atmospheric pressure to a predetermined magnitude that is substantially independent of the magnitude of intake manifold vacuum that is communicated to the pressure regulator exit.

Summary of the Invention

One aspect of the present invention relates to an improvement in how the pressure regulating function is performed.

In a fuel vapor collection system having a regulated canister purge valve, the inlet port of the canister purge valve is communicated through the canister to the fuel tank headspace such that the pressure at the purge valve inlet port is essentially that which exists in the tank vapor headspace. It has been observed that certain conditions may cause the vapor pressure in the tank headspace to depart from atmospheric pressure such that the pressure applied to the purge valve inlet port departs from atmospheric. Because both the intake manifold vacuum that is applied to the purge valve outlet port and the pressure in the first chamber space of the pressure regulator are referenced to atmospheric pressure, a departure of tank headspace pressure from atmospheric pressure may degrade the intended control strategy.

One general aspect of the invention relates to a pressure-regulated purge valve that negates effects of departures of tank vapor headspace pressure from atmospheric on intended control strategy. This is accomplished by communicating the first chamber space of the pressure regulator to the tank vapor headspace pressure, either by a

devoted communication path directly to the tank headspace or through any path that is in communication with the tank headspace, such as through the canister.

Another aspect relates to an electric-operated pressure-regulated fluid flow control valve comprising a valve mechanism that is positioned within a valve body by an electric control signal to control fluid flow through the valve body, and a pressure regulator comprising a flow path having an entrance through which fluid flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit from which fluid flow that has entered the pressure regulator flow path exits the pressure regulator flow path, the pressure regulator comprising a pressure regulating mechanism that regulates the pressure at the entrance of the pressure regulator flow path to a pressure that is essentially independent of pressure at the exit of the pressure regulator flow path and that causes flow through the valve to be essentially independent of change in pressure at the valve inlet.

Another general aspect relates to an electric-operated pressure-regulated fuel vapor purge valve for purging fuel vapor from a fuel tank to an intake manifold of an internal combustion engine comprising a valve mechanism that is positioned within a valve body by an electric control signal to control flow through the valve body, and a pressure regulator comprising a flow path having an entrance through which flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit for communicating the pressure regulator flow path to an engine intake manifold, the pressure regulator comprising a pressure regulating mechanism that regulates the pressure at the entrance of the pressure regulator flow path to a pressure that is essentially independent of pressure at the exit of the pressure regulator flow path and that causes flow through the valve to be essentially independent of change in pressure at the valve inlet.

A further aspect relates to the valve mechanism having a frequency response characteristic which renders the valve mechanism incapable of faithfully tracking the fundamental frequency of an electric control signal whose fundamental frequency is greater than a predetermined frequency so that, when applied in control of the valve mechanism, the signal positions the valve mechanism to a position corresponding to a most recent time average of the electric control signal free of any significant pulsing of the valve mechanism.

A further aspect relates to an LSPV, including a pressure regulator, that is believed to provide further improvements in purge flow control accuracy over a substantial range of valve operation and under diverse operating conditions.

A still further aspect relates to the provision of certain constructional features in a pressure regulator that, in association with a CPS valve, are believed to provide improved purge flow control accuracy by significantly attenuating the influence of

conditions that would otherwise produce degradation of the purge flow control for a given valve opening.

The foregoing, along with additional features, and other advantages and benefits of the invention, will be seen in the ensuing description and claims which are accompanied by drawings. The drawings disclose a preferred embodiment of the invention according to the best mode contemplated at this time for carrying out the invention.

Brief Description of the Drawings

Fig. 1 is a schematic diagram of an on-board evaporative emission control system, including an enlarged longitudinal cross-sectional view through a canister purge solenoid valve.

Fig. 2 is a representative graph plot related to Fig. 1.

Fig. 3 is a longitudinal cross-sectional view through another canister purge solenoid valve.

Fig. 4 is a longitudinal cross-sectional view through the canister purge solenoid valve of Fig. 3 and an associated pressure regulator in accordance with the inventive principles.

Fig. 5 is a series of graph plots useful in explaining the inventive principles in relation to Fig. 4.

Fig. 6 is a longitudinal view, partly in cross-section, through another embodiment in accordance with the inventive principles.

Fig. 7 is a longitudinal view of the embodiment of Fig. 6, but having a different portion in cross-section.

Description of the Preferred Embodiment

Fig. 1 shows an evaporative emission control system 10 of a motor vehicle comprising a vapor collection canister (carbon canister) 12 and a canister purge solenoid (CPS) valve 14 connected in series between a fuel tank 16 and an intake manifold 18 of an internal combustion engine 20 in a known fashion. An engine management computer 22 supplies a valve control signal as an input to a pulse width modulation (PWM) circuit 24 to create a pulse width modulated signal which is amplified by a drive circuit 26 and applied to electric terminals 14et of valve 14.

Valve 14 comprises a housing 28 having an inlet port 14i that is fluid-coupled via a conduit 30 with a purge port 12p of canister 12 and an outlet port 14o that is fluid-coupled via a conduit 32 with intake manifold 18. A conduit 34 communicates a canister tank port 12t to headspace of fuel tank 16. An operating mechanism comprising a solenoid actuator 14sa is disposed within housing 28 for opening and

closing an internal passage that extends between ports 14i and 14o. The mechanism includes a bias spring 14bs that acts to urge a valve element 14ve closed against a valve seat 14vs for closing the internal passage to flow. When the solenoid actuator is progressively energized by engine management computer 22, electromagnetic force is applied to an armature 14a in opposition to the bias spring force to unseat valve element 14ve from valve seat 14vs and thus open the internal passage so that flow can occur between ports 26 and 30.

Canister 12 is also seen to comprise a vent port 12v via which the evaporative emission space where the fuel vapors are contained is vented to atmosphere. Such venting may be via an atmospheric vent valve (not shown) that is operated closed at certain times, such as during OBDII testing.

Fig. 2 depicts a representative control characteristic for valve 14 wherein fluid flow rate through the valve is related to the duty cycle of a pulse width modulated voltage that is applied across terminals 14et. A certain minimum duty cycle, about 10% in the example, is required before the valve begins to open. As the duty cycle increases beyond 10%, the flow rate bears a generally straight line relationship to duty cycle. At 100% duty cycle a constant D.C. voltage is applied across terminals 14et. The frequency of the pulse waveform that accomplishes this type of operation is relatively low, a representative frequency being within a range from about 5 Hz to about 20 Hz, but possibly as high as about 50 Hz. For valve mechanisms whose frequency response extends beyond such a range, the mechanism will experience significant reciprocal motion as it follows the pulse waveform.

Because the valve is not pressure-regulated, flow rate will also be a function of the pressure differential across the valve ports. Temperature and voltage variations may also influence the relationship.

It is known that the use of a linear solenoid can improve control accuracy, and Fig. 3 shows an example of a linear solenoid purge valve 14', certain parts of which correspond to parts of valve 14 already mentioned, and they will be designated by corresponding primed reference numerals.

Valve 14' comprises a two-piece body B1, B2 having an inlet port 14i' and an outlet port 14o'. Valve 14' has a longitudinal axis AX, and body piece B1 comprises a cylindrical side wall 40 that is coaxial with axis AX and that is open at its upper axial end where it is in assembly with body piece B2. Side wall 40 comprises upper and lower side wall portions 40A, 40B joined by a shoulder 42; the former side wall portion is fully cylindrical while the latter is cylindrical except in the region where it is radially intercepted by port 14o'. Port 14i' is in the shape of an elbow that extends from the lower axial end of side wall 40. By itself, body piece B1 is enclosed except for its open upper axial end and the two ports 14o' and 14i'.

