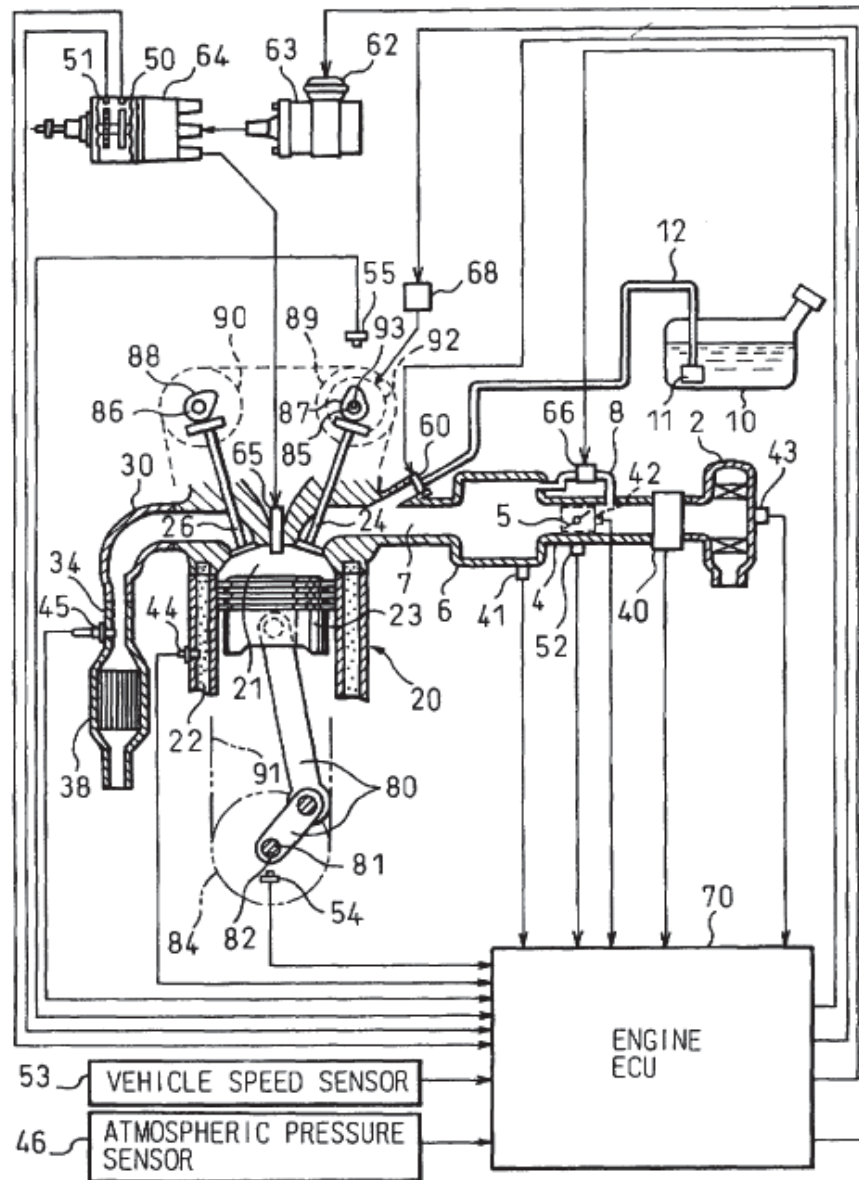


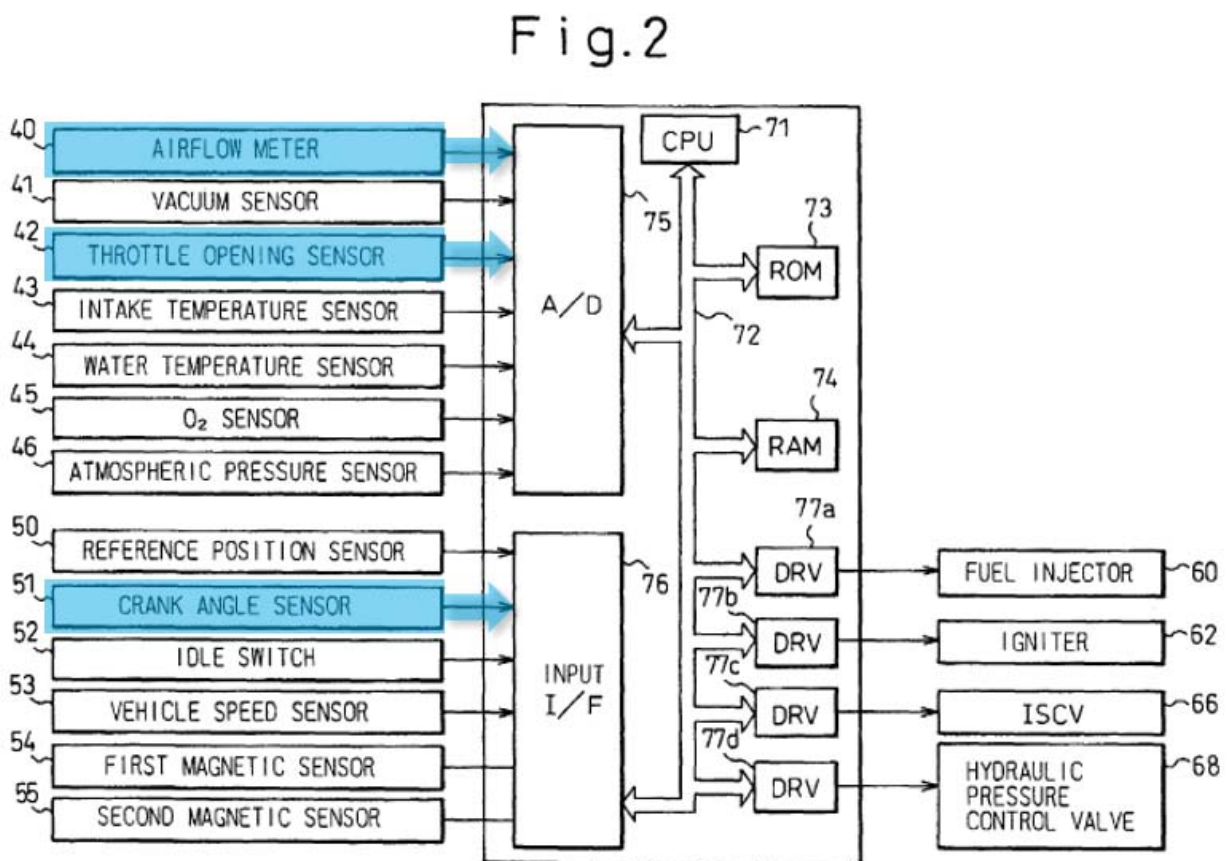
Fig. 1



EX1005, FIG. 1.

115. Figure 2 of Yoshioka illustrates the ECU 70 receiving input signals from the airflow meter 40, throttle opening sensor 42, and crank angle sensor 51,

and processing those input signals (together with the input signals from the other sensors 41 and 43-56) to generate engine control signals. EX1005, 5:25-31.



EX1005, FIG. 2 (annotated).

116. Yoshioka discloses that the engine revolution speed NE is calculated according to the output of the crank angle sensor 51. EX1005, 7:18-20. Yoshioka also discloses that the “airflow meter 40 measures the mass flow rate of intake air,” where the quantity of intake air is controlled by adjusting the throttle valve 5. *Id.*, 4:14-15, 5:43-46. The ECU 70 then “reads an engine load [GN] (the mass of intake air per engine revolution) calculated according to the mass flow rate of intake air

detected by the airflow meter 40” and the engine revolution speed NE. *Id.*, 7:20-24.

In my opinion, a POSA would have thus understood that the ECU 70 of Yoshioka receiving the signal from the crank angle sensor 51 indicating the engine revolution speed NE or the signal from the airflow meter 40 indicating the mass flow rate of intake air, both of which relate to the performance of Yoshioka’s engine, each correspond to the claimed “obtaining an engine performance command.”

117. Further, a POSA would have understood that a driver would have controlled the desired engine acceleration by opening and closing the throttle valve 5 detected by the throttle opening sensor 42 of Yoshioka, *e.g.*, by depressing a conventional accelerator pedal. For example, Toboldt discloses that:

The throttle valve is a device for varying the amount of air–fuel mixture that enters the intake manifold. This control is necessary so the speed of the vehicle can be changed. The throttle valve, Fig. 21-5, is simply a round disk mounted on a shaft, so that it can be tilted at various angles in the carburetor throttle valve body. It is connected by means of suitable linkage to the accelerator pedal in the driving compartment of the vehicle. **Depressing the pedal opens the throttle valve, permitting an increased amount of air-fuel mixture to reach the manifold.**

EX1011, 239.

118. Bosch similarly described that in “conventional injection systems, the driver directly controls the opening of the throttle valve by depressing the accelerator pedal” and that the “driver uses the accelerator pedal to vary the throttle valve’s aperture so that the airflow into the engine is varied and with it the engine’s power output.” EX1012, 452, 460; *see also* 478 (“At full load, the accelerator pedal is depressed fully and the throttle is wide open (WOT).”), 498 (“In order to set the operating state desired by the driver (main function), the position of the gas pedal is converted by the microprocessor into a setpoint value for the engine torque.”), 499 (“Systems with ETC (electronic throttle control, integrated in the engine control unit for ignition, injection and other auxiliary functions) detect the position of the accelerator pedal with a pedal-travel sensor (potentiometer). Taking into account the current engine operating status, the ECU calculates how far the throttle valve needs to open and actuated the throttle-valve drive accordingly (see Fig.). A throttle-valve angle sensor (potentiometer) monitors exact compliance with the desired throttle-valve position.”), 507 (“The most important torque demand derives from the vehicle-operator command generated by the accelerator pedal, from the position of which the engine-management system interprets the demand to be made on the IC engine for a particular torque.”)

119. Moreover, Probst discloses that: “Fuel delivery requirements depend more than anything else on how much work you are asking the engine to do—on

how much of a ‘load’ you are placing on it. To accelerate, you step down harder on the accelerator. This opens the throttle valve, increasing manifold pressure. The greater pressure difference between the manifold and the cylinders increases air flow, and therefore fuel flow, to increase power and accelerate the car.” EX1016, 2:7. And even the ’540 acknowledges that the claimed “engine performance command” can be obtained from a “conventional pedal.” EX1001, 2:6-8.

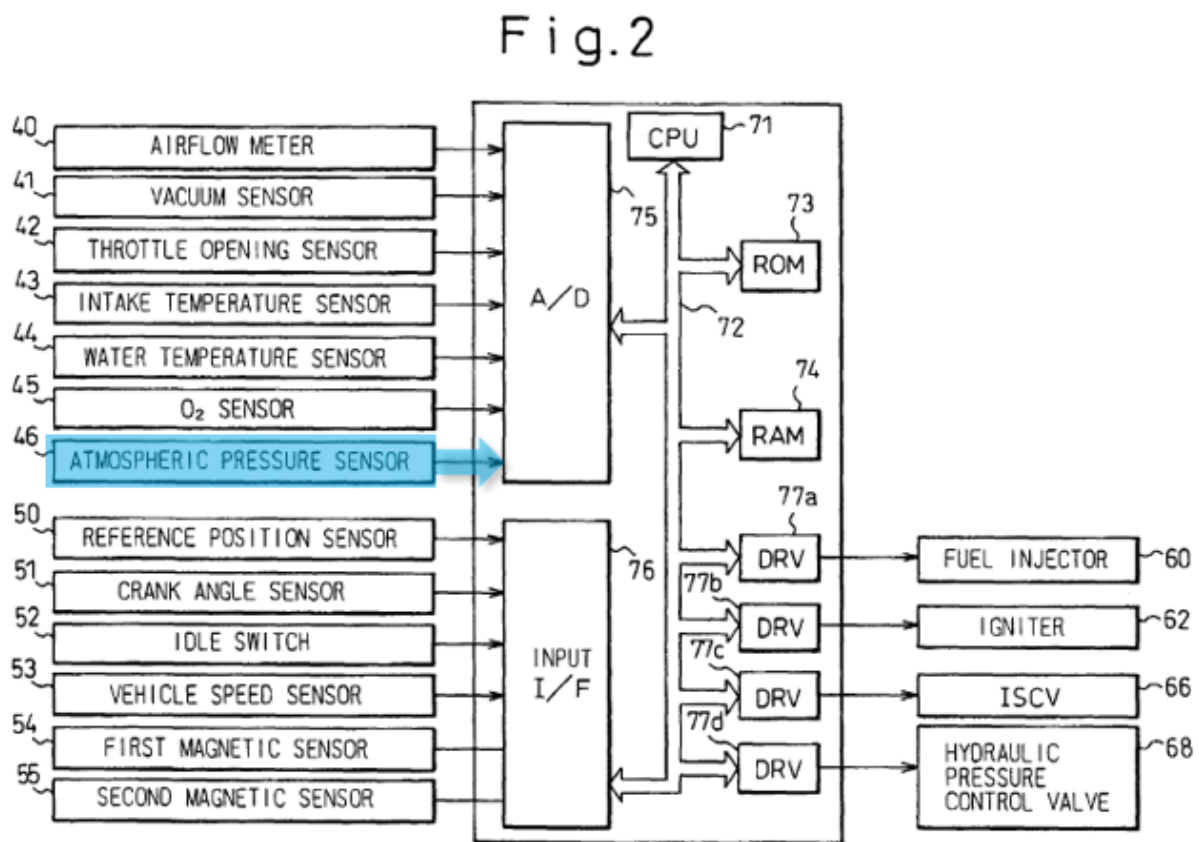
120. In my opinion, a POSA would have thus understood that Yoshioka’s ECU 70 receiving a signal from the throttle opening sensor 42 indicating the detected opening of the throttle valve 5 is a third instance of Yoshioka disclosing the claimed “obtaining an engine performance command.” Thus, Yoshioka discloses [1.A].

3. [1.B] “receiving an environmental conditions signal;”

121. The ’540 patent broadly characterizes the claimed “environmental conditions signal” as encompassing ambient pressure, temperature, or any other measureable environmental condition. EX1001, 2:23-35 (“The action of receiving an environmental conditions signal 18 preferably includes receiving an environmental conditions signal from an environmental sensor 30 in the vehicle. The environmental sensor 30 preferably senses the ambient temperature outside the vehicle and communicates this data to the control unit 10, which uses the data to determine the valve feedforward term. **Other information, such as the**

ambient pressure, may be useful in the determination of the valve feedforward term. For this reason, the environmental sensor 30 may alternatively sense other suitable information.”). In my opinion, a POSA would have thus understood that the “environmental conditions signal” to include ambient pressure, temperature, or any measureable information environmental condition.