A linear solenoid S is disposed in body piece B1, having been introduced through the open upper end of body piece B1 during fabrication of the valve. The solenoid comprises a bobbin 44, magnet wire wound on bobbin 44 to form a bobbin-mounted electromagnetic coil 46, and stator structure associated with the bobbin-coil. This stator structure comprises an upper stator end piece 48 disposed at the upper end of the bobbin-mounted coil, a cylindrical side stator piece 50 disposed circumferentially around the outside of the bobbin-mounted coil, and a lower stator end piece 52 disposed at the lower end of the bobbin-mounted coil.

Upper stator end piece 48 includes a flat circular disk portion whose outer perimeter fits to the upper end of side piece 50 and that contains a hole into which a bushing 54 is pressed so as to be coaxial with axis AX. The disk portion also contains another hole to allow for upward passage of a pair of bobbin-mounted electrical terminals 56 to which ends of magnet wire 46 are joined. Piece 48 further comprises a cylindrical neck 58 that extends downward from the disk portion a certain distance into a central through-hole in bobbin 44 that is co-axial with axis AX. The inner surface of neck 58 is cylindrical while its outer surface is frusto-conical so as to provide a radial thickness that has a progressively diminishing taper as the neck extends into the bobbin through-hole.

Lower stator end piece 52 includes a flat circular disk portion whose outer perimeter fits to the lower end of side piece 50 and that contains a hole into which a bushing 60 is pressed so as to be coaxial with axis AX. Piece 52 further comprises an upper cylindrical neck 62 that extends upwardly from the disk portion a certain distance into the central through-hole in bobbin 44 and that is co-axial with axis AX. Neck 62 has a uniform radial thickness. Piece 52 still further comprises a lower cylindrical neck 64 that extends downward from the disk portion a certain distance so that its lowermost end fits closely within lower side wall portion 40B. A valve seat element 66 is necked to press-fit into the open lower end of neck 64 and is sealed to the inside of wall portion 40B by an O-ring 67. Above the lowermost end that fits to side wall 40, neck 64 contains several through-holes 68 that provide for communication between port 14o' and the space disposed above seat element 66 and bounded by neck 64. Side wall 40 allows this communication by not restricting through-holes 68.

Bushings 54 and 60 serve to guide a valve shaft 70 for linear travel motion along axis AX. A central region of shaft 70 is slightly enlarged for press-fit of a tubular armature 72 thereto. The lower end of shaft 70 comprises a valve 74 that coacts with valve seat element 66. Valve 74 comprises a head, integrally formed with shaft 70 and having the general shape of a tapered pintle, comprising a rounded tip 74A, a frustoconical tapered section 74B extending from tip 74A, a grooved

cylindrical section 74C extending from section 74B, and an integral back-up flange 74D that in part defines the upper axial end of the groove of section 74C. An O-ring type seal 76 of suitable fuel-resistant elastomeric material is disposed in the groove of section 74c.

Seat element 66 comprises an inwardly directed shoulder 66A that contains a portion of a through-hole that extends axially through the seat element. This portion of the through-hole comprises a straight cylindrical section 78 and a frustoconical seat surface 80 that extends from the upper end of section 78 and is open to the interior space bounded by neck 64. The remainder of the through-hole axially below section 78 is designated by the reference numeral 81.

The upper end of shaft 70 protrudes a distance above bushing 54 and is shaped to provide for attachment of a spring seat 79 thereto. With piece B2 being attached to piece B1 by a clinch ring 82 which grips confronting, mated flanges to sandwich a seal 84 between them, a helical coiled linear compression spring 86 is captured between seat 79 and another spring seat 87 that is received in a suitably shaped pocket of piece B2. A calibration screw 88 is threaded into a hole in the end wall of this pocket coaxial with axis AX, and it is externally accessible by a suitable turning tool (not shown) for setting the extent to which spring seat 87 is positioned axially relative to the pocket. Increasingly threading screw 88 into the hole increasingly moves seat 87 toward spring seat 79, increasingly compressing spring 86 in the process. Terminals 56 are also joined with terminals 90 mounted in piece B2 to form an electrical connector 92 for mating engagement with another connector (not shown) that connects to drive circuit 26.

In the valve closed position shown in Fig. 3, a rounded surface portion of seal 76 has circumferentially continuous sealing contact with seat surface 80 so that the valve closes the flow path between ports 14o' and 14i'. In this position the upper portion of armature 72 axially overlaps the air gap that exists between the upper end of neck 62 and the lower end of neck 58, but slight radial clearance exists so that armature 72 does not actually touch the necks, thereby avoiding magnetic shorting.

Generally speaking, the degree of valve opening depends on the magnitude of electric current flow through the solenoid coil 46 so that the purge flow through the valve is effectively controlled by controlling the electric current flow through the coil. As the magnitude of electric current flow progressively increases from zero, it reaches a value sufficient to break whatever stiction exists between the seated O-ring 76 and seat surface 80. At that point the valve mechanism begins to open against the opposing force of spring 86. Valve opening commences as soon as O-ring seal 76 loses contact with seat surface 80.

Depending on the specific geometric relationships that are present between the valve pintle, its O-ring seal, and the angle of the valve seat surface, a certain initial axial travel of the pintle that unseats O-ring seal 76 from seat surface 80 may have to occur before tapered section 74B can become effective by itself to set the effective flow area through the seat element through-hole. In other words, it is only after the valve has traveled more than some initial minimum travel distance that the tapered section can become effective by itself to control the area open to flow. Beyond this initial minimum, the open area progressively uniformly increases as the pintle is increasingly positioned away from the seat element.

A representative graph plot of fluid flow rate vs. electric current reveals three distinct spans: a first span where current increases without any valve opening; a second span where the valve begins to open but the tapered section 74B is not yet fully effective to control the flow by itself; and a third span where the valve has opened sufficiently to allow section 74B to alone control the flow. The second span may be characterized by a relationship wherein a small incremental change in average electric current in solenoid S causes an incremental change in fluid flow rate that is substantially different from the incremental change results when the valve operates instead within the third span.

Coil 46 of solenoid S is connected across a source of D.C. voltage pulses, such as a pulse-width modulator circuit operating at a selected frequency. Electric current flow to the coil may be controlled by a solid-state driver in accordance with a control output signal from an engine management computer, and the circuit may include a feedback loop for feeding back a signal representative of electric current flow through the solenoid coil so as to endow the control with the ability to compensate for certain environmentally induced changes that could otherwise impair control accuracy. For example, the feedback loop can automatically regulate the current flow through coil 46 such that the influences of changes in ambient conditions, such as temperature and D.C. supply voltage to the circuit, are essentially negated, thereby enabling the valve to operate to a desired position commanded by the circuit substantially free of such influences.

Fig. 4 shows a mechanical pressure regulator 200 operatively associated with valve 14'. Pressure regulator 200 comprises a two-piece body 202 having a base 202b and a cover 202c, both of which are fabricated from suitable material, such as fuel tolerant injection molded plastic. Base 202b comprises an inlet port 204 and an outlet port 206 each of which is in the form of a nipple. A conduit 208 fluid connects port 204 with outlet port 14o' of valve 14', and outlet port 206 is fluid connected with engine intake manifold by another conduit that is not specifically illustrated in the Fig.