122. Yoshioka discloses “receiving an environmental conditions signal.” As shown below in Figure 2 of Yoshioka, the ECU receives a signal input from the atmospheric (*i.e.*, ambient) pressure sensor 46 (*i.e.*, an “environmental conditions signal”), and then processes that input (together with inputs received from the other sensors 41 and 43-56) to generate engine control signals. EX1005, 5:25-30, 7:24-25 (“Step 106 reads an atmospheric pressure PA detected by the atmospheric pressure sensor 46.”); compare to EX1001, FIGs. 7, 9, 12, and 15 (showing steps for reading atmospheric pressure PA at 106, 206, 306, and 406, respectively).



EX1005, FIG. 2 (annotated).

123. Thus, Yoshioka discloses [1.B].

4. [1.C] “determining a valve feedforward term based on the engine performance command and the environmental conditions signal”

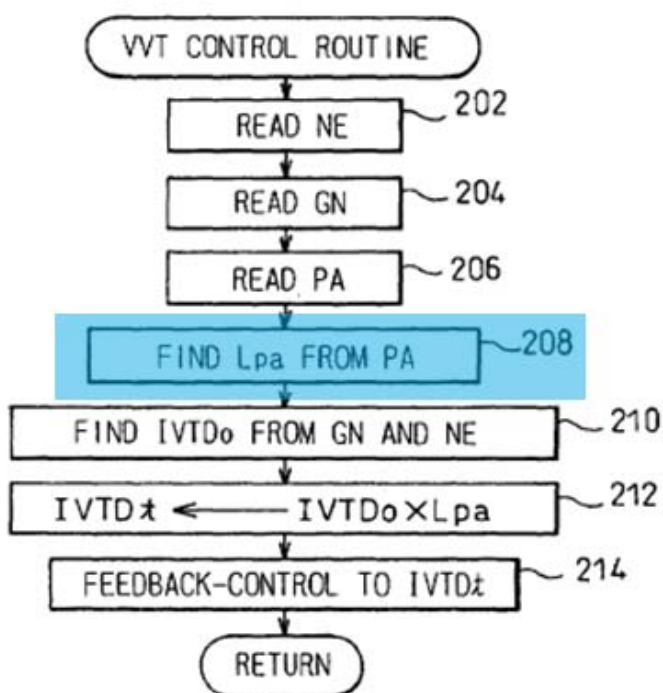
124. Yoshioka discloses “determining a valve feedforward term based on the engine performance command and the environmental conditions signal.”

Specifically, Yoshioka describes setting a target displacement to be imposed on the open/close timing of the intake valve 24 according to a detected engine load and decreasing the target displacement as an atmospheric pressure becomes lower, to

maintain the output of the engine at high altitudes under high load. EX1005, 7:46-53.

125. First, as shown in Figure 9 of Yoshioka below (step 208), the ECU determines a correction coefficient L_{pa} based on atmospheric pressure PA (“environmental conditions signal”) using a map stored in ROM 73 of the ECU. *Id.*, 7:57-60.

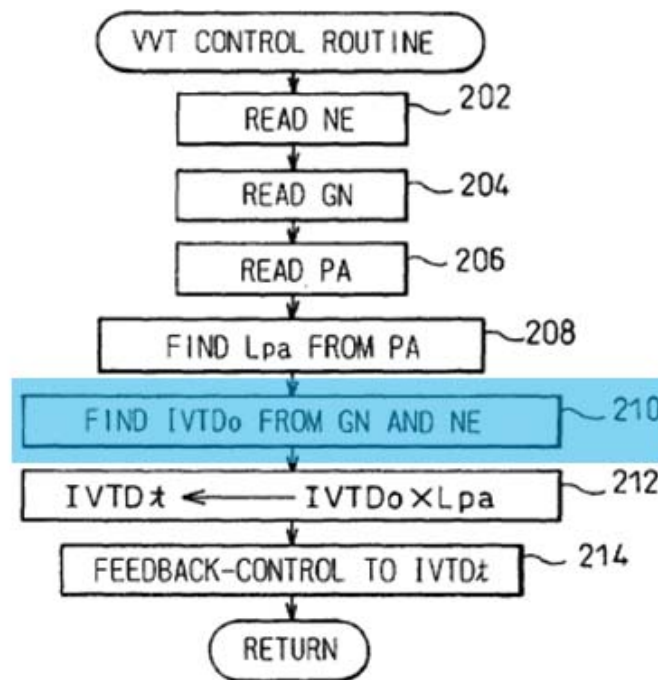
Fig. 9



EX1005, FIG. 9 (annotated).

126. Second, the ECU determines (step 210) an initial valve timing displacement IVTD0 based on the engine load GN and engine speed NE. EX1005, 7:63-67. Here, the engine load GN is “the mass of intake air per engine revolution” calculated according to the mass flow rate of intake air detected by the air flow meter 40 and controlled based on the opening of the throttle valve 5 (*i.e.*, the “engine performance command”). *Id.*, 7:20-24, 7:55-56, 6:37-42, 5:43-46 (“The mass of intake air per engine revolution is calculated according to the mass flow rate of intake air measured by the airflow meter 40 . . .”), 4:13-15 (“A throttle valve 5 arranged in the throttle body 4 adjusts the quantity of intake air. An airflow meter 40 measures the mass flow rate of intake air.”); *see also* EX1013, 14 (“The mass air flow rate into the manifold (m_θ) through the primary throttle body is a function of throttle angle (θ), the upstream pressure (P_θ), which we assume to be the atmospheric, *i.e.*, $P_\theta = 1$ bar, and the downstream pressure, which is the manifold pressure (P_m).”).

Fig. 9

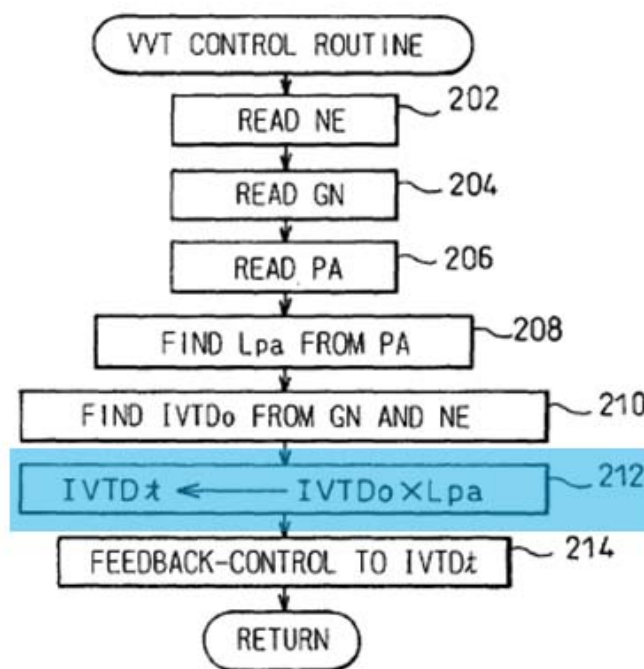


EX1005, FIG. 9 (annotated).

127. Third, the ECU determines (step 212) the target valve timing displacement, $IVTD_t$, (i.e., the “valve feedforward term”) based on (1) the L_{pa} (correction coefficient based on the “environmental conditions signal”) and the (2) $IVTD_0$ (initial valve timing displacement that is based on the “engine performance command”). EX1005, 7:47-8:5. A POSA would have understood the target valve timing displacement $IVTD_t$ to be a “valve feedforward term” because it is the displacement to be imposed on the open/close timing of the intake valve 24 instead of a measurement of the actual open/close timing of the intake valve 24 (i.e.,

feedback). See EX1010, McGraw-Hill Electronics Dictionary, 169 (defining “feed-forward control” as “Process control in which changes are detected at the process input, and an anticipating correction signal is applied before process output is affected.”).

Fig. 9



EX1005, FIG. 9 (annotated).

128. Thus, Yoshioka discloses [1.C].

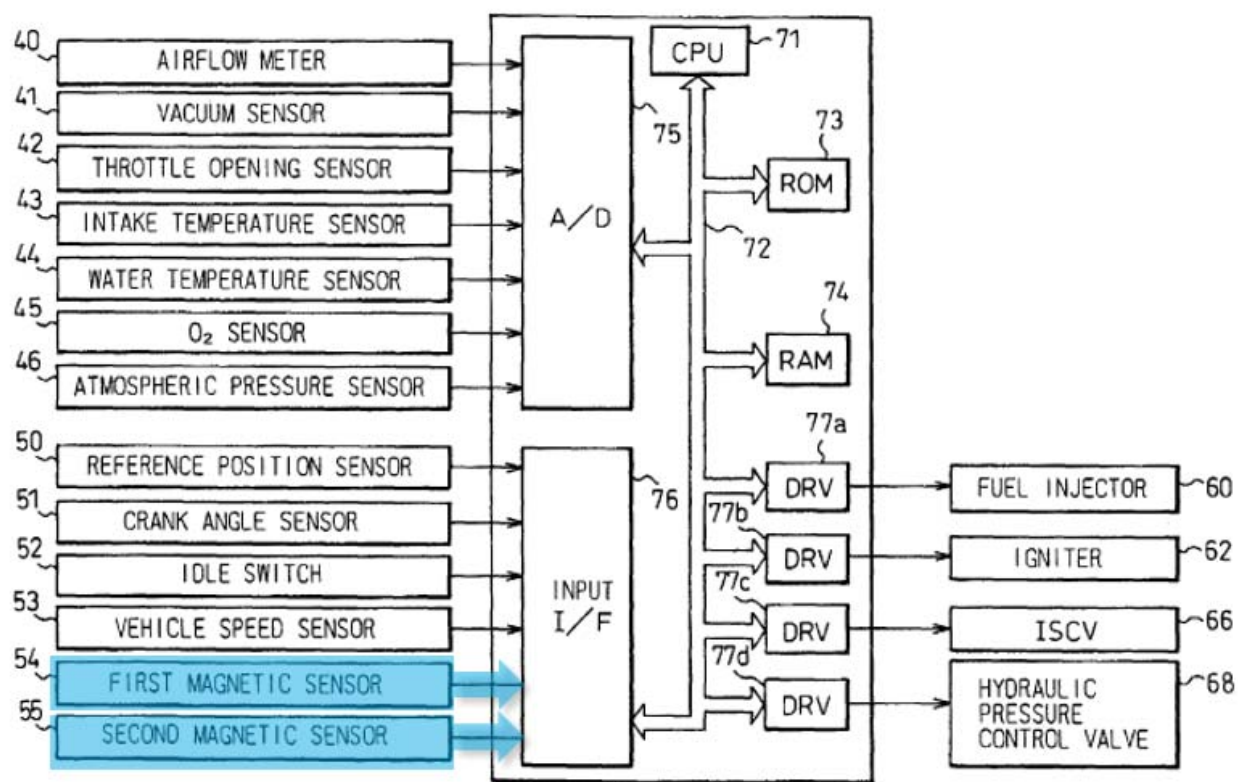
5. [1.D] “receiving an engine performance feedback;”

129. The '540 patent broadly characterizes the claimed “engine performance feedback” as encompassing any measureable information relating to engine performance including, for example, engine speed or engine torque. EX1001, 2:36-47 (“Similarly, the action of receiving an engine performance feedback 22 preferably includes receiving the engine performance feedback from an engine sensor 32 in the vehicle. The engine sensor 32 preferably senses the engine speed and communicates this data to the control unit 10, which uses the data to determine the valve feedforward term and the valve feedback term. **Other engine measurables**, such as engine torque data, **may be useful in the determination of the terms**. For this reason, **the engine sensor 32 may alternatively sense other suitable information. The engine sensor 32 is preferably a conventional engine sensor, but may alternatively be any suitable sensor.**”). In my opinion, a POSA would have thus understood the “engine performance feedback” to include any measureable information of an engine relating to engine performance.

130. Yoshioka discloses “receiving an engine performance feedback.” Specifically, Yoshioka’s ECU 70 receives signals input from magnetic sensor 54 and magnetic sensor 55 indicating an actual timing displacement of the intake valve 24 for “feedback-control.” EX1005, 5:4-9 (“The crank shaft 81 has a buried

magnet 82, so that a first magnetic sensor 54 arranged close to the crankshaft 81 may generate a reference pulse. The cam shaft 85 of the intake valve 24 has a buried magnet 93, so that a second magnetic sensor 55 arranged close to the cam shaft 85 may generate a reference pulse.”); 7:42-45 (“To bring an actual displacement to the target displacement IVTDt, signals from the first and second magnetic sensors 54 and 55 are monitored to feedback-control the hydraulic pressure control valve 68.”); 8:5-7 (“To bring the actual displacement of the intake valve 24 to the target displacement IVTDt, the hydraulic control valve 68 is feedback-controlled.”). This is an “engine performance feedback,” as claimed.

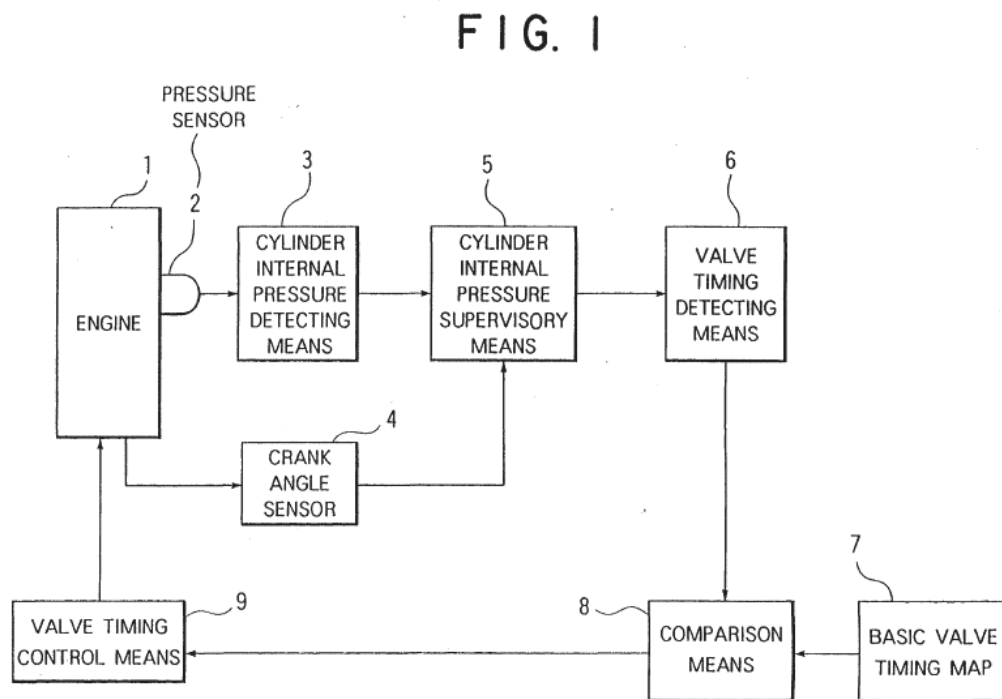
Fig.2



EX1005, FIG. 2 (annotated).