The nipple forming outlet port 206 comprises a radial segment that extends inwardly of body 202 to form an axial segment that is coaxial with an axis 210 of pressure regulator 200. This axial segment terminates as a circular rim forming a seal seat 212. Base 202b further comprises a cylindrical walled cup having a circular annular radial shoulder 214. This cup terminates in a circular rim 216 that is coaxial with axis 210.

Cover 202c has a generally circular shape whose outer periphery contains one or more catches 218 that attach the cover to the otherwise open end of the cup of base 202b at rim 216 by snapping over a lip of the rim as shown. The beaded outer circular perimeter of an impermeable flexible member 220 is held captured between the outer margin of cover 202c and rim 216 in a sealed manner. Centered with member 220 coaxial with axis 210 is a rigid circular disk 222. Secured centrally to disk 222 in confrontation to rim 216 is a circular seal element 224. In the illustrated embodiment, element 224 is secured to disk 222 by being molded onto the disk, with a portion of the molded material passing from the element, through a small hole in the center of the disk, to create an interlocking circular formation 226 on the opposite face of the disk.

It can be seen that the outer margin of disk 222 contains an annular area free of molded material. One end of a helical coiled compression spring 228 bears against this annular area. The opposite end of the spring bears against a wall of base 202b that extends circumferentially partially around the axial segment of the outlet port nipple below rim 212.

Cover 202c is formed with a central depression 230, and in the condition shown by Fig. 4, spring 228 is seen forcing disk 222 away from rim 212 such that the flat end surface of formation 226 is biased against the flat end surface of depression 230.

The assembled parts 220, 222, 224 form a fluid impermeable wall 232 that divides the interior of body 202 into first and second chamber spaces 234, 236. In the position shown by the Fig., chamber 236 provides free communication between ports 204 and 206. The flow path thus provided is depicted by the unnumbered arrows which represent purge flow from valve 14', through inlet port 202, through chamber space 236, and through outlet port 204 to the engine intake manifold.

Chamber space 234 is communicated to the fuel tank vapor headspace via a vapor collection (charcoal) canister 237. Cover 202c comprises a nipple 202n onto which one end of a tubular conduit 238 is fitted in a sealed manner. The opposite end of conduit 238 is fitted in a sealed manner onto a tee 14t that is fashioned with the nipple forming inlet port 14i'. Although not explicitly shown in the drawing, 202n could alternatively be directly communicated to the tank headspace by a devoted conduit, making tee 14t unnecessary.

Pressure regulator 200 operates in the following manner. For purposes of explanation, assume that it is in the position illustrated in the Fig., that equal pneumatic pressures exist in the two chamber spaces 234, 236, and that valve 14' is open. The creation of increasing intake manifold vacuum in chamber space 236 will begin to create an increasing pressure differential on wall 232. At a certain differential, the bias force of spring begins to be overcome, and the central region of wall 232 begins moving toward rim 212. Tank headspace pressure is maintained in chamber space 234 because vapor is drawn through canister 237, inlet port 14i', tee 14t, and conduit 238 as wall 232 moves toward rim 212. When the vacuum has increased to a certain larger magnitude, seal element 224 will be sufficiently close to rim 212 to create a restriction of the purge flow. The seal element may actually close on rim 212, albeit only momentarily. Such restriction or closure, tends to reduce the pneumatic pressure differential acting on wall 232 so that spring 228 then tends to move the central region of the wall away from rim 212. Tank vapor pressure in chamber space 234 is maintained as vapors are forced out opposite the direction they flowed in.

The overall effect is such that sealing element 224 will assume an average position that causes the vacuum in chamber space 236 to be regulated to a predetermined magnitude that is substantially independent of the magnitude of intake manifold vacuum and that causes flow through the valve to be essentially independent of change in pressure at the valve inlet. Hence, a substantially constant pressure differential is maintained across valve 14'. Now as valve 14' operates to different positions as commanded by the signal applied to solenoid S, the commanded positions will produce substantially the correspondingly intended purge flow rate, substantially free of variation in intake manifold vacuum and tank headspace pressure. Because flexible member 220 is provided with a convolution, it imposes no restriction of the movement of the central region of the movable wall relative to the open end of the walled axial conduit segment that contains rim 212.

In distinction to prior uses of pressure regulators in conjunction with pulsating purge valves, the disclosed embodiment, when operated at a fundamental pulse waveform frequency substantially greater than the frequency response of the valve mechanism, does not utilize a pressure regulator for the purpose of dampening purge flow pulsations. Rather, the creation of a predetermined pressure differential acting across valve 14' enables a given command signal to directly provide the intended flow rate, free of manifold vacuum variations and tank headspace pressure variations. It is believed that this can eliminate the need for the engine management computer to include a map for processing an input representing intake manifold vacuum and a map for processing an input representing tank headspace pressure when the computer calculates what the command signal to the solenoid coil of the valve should be.

Fig. 5 shows a series of representative graph plots of purge flow rate through valve 14' vs. time-averaged D.C. current flow in the solenoid coil. Each graph plot corresponds to a different value of intake manifold vacuum as indicated in Fig. 5, but the important effect of pressure regulator 200 can be seen by the substantial congruence of graph plots for 200, 300, 400, 500, and 600 mm Hg intake manifold vacuum. In the examples of Fig. 5, purge flow commences at about 183 milliamps current for the substantially congruent plots.

Figs. 6 and 7 illustrate another embodiment in which an LSPV and a pressure regulator are integrated into a single assembly. Like reference numerals from the preceding Figs. are used to identify like parts, although from comparison it can be seen that certain parts differ in certain details of construction. Figs. 6 and 7 show that pressure regulator 200 has been integrated into the lower end of LSPV 14'. The nipples that formed valve outlet port 14o' and regulator inlet port 208 have been eliminated. The portion of the flow path downstream of the valve pintle is communicated to chamber space 236 directly within the body of the assembly.

Flexible member 220, seal element 224, and formation 226 are embodied as a single part that is created by insert molding onto disk 222. The interior of cover 202c comprises a circular ridge 202r against which a central annular region of wall 232 bears when spring 228 is biasing seal element 224 maximally away from rim 212.

The flow path that communicates chamber space 234 with the nipple forming inlet port 14i' comprises an internal tee passage 14t' extending from the nipple passage. An annular seal 241 that is integrally formed with flexible member 220 seals around the outside of the tee passage where it communicates to chamber space 234. Pressure regulator 200 and valve 14' of the Fig. 6 and 7 embodiment function in the same manner as described above for the earlier embodiment.

While the solenoid S shown in Fig. 6 functions in the same manner as the solenoid shown in Figs. 3 and 4, it differs in certain constructional respects. The coil-containing bobbin 44, 46 and stator parts 48, 50, and 52 are encased in an overmolding 300 to form an assemblage that also includes the body part B2 as part of the overmolding. The overmolding includes features forming the shell of connector 92. The accommodations for acceptance of spring 86 and its associated adjustment mechanism are provided by stator part 48.

Stator part 48 is a screw-machined part. Stator part 50 is a strap, rather than a full cylindrical tube. The two parts 48 and 50 are joined together. Part 48 has a head end passing through a hole in a radial portion of strap 50. Part 48 is staked over the margin of the hole in strap 50 to unite the two parts. An axial portion of strap 50 extends from the radially outer end of the strap's radial portion, passing axially over the exterior of coil 44 and extending into contact with stator part 52.