131. Accordingly, the actual timing displacement of Yoshioka's intake valve 24 corresponds to the claimed "engine performance feedback."

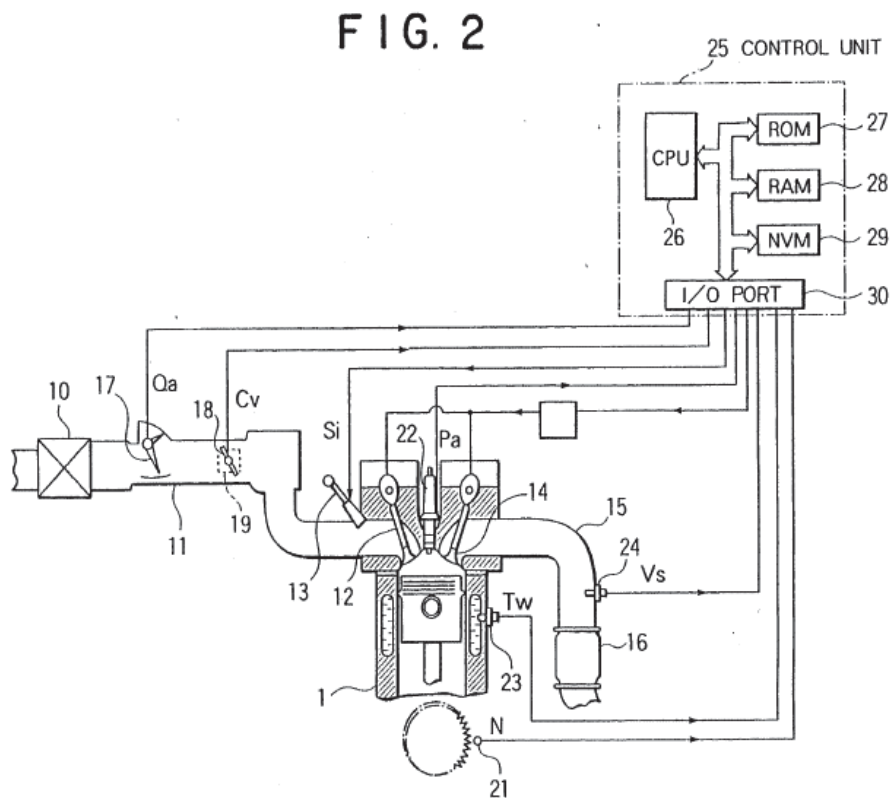
132. Ohkubo similarly discloses a valve timing control apparatus for an internal combustion engine having a valve timing detecting means 6, as shown below in Figure 1.



EX1006, FIG. 1.

Specifically, Ohkubo discloses detecting the opening and closing timings of the engine intake and exhaust valves by valve detecting means 6 based on an input from the cylinder internal pressure supervisory means. EX1006, 3:25-40. The control unit 25 having a CPU 26, as shown in Figure 2 below, determines the

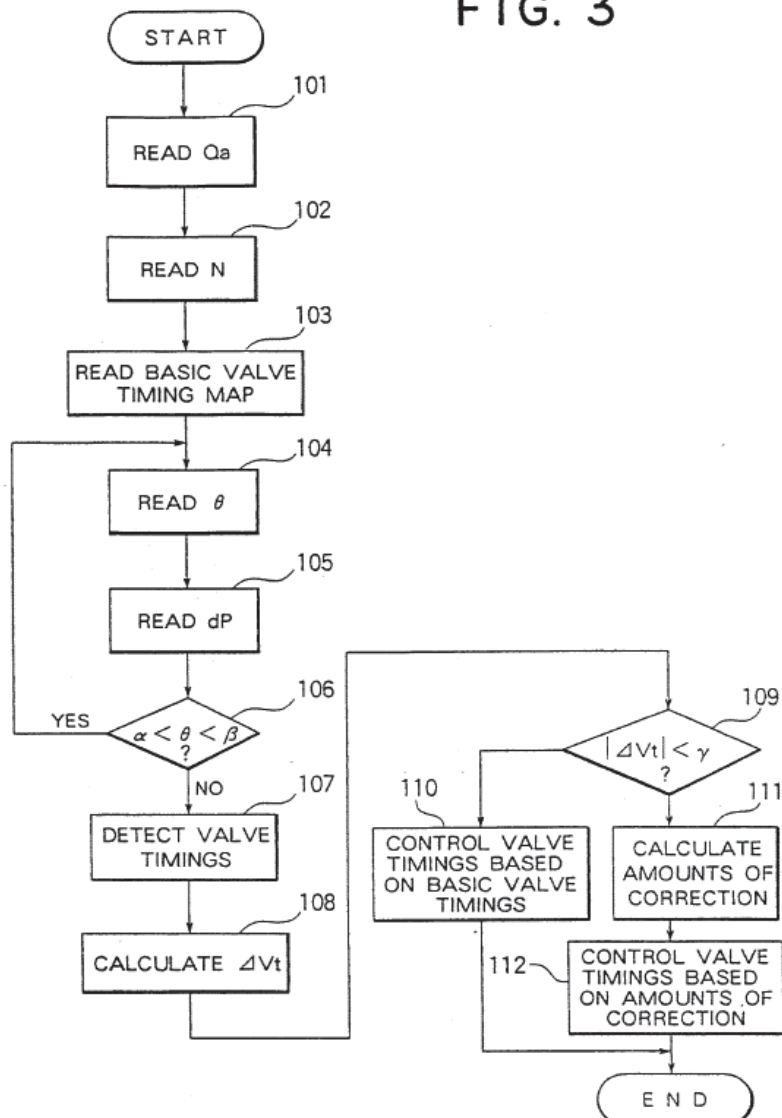
opening and closing timings of the intake and exhaust valves. *Id.*, 4:46-65. The detected opening and closing timings of the intake and exhaust valves are then compared with predetermined valve opening and closing timings to calculate deviations there between. EX1006, 4:46-65.



EX1006, FIG. 2.

133. Figure 3 of Ohkubo illustrates detecting the actual opening and closing timings of the intake and exhaust valves at step 107 by detecting a crank angle at which an abrupt change in pressure takes place. *Id.*, 5:38-41.

FIG. 3



EX1006, FIG. 3.

134. Accordingly, the detected opening and closing timings of the intake and exhaust valves (step 107) of Ohkubo also disclose the claimed “engine performance feedback.” Should MMT argue, incorrectly, that Yoshioka does not disclose the claimed “engine performance feedback,” it would have been obvious to implement this known technique of detecting opening and closing timings of the

intake and exhaust valves of Ohkubo in Yoshioka. Doing so would have merely been applying a known technique to a similar device in the same way.

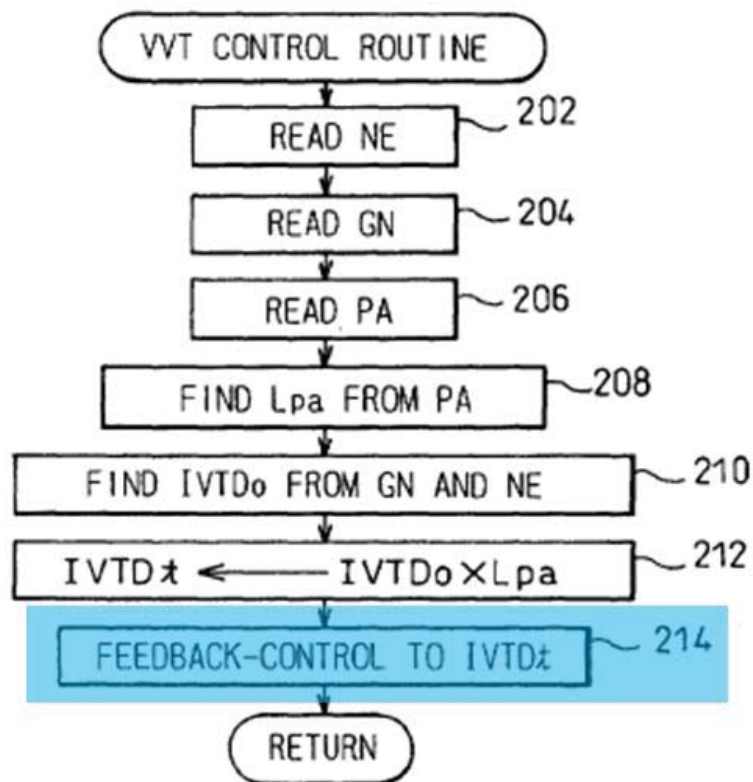
135. Thus, Yoshioka and Ohkubo both disclose [1.D].

6. [1.E] “calculating a valve feedback term based on the engine performance command and the engine performance feedback”

136. Yoshioka, alone or in combination with Ohkubo, renders obvious “calculating a valve feedback term based on the engine performance command and the engine performance feedback.” Yoshioka, in particular, discloses comparing the target timing displacement of the intake valve 24, IVTDt (“valve feedforward term,” which is based on the signal from the airflow meter 40—“engine performance command”) and the actual timing displacement of the intake valve 24 (“engine performance feedback”) to control the open/close timing of the intake valve 24. EX1005, 6:6-15 (“Valve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92. ...”); 7:42-45 (“To bring an actual displacement to the target displacement IVTDt, signals from the first and second magnetic sensors 54 and 55 are monitored to feedback-control the hydraulic pressure control valve 68.”); 7:47-56 (“The second embodiment sets a target displacement to be imposed on the open/close timing of the intake valve 24 according to a detected engine load and decreases the target displacement as an

atmospheric pressure becomes lower, to maintain the output of the engine at high altitudes under high load, similar to the first embodiment. ..."); 8:6-8 ("To bring the actual displacement of the intake valve 24 to the target displacement IVTD_t, the hydraulic control valve 68 is feedback-controlled.").

Fig. 9



EX1005, FIG. 9 (annotated).

137. Yoshioka does not explicitly state that a correction is calculated for the intake valve 24 based on this comparison. However, it is my opinion that a

POSA would have understood Yoshioka's disclosure of "feedback control" to mean calculating a correction based on the comparison of the target timing displacement, IVTDt, and the actual timing displacement of intake valve 24 to drive the actual timing displacement to the target timing displacement. See EX1013, 12-13; EX1010, 162, 168 (defining "feedback controller" as "[a] circuit or element in a *closed-loop system* that controls the motor or other actuator in a closed loop system to correct an *error signal*" and defining "error signal" as "[a] voltage or current proportional to the difference between the desired and actual response in a *closed-loop system*").

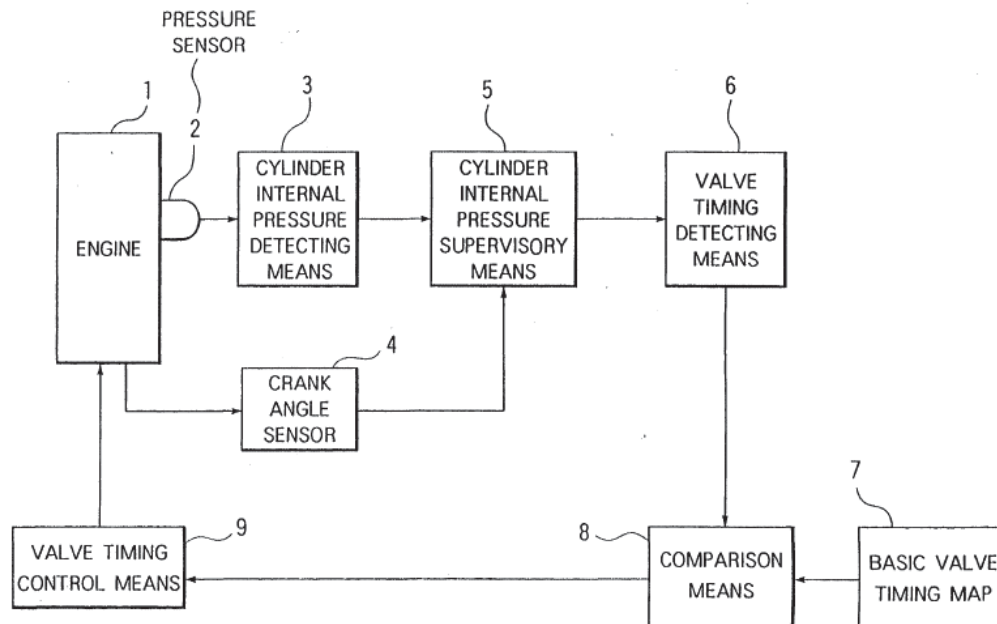
138. It is my opinion that a POSA would have thus understood that in order for Yoshioka's ECU 70 to "bring the actual displacement of the intake valve 24 to the target displacement IVTDt," it calculates a correction to the actual valve timing displacement by comparing the actual valve timing displacement and the target valve timing displacement IVTDt.

139. Accordingly, it is my opinion that a POSA would have understood, or at least found it obvious, that Yoshioka's ECU 70 calculates a correction for the timing of the intake valve 24 based on the comparison of the target valve timing displacement of the intake valve 24 IVTDt ("valve feedforward term," which is based on the signal from the airflow meter 40—"engine performance command")

and the actual valve timing displacement of intake valve 24 (“engine performance feedback”). Thus, a POSA would have understood Yoshioka discloses [1.E].

140. Should MMT argue, incorrectly, that Yoshioka does not disclose “calculating a valve feedback term based on the engine performance command and the engine performance feedback,” Ohkubo also discloses calculating a valve feedback term based on the engine performance command and the engine performance feedback, as claimed. As shown in Figure 1 of Ohkubo, the valve timing control apparatus includes a comparison means 8 for “compar[ing] the opening and closing timings ... detected by the valve timing detecting means 6” with predetermined target valve opening and closing timings (from basic valve timing map 7) and calculating deviations there between. EX1006, 3:33-40.