The parts that are to be overmolded are placed in a suitably shaped mold cavity in a machine that forms the overmold around them. As overmold material flows, it covers the coil/bobbin 44, 46, and stator parts as shown, but leaves the outer head end of stator part 48 exposed. This allows access to adjustment screw 88, which is threaded into stator part 48 and contains a polygonally shaped socket 88' for engagement by a correspondingly shaped adjustment tool (not shown). Stator part 48 also contains a hollow interior space for spring 86. One end of spring 86 seats on an internal axial end of screw 88, being centered by a nose in that axial end. The opposite end of spring 86 seats in a counterbore of armature 72.

The overmold material forms around a perimeter 399 of a molded plastic part 400 that integrates inlet port 14i', exit 206, and regulator base 202b thereby interlocking the overmold with part 400. Upon curing of the overmold material, the overmold assumes a final shape as shown.

The valve seat element 66 is assembled to part 400, the element having a lower cylindrical wall fitted in a sealed manner by an O-ring 402 to the open internal end (co-axial with axis AX) of the nipple that forms inlet port 14i'. Above the valve seat element's transverse wall that contains the through-hole controlled by valve 74, the cylindrical tubular wall of the seat element contains several circumferentially spaced apart windows to provide for vapor that has passed through the through-hole controlled by valve 74 to flow to an internal space of part 400 and thence enter regulator chamber space 236. The vapor flow path is indicated by the unnumbered arrows in Figs. 6 and 7. At its far end, the seat element comprises an annular flange that seats on perimeter 399 of part 400, and a circular rim that fits a short distance into stator part 52. The pintle shaft is guided by bushing 60 while the armature is guided by a thin-walled, non-ferromagnetic sleeve 408.

Embodiments utilizing the inventive principles may be constructed in diverse ways. Because automotive electronic technology commonly employs electronic processors, the development of the electric control signal for the solenoid may be accomplished by utilizing conventional software programming techniques to develop the desired waveform or waveforms for any specific control strategy.

While the present invention has been described with reference to a preferred embodiment as currently contemplated, it should be understood that the invention is not intended to be limited to that embodiment. Accordingly, the invention is intended to encompass various modifications and arrangements that are within the scope of the claims.

WHAT IS CLAIMED IS:

1. An electric-operated pressure-regulated fluid flow control valve comprising an inlet at which fluid flow enters a valve body, a valve mechanism that is positioned within the valve body by an electric control signal to control fluid flow through the valve body, and a pressure regulator comprising a flow path having an entrance through which fluid flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit from which fluid flow that has entered the pressure regulator flow path exits the pressure regulator flow path, said pressure regulator comprising a pressure regulating mechanism that regulates the pressure at the entrance of the pressure regulator flow path to a pressure that is essentially independent of pressure at the exit of the pressure regulator flow path and that causes flow through the valve to be essentially independent of change in pressure at the valve inlet.

2. A fluid flow control valve as set forth in claim 1 wherein said pressure regulator comprises a body enclosing an interior space, and said pressure regulating mechanism comprises a movable wall dividing the interior space into a first chamber space and a second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to pressure at the valve inlet.

3. A fluid flow control valve as set forth in claim 2 wherein the valve mechanism has a frequency response characteristic which renders the valve mechanism incapable of faithfully tracking the fundamental frequency of an electric control signal whose fundamental frequency is greater than a predetermined frequency that, when applied in control of the valve mechanism, positions the valve mechanism to a position corresponding to a most recent time average of the electric control signal free of any significant pulsing of the valve mechanism.

4. A fluid flow control valve as set forth in claim 2 wherein the exit of said pressure regulator flow path is communicated to a variable vacuum.

5. A fluid flow control valve as set forth in claim 2 wherein said pressure regulator entrance comprises an external nipple, and further including a conduit fitted to said nipple to convey fluid from the valve body to the nipple.

6. A fluid flow control valve as set forth in claim 2 wherein the valve body and the pressure regulator body are assembled together to form an enclosure through which fluid flow passes from the valve mechanism to the pressure regulating mechanism.

7. A fluid flow control valve as set forth in claim 1 wherein said pressure regulating mechanism comprises a movable wall separating a variable volume first chamber space from a variable volume second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to pressure at the valve inlet.

8. A fluid flow control valve as set forth in claim 7 wherein said movable wall comprises a rigid disk disposed centrally on said movable wall and a flexible member circumscribing said disk, and further including a seal element disposed centrally on said disk.

9. A fluid flow control valve as set forth in claim 8 wherein said pressure regulating mechanism includes a helical coiled spring having one axial end bearing against said disk and circumscribing said seal element.

10. A fluid flow control valve as set forth in claim 8 wherein said helical coiled spring and said seal element are disposed in said second chamber space.

11. A fluid flow control valve as set forth in claim 7 wherein said pressure regulator further comprises a walled conduit having an open end disposed in the second chamber space in juxtaposition to a central region of the movable wall and leading to the pressure regulator flow path exit, the movable wall further comprising a flexible convoluted member circumscribing the central region of the movable wall to allow unrestricted movement of the central region relative to the open end of the walled conduit.

12. A fluid flow control valve as set forth in claim 11 wherein said pressure regulating mechanism includes a spring bearing against the central region of the movable wall and urging the central region of the movable wall away from the open end of the walled conduit.

13. A fluid flow control valve as set forth in claim 12 wherein said spring comprises a helical coiled spring having an axial end bearing against the central region of the movable wall, said spring being disposed in said second chamber space.

14. A fluid flow control valve as set forth in claim 13 wherein the central region of the movable wall comprises a rigid disk against which the axial end of the spring bears.

15. A fluid flow control valve as set forth in claim 14 wherein the central region of the movable wall comprises a seal element disposed on a central region of the rigid disk in juxtaposition to the open end of the walled conduit and circumscribed by the axial end of the spring.

16. A fluid flow control valve as set forth in claim 1 wherein the valve mechanism comprises a linear solenoid actuator to which the electric control signal is applied.

17. A fluid flow control valve as set forth in claim 16 wherein the linear solenoid actuator comprises a bobbin, a coil on the bobbin to which the electric control signal is applied, stator structure associated with the coil, and an overmold joining the bobbin and the stator structure in assembly and covering the coil.

18. A fluid flow control valve as set forth in claim 16 further including an electric control circuit that applies the electric signal to the linear solenoid actuator at a fundamental frequency substantially greater than the frequency response characteristic of the valve mechanism.

19. An electric-operated pressure-regulated fuel vapor purge valve for purging fuel vapor from a fuel tank to an intake manifold of an internal combustion engine comprising a valve mechanism that is positioned within a valve body by an electric control signal to control flow through the valve body, and a pressure regulator comprising a flow path having an entrance through which flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit for communicating the pressure regulator flow path to an engine intake manifold, said pressure regulator comprising a pressure regulating mechanism that regulates the pressure at the entrance of the pressure regulator flow path to a pressure that is essentially independent of pressure at the exit of the pressure regulator flow path and

that causes flow through the valve to be essentially independent of change in pressure at the valve inlet.

20. A fuel vapor purge valve as set forth in claim 19 wherein said pressure regulator comprises a body enclosing an interior space, and said pressure regulating mechanism comprises a movable wall dividing the interior space into a first chamber space and a second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to the tank headspace.

21. A fuel vapor purge valve as set forth in claim 20 wherein said first chamber space is communicated to the tank headspace through a fuel vapor collection canister.