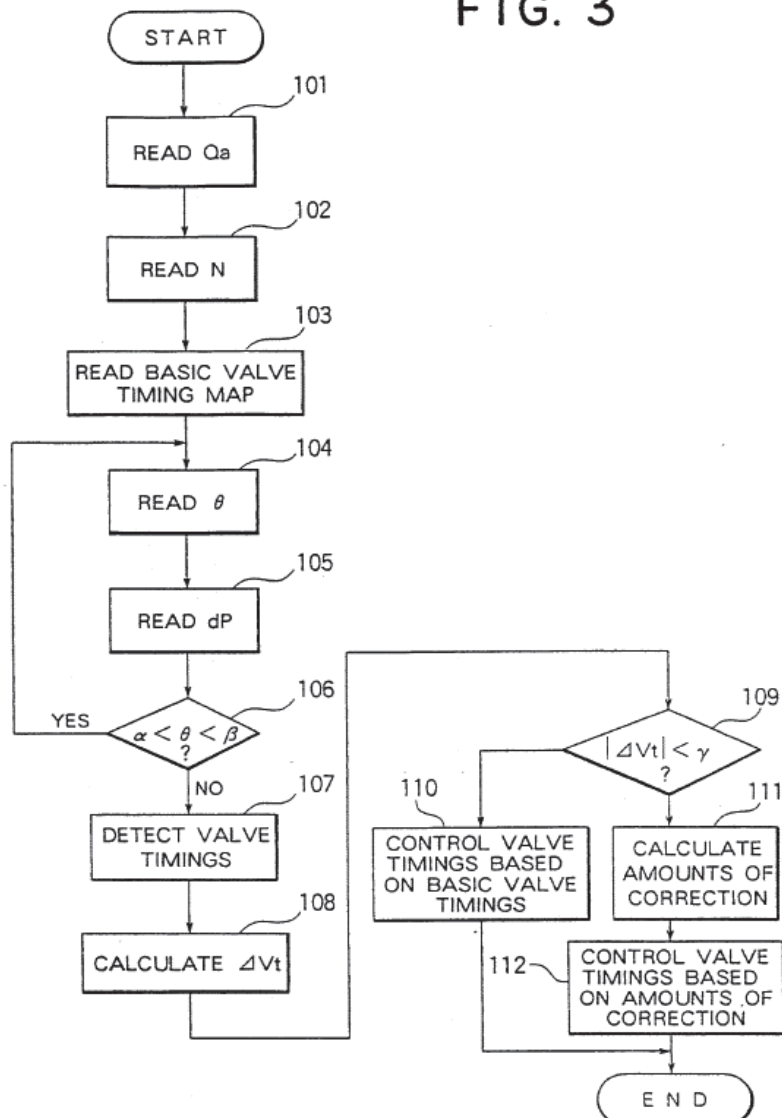
FIG. 1



EX1006, FIG. 1.

141. Figure 3 of Ohkubo also illustrates, at step 108, a deviation ΔVt is calculated between each of the detected timing values and the corresponding predetermined basic values for valve opening and closing timings; at step 109, it is determined whether each ΔVt falls within an allowable range γ . EX1006, 5:41-46. If the ΔVt is outside of the allowable range γ , the program goes to step 111 where appropriate amounts of correction for properly correcting the valve timings are calculated. *Id.*, 5:51-54.

FIG. 3



EX1006, FIG. 3.

142. Accordingly, Ohkubo discloses calculating a correction for the timing of the intake and exhaust valves based on the comparison of the predetermined valve timing of the intake and exhaust valves (“valve feedforward term” based on the “engine performance command”) and the actual valve timing of the intake and exhaust valves (“engine performance feedback”).

143. As described above, Yoshioka and Ohkubo both disclose an ECU 70/control unit 25 for controlling valve timing to bring actual valve timing to target valve timing. And it would have been obvious to combine the known technique of calculating a correction for the timing of the intake (and exhaust) valve based on the comparison of the predetermined valve timing of the intake (and exhaust) valve and the actual valve timing of Ohkubo so that Yoshioka's ECU 70 may accurately "bring the actual displacement of the intake valve 24 to the target displacement IVTDt." It is my opinion that doing so would have merely been applying a known technique to a similar device in the same way. *See supra* Section X.A.

144. Thus, Yoshioka, alone or in combination with Ohkubo, discloses or renders obvious [1.E].

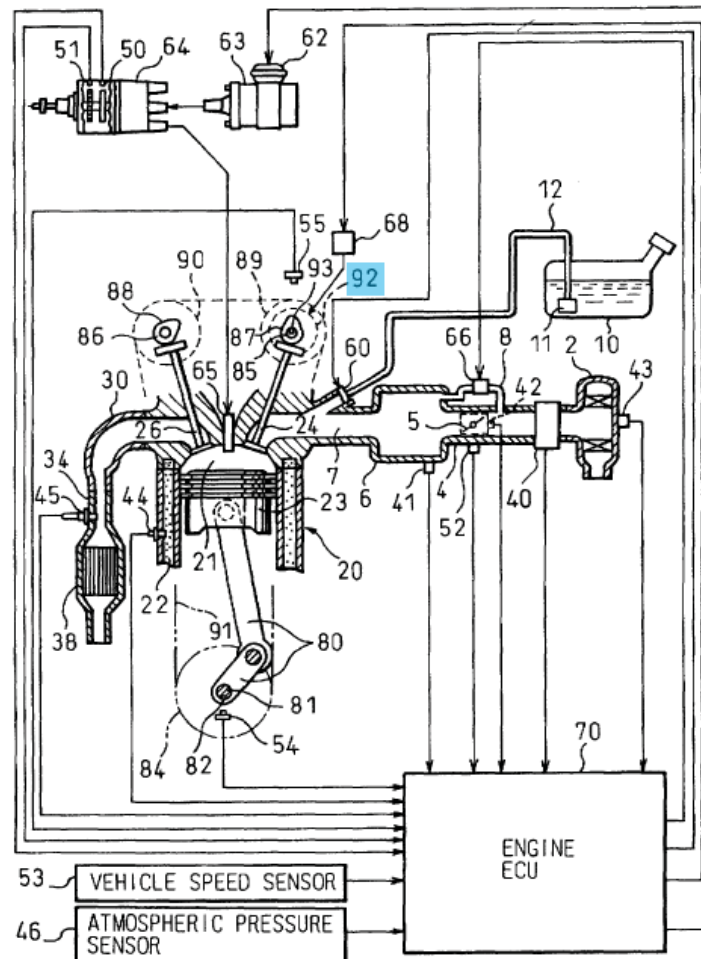
7. [1.F] "calculating a valve timing command based on the valve feedforward term and the valve feedback term."

145. The '540 patent broadly characterizes the claimed "valve timing command" as encompassing any "suitable parameter" for controlling valve timing. EX1001, 3:21-26 ("The action of calculating a valve timing command 26 preferably includes calculating an EVO command, an EVC/IVO command, and an IVC command. **The control unit 10, of course, may alternatively include other suitable parameters for the control of the variable timing valves.**"). It is my opinion that a POSA would have thus understood the claimed "valve timing command" to include any suitable parameter for controlling valve timing.

146. Yoshioka discloses “calculating a valve timing command based on the valve feedforward term and the valve feedback term.” Specifically, Yoshioka discloses that the ECU 70 feedback-controls the valve timing of the intake valve 24 by calculating an adjustment for the hydraulic pressure control valve 68 based on the correction for the timing of the intake valve 24 (“valve feedback term”) from the comparison of the target timing displacement IVTDt (“valve feedforward term”) and the actual timing displacement of intake valve 24 (“valve feedback term”).

147. Yoshioka, in particular, discloses a continuously variable mechanism 92 that uses the cam shaft 85 and timing pulley 89 as external gears and connects them to each other through an intermediate helical gear, which is axially moved by hydraulic pressure controlled by a control valve 68 to drive the cam shaft 85 and timing pulley 89 relative to each other. EX1005, 5:10-20.