22. A fuel vapor purge valve as set forth in claim 20 wherein said pressure regulator entrance comprises an external nipple, and further including a conduit fitted to said nipple to convey flow from the valve body to the nipple.

23. A fuel vapor purge valve as set forth in claim 20 wherein the valve body and the pressure regulator body are assembled together to form an enclosure through which flow passes from the valve mechanism to the pressure regulating mechanism.

24. A fuel vapor purge valve as set forth in claim 19 wherein said pressure regulating mechanism comprises a movable wall separating a variable volume first chamber space from a variable volume second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to being communicated to the tank headspace.

25. A fuel vapor purge valve as set forth in claim 24 wherein said movable wall comprises a rigid disk disposed centrally on said movable wall and a flexible member circumscribing said disk, and further including a seal element disposed centrally on said disk.

26. A fuel vapor purge valve as set forth in claim 25 wherein said pressure regulating mechanism includes a helical coiled spring having one axial end bearing against said disk and circumscribing said seal element.

27. A fuel vapor purge valve as set forth in claim 26 wherein said helical coiled spring and said seal element are disposed in said second chamber space.

28. A fuel vapor purge valve as set forth in claim 24 wherein said pressure regulator further comprises a walled conduit having an open end disposed in the second chamber space in juxtaposition to a central region of the movable wall and leading to the pressure regulator flow path exit, the movable wall further comprising a flexible convoluted member circumscribing the central region of the movable wall to allow unrestricted movement of the central region relative to the open end of the walled conduit.

29. A fuel vapor purge valve as set forth in claim 28 wherein said pressure regulating mechanism includes a spring bearing against the central region of the movable wall and urging the central region of the movable wall away from the open end of the walled conduit.

30. A fuel vapor purge valve as set forth in claim 29 wherein said spring comprises a helical coiled spring having an axial end bearing against the central region of the movable wall, said spring being disposed in said second chamber space.

31. A fuel vapor purge valve as set forth in claim 30 wherein the central region of the movable wall comprises a rigid disk against which the axial end of the spring bears.

32. A fuel vapor purge valve as set forth in claim 31 wherein the central region of the movable wall comprises a seal element disposed on a central region of the rigid disk in juxtaposition to the open end of the walled conduit and circumscribed by the axial end of the spring.

33. A fuel vapor purge valve as set forth in claim 19 wherein the valve mechanism comprises a linear solenoid actuator to which the electric control signal is applied.

34. A fuel vapor purge valve as set forth in claim 33 wherein the linear solenoid actuator comprises a bobbin, a coil on the bobbin to which the electric control signal is applied, stator structure associated with the coil, and an overmold joining the bobbin and the stator structure in assembly and covering the coil.

35. A fuel vapor purge valve as set forth in claim 34 further including an electric control circuit that applies the electric signal to the linear solenoid actuator at a fundamental frequency substantially greater than the frequency response characteristic of the valve mechanism.

36. An electric-operated pressure-regulated fluid flow control assembly comprising an inlet at which fluid flow enters a body, a valve mechanism that is positioned within the body by an electric control signal to control fluid flow through the body, and a pressure regulator comprising a flow path having an entrance through which fluid flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit from which fluid flow that has entered the pressure regulator flow path exits the pressure regulator flow path, said pressure regulator including a movable wall separating a variable volume first chamber space from a variable volume second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to pressure at the inlet.

37. An electric-operated pressure-regulated fuel vapor purge valve for purging fuel vapor from a fuel tank to an intake manifold of an internal combustion engine comprising a valve mechanism that is positioned within a body by an electric control signal to control flow through the body, and a pressure regulator comprising a flow path having an entrance through which flow that has passed through the valve mechanism enters the pressure regulator flow path and an exit for communicating the pressure regulator flow path to an engine intake manifold, said pressure regulator including a movable wall separating a variable volume first chamber space from a variable volume second chamber space, said second chamber space forming a portion of the pressure regulator flow path, and said first chamber space being communicated to pressure at an inlet of the body.

FIG. 1

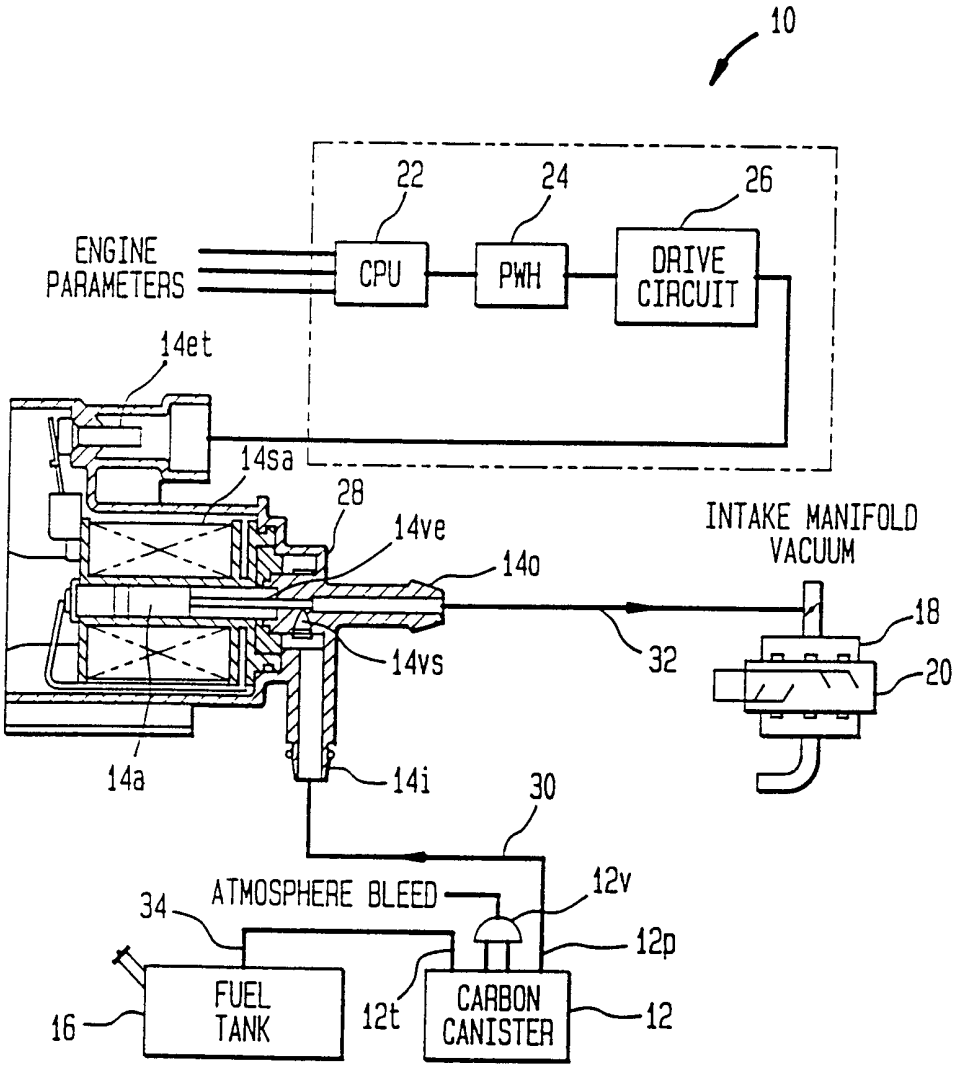

10
22
24
26
ENGINE PARAMETERS
CPU
PWM
DRIVE CIRCUIT
14et
14sa
28
14ve
14o
INTAKE MANIFOLD VACUUM
18
20
32
14vs
14a
14i
30
ATMOSPHERE BLEED
12v
34
12p
FUEL TANK
16
12t
CARBON CANISTER
12

FIG. 2

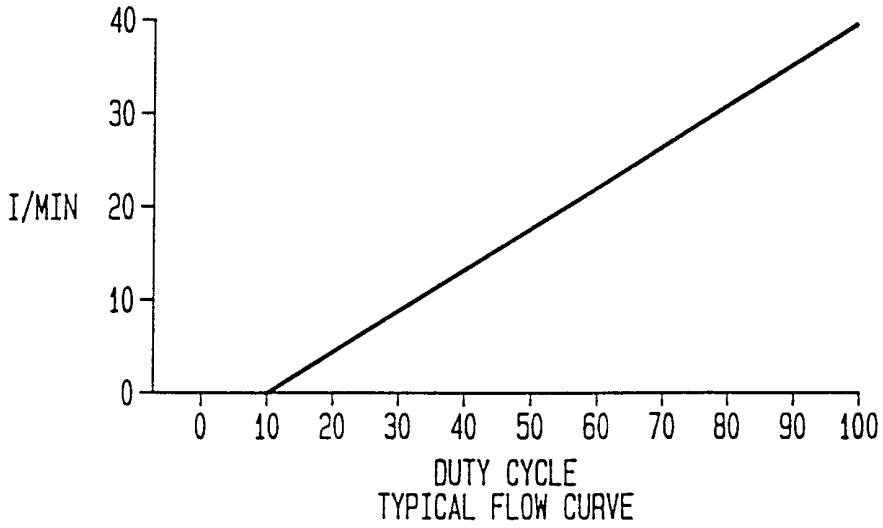

40
30
I/MIN 20
10
0
0 10 20 30 40 50 60 70 80 90 100
DUTY CYCLE
TYPICAL FLOW CURVE

FIG. 3

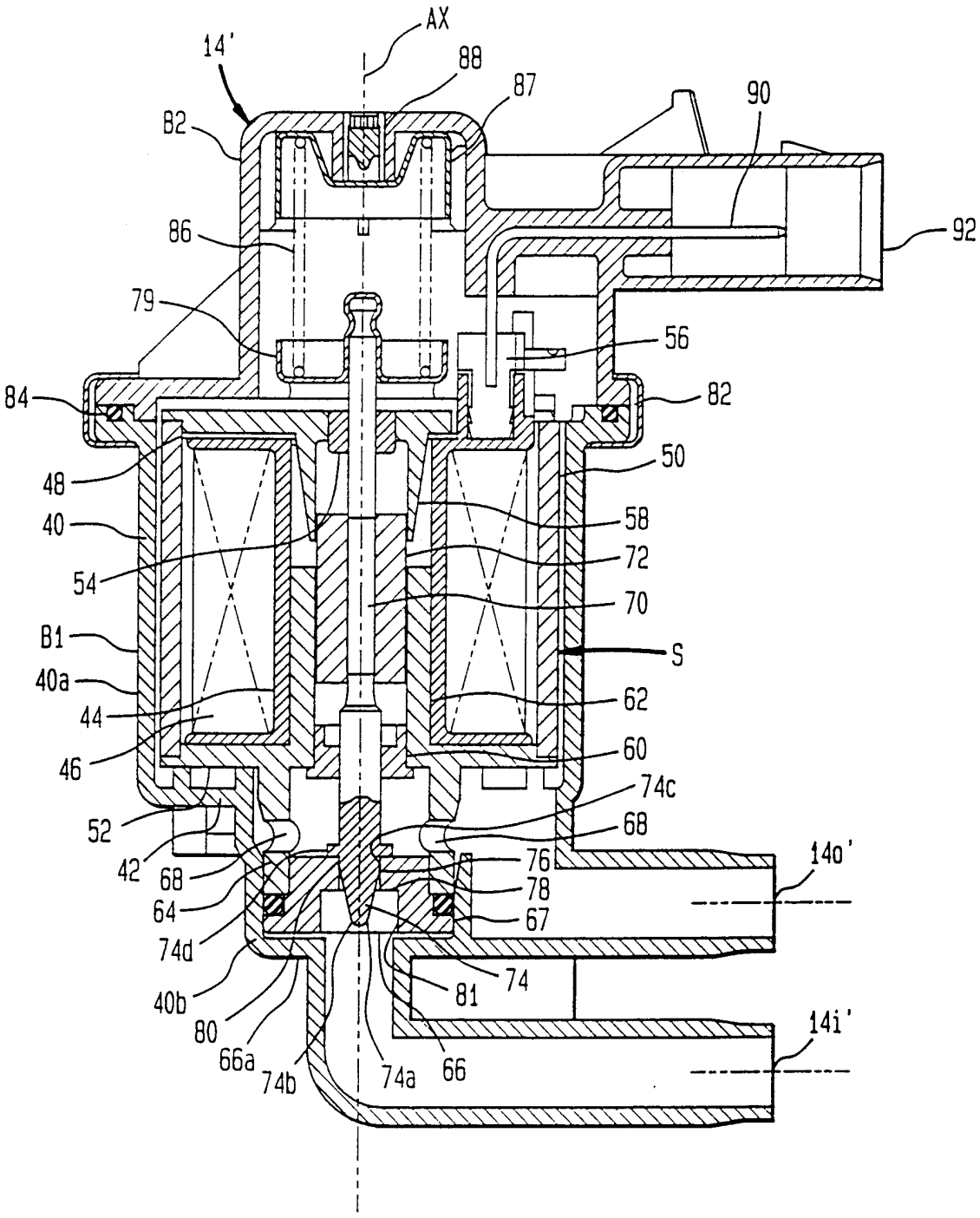

AX
14'
88
87
90
B2
92
86
79
56
84
82
50
48
58
40
72
54
70
B1
S
40a
62
44
60
46
74c
52
68
42
76
14o'
68
78
64
67
74d
74
81
40b
14i'
80
66a
74b
74a
66