Fig. 1



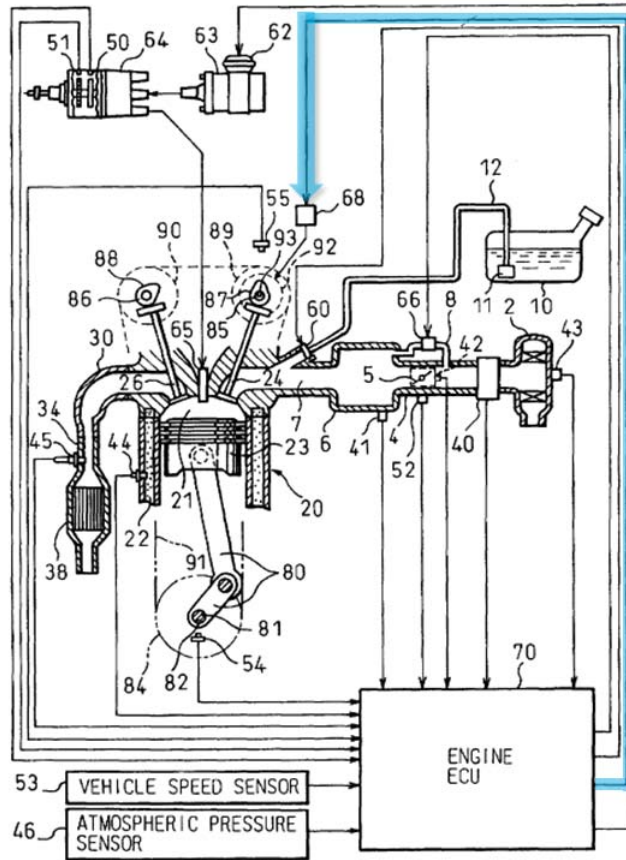
EX1005, FIG. 1 (annotated).

148. Yoshioka sets the target open/close timing of the intake valve 24 (*i.e.*, valve opening timing IVO and valve close timing IVC) using the target displacement according to operating conditions (*e.g.*, engine load and atmospheric pressure). EX1005, 6:6-8, 7:46-56. Once the target displacement IVTDt is set, the open/close timing of the intake valve 24 is feedback-controlled using the hydraulic pressure control valve 68 according to signals from the first and second magnetic

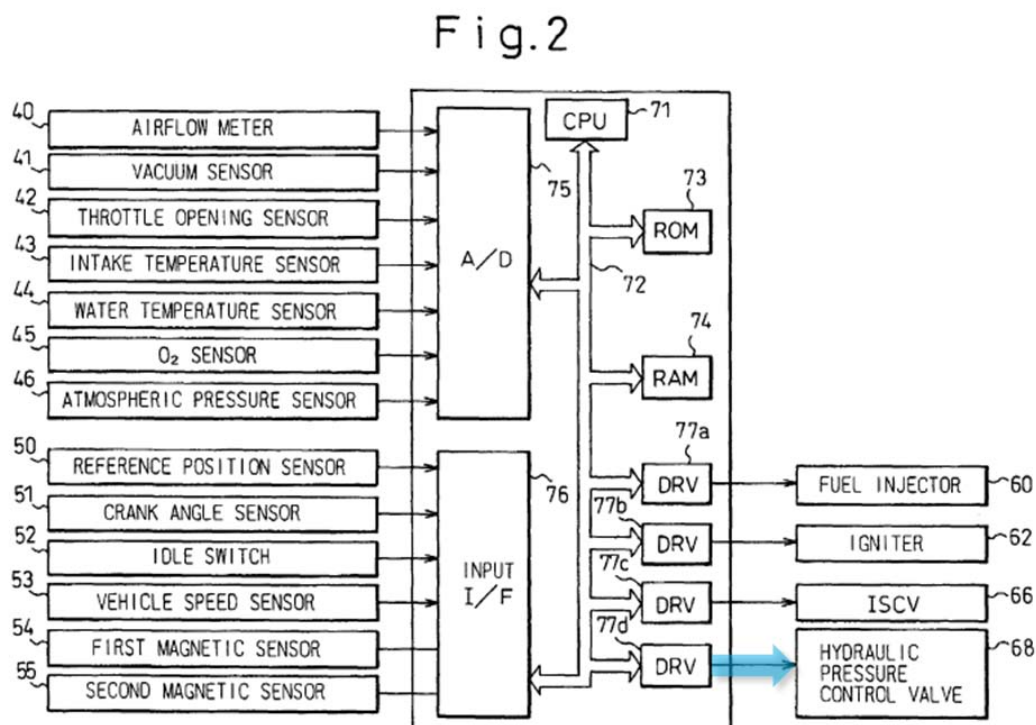
sensors 54 and 55, so that the cam shaft 85 of the intake valve 24 maintains a required rotational phase with respect to the crankshaft 81 (*i.e.*, “bring the actual displacement of the intake valve 24 to the target displacement IVTDt.”). EX1005, 6:6-36; 7:46-56; 8:5-7.

149. In other words, Yoshioka’s ECU 70 calculates an adjustment to the hydraulic control valve 68 (“a valve timing command”) based on the correction needed to bring the detected actual timing displacement for the open/close timing of the intake valve 24 to the target valve timing displacement IVTDt. This valve timing command provided to the hydraulic control valve 68 by the ECU 70 is shown below in Figures 1 and 2 (in blue).

Fig. 1



EX1005, FIG. 1 (annotated).



EX1005, FIG. 2 (annotated).

150. Thus, Yoshioka discloses [1.F].

C. Yoshioka and Ohkubo renders obvious claim 2.

151. Claim 2 depends from claim 1 and recites “said obtaining an engine performance command includes receiving a vehicle performance command from a driver of the vehicle and deriving the engine performance command based on the vehicle performance command.” Yoshioka renders obvious this limitation.

152. The ’540 patent does not purport to have invented obtaining an engine performance command by receiving a vehicle performance command from a driver of a vehicle. The ’540 patent instead admits that a vehicle performance command

is simply “received from the foot of a driver with the use of a conventional pedal” to obtain an engine performance command. EX1001, 2:4-8.

153. Yoshioka does not explicitly state that the opening and closing of the throttle valve 5 detected by Yoshioka’s throttle opening sensor 42 are based on a vehicle performance command from a driver of the vehicle. As discussed above, however, a POSA would have understood that a driver would have controlled the desired engine acceleration by opening and closing the throttle valve 5 detected by Yoshioka’s throttle opening sensor 42, *e.g.*, by depressing a conventional accelerator pedal. *See supra* Section X.B.2. For example, Toboldt discloses that:

The throttle valve is a device for varying the amount of air-fuel mixture that enters the intake manifold. This control is necessary so the speed of the vehicle can be changed. The throttle valve, Fig. 21-5, is simply a round disk mounted on a shaft, so that it can be tilted at various angles in the carburetor throttle valve body. It is connected by means of a suitable linkage to the accelerator pedal in the driving compartment of the vehicle. **Depressing the pedal opens the throttle valve, permitting an increased amount of air-fuel mixture to reach the manifold.**

EX1011, Toboldt, 239.

154. Bosch similarly described that in “conventional injection systems, the driver directly controls the opening of the throttle valve by depressing the accelerator pedal” and that the “driver uses the accelerator pedal to vary the throttle

valve's aperture so that the airflow into the engine is varied and with it the engine's power output." EX1012, 452, 460; *see also* 478 ("At full load, the accelerator pedal is depressed fully and the throttle is wide open (WOT)."), 498 ("In order to set the operating state desired by the driver (main function), the position of the gas pedal is converted by the microprocessor into a setpoint value for the engine torque."), 501 ("Systems with ETC (electronic throttle control, integrated in the engine control unit for ignition, injection and other auxiliary