FIG. 4

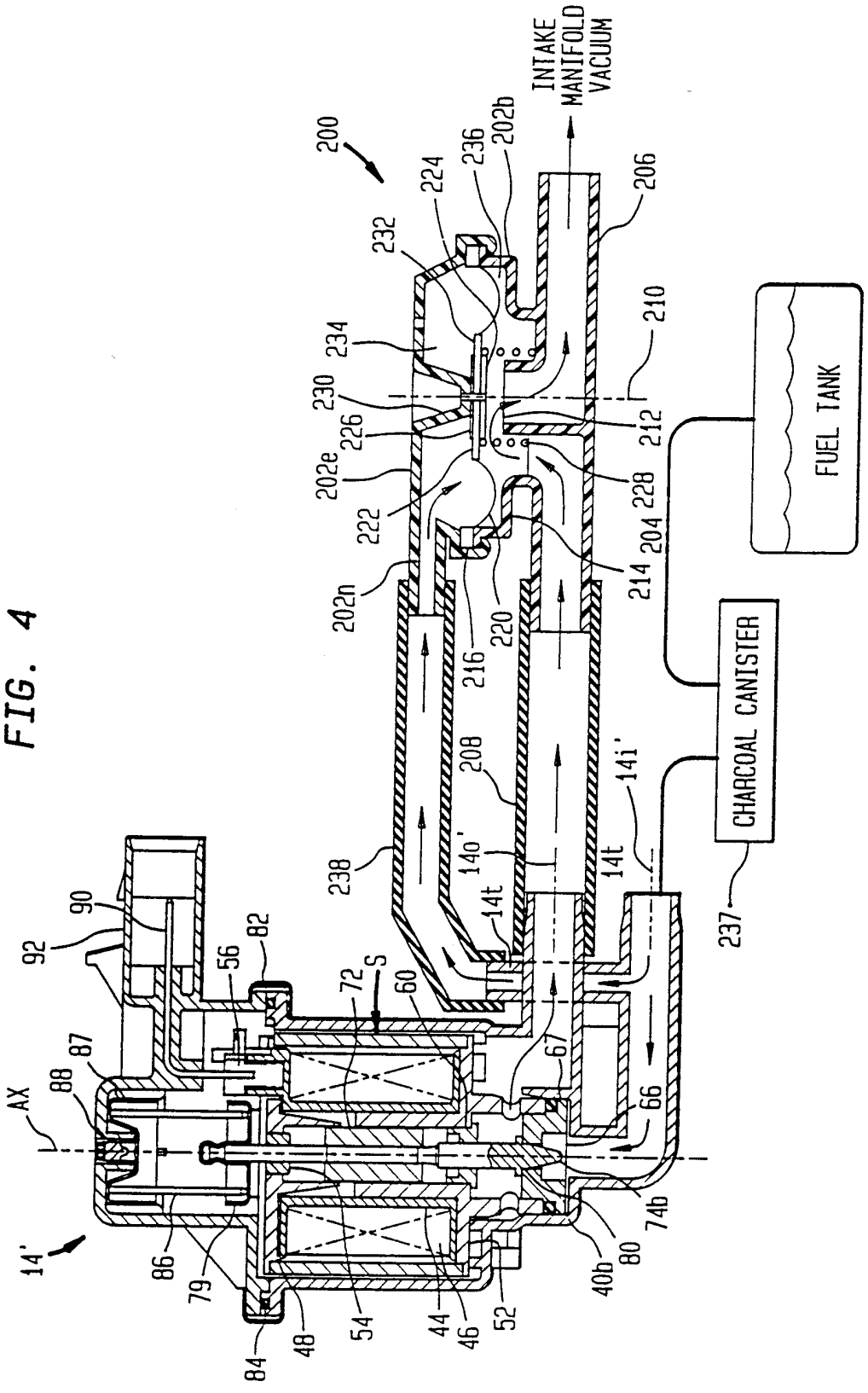

INTAKE MANIFOLD VACUUM
FUEL TANK
CHARCOAL CANISTER
200
202b
236
224
232
206
210
234
230
226
212
202e
228
222
204
214
202n
220
216
208
14i'
238
14o'
14t
237
92
90
82
56
72
S
60
67
87
88
AX
66
74b
14'
86
79
80
40b
48
54
44
46
52
84

FIG. 5

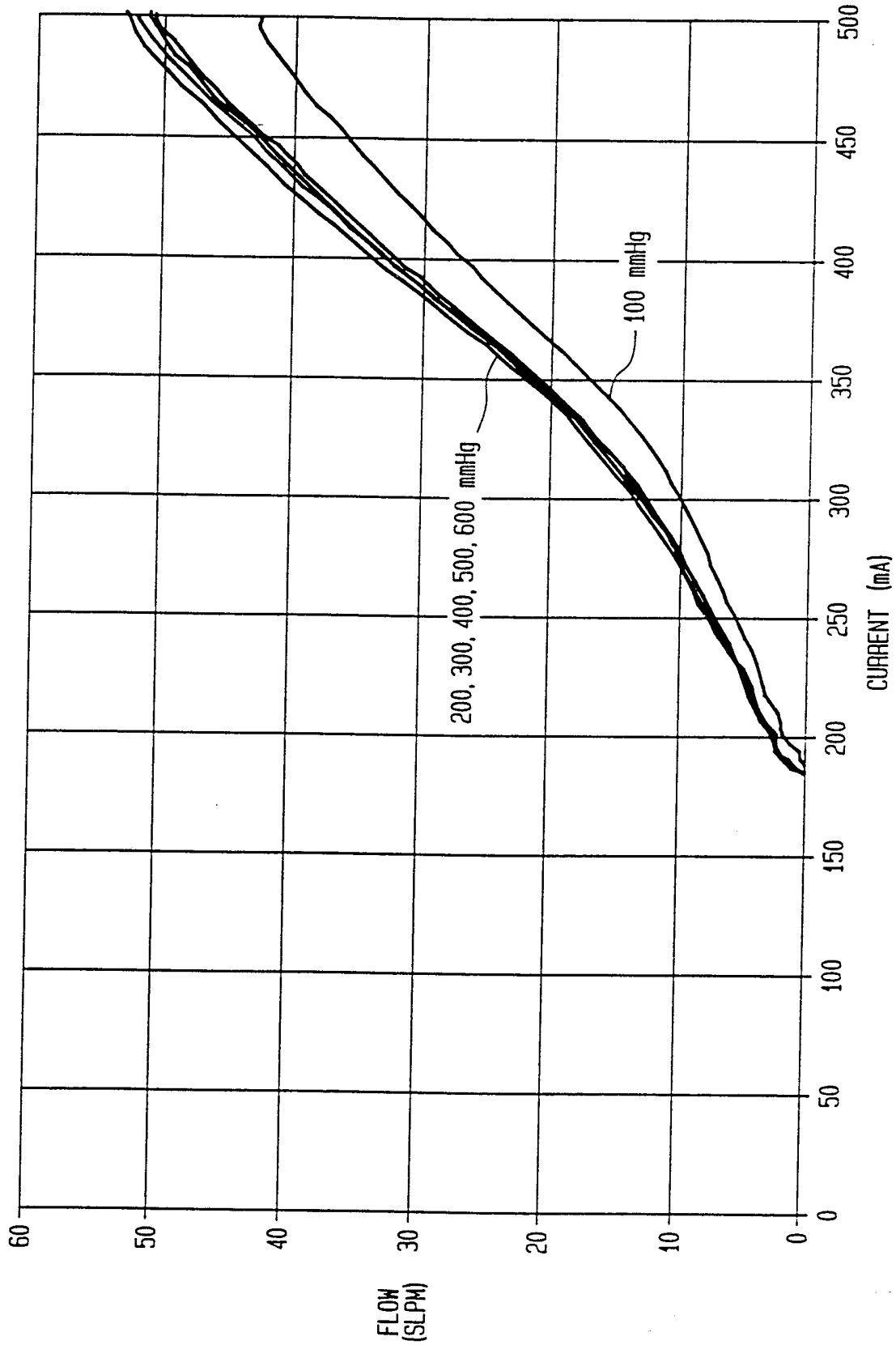

FLOW (SLPM)
60
50
40
30
20
10
0
CURRENT (mA)
0
50
100
150
200
250
300
350
400
450
500
100 mmHg
200, 300, 400, 500, 600 mmHg

FIG. 6

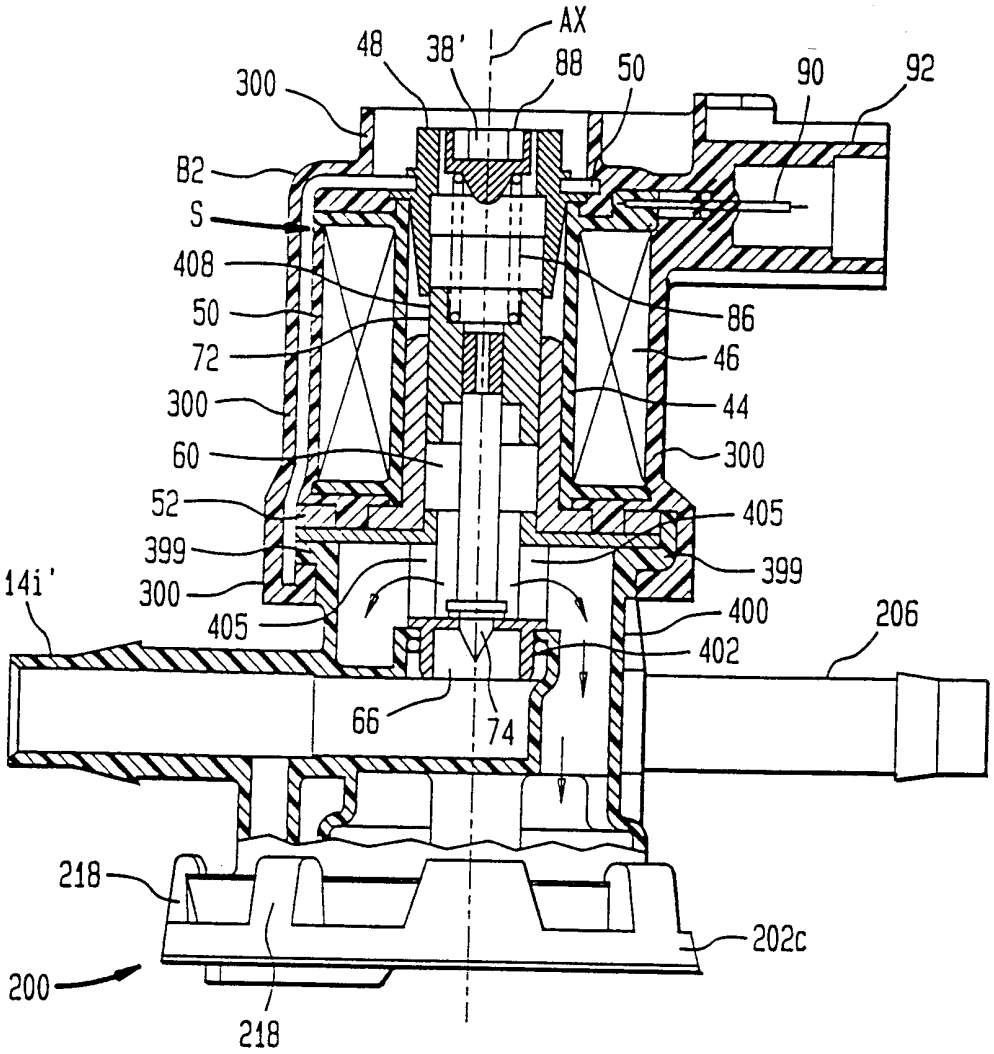
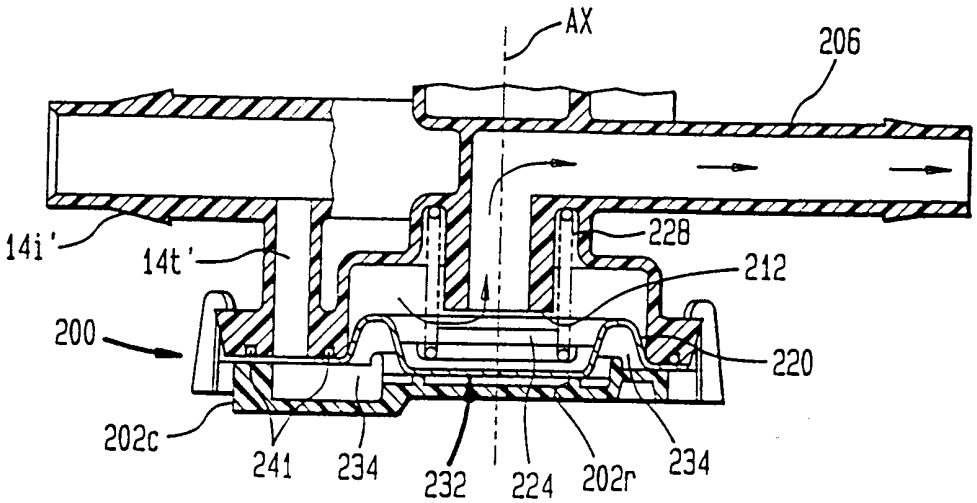

AX
48
38'
88
50
90
92
300
82
S
408
50
72
300
60
52
399
300
405
14i'
86
46
44
300
405
399
400
402
206
66
74
218
202c
200
218

FIG. 7


AX
206
14i'
14t'
228
212
200
220
202c
241
234
232
224
202r
234

INTERNATIONAL SEARCH REPORT

Inter nal Application No
PCT/CA 98/00663

A. CLASSIFICATION OF SUBJECT MATTER
IPC 6 F02M25/08

According to International Patent Classification (IPC) or to both national classification and IPC

B. FIELDS SEARCHED

Minimum documentation searched (classification system followed by classification symbols)
IPC 6 F02M

Documentation searched other than minimum documentation to the extent that such documents are included in the fields searched

Electronic data base consulted during the international search (name of data base and, where practical, search terms used)

C. DOCUMENTS CONSIDERED TO BE RELEVANT

Category °	Citation of document, with indication, where appropriate, of the relevant passages	Relevant to claim No.
X	US 5 069 188 A (SIEMENS AUTOMOTIVE LIMITED) 3 December 1991	1,2,4, 6-15, 19-21, 23-32, 36,37
A	see column 2, line 14 - column 3, line 67; figures	5,22
A	US 5 551 406 A (SIEMENS ELECTRIC LIMITED) 3 September 1996 see column 2, line 37 - column 3, line 13; figures	16,17,33
A	US 5 509 395 A (SIEMENS ELECTRIC LIMITED) 23 April 1996	
A	US 5 649 687 A (BORG-WAGNER AUTOMOTIVE INC) 22 July 1997	

☐ Further documents are listed in the continuation of box C.

☒ Patent family members are listed in annex.

° Special categories of cited documents :

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"O" document referring to an oral disclosure, use, exhibition or other means

"P" document published prior to the international filing date but later than the priority date claimed

"T" later document published after the international filing date or priority date and not in conflict with the application but cited to understand the principle or theory underlying the invention

"X" document of particular relevance; the claimed invention cannot be considered novel or cannot be considered to involve an inventive step when the document is taken alone

"Y" document of particular relevance; the claimed invention cannot be considered to involve an inventive step when the document is combined with one or more other such documents, such combination being obvious to a person skilled in the art.

"&" document member of the same patent family

Date of the actual completion of the international search	Date of mailing of the international search report
7 October 1998	14/10/1998

Name and mailing address of the ISA	Authorized officer
European Patent Office, P.B. 5818 Patentlaan 2 NL - 2280 HV Rijswijk Tel. (+31-70) 340-2040, Tx. 31 651 epo nl, Fax: (+31-70) 340-3016	Klinger, T

1

INTERNATIONAL SEARCH REPORT

Information on patent family members

Inte ›nal Application No
PCT/CA 98/00663

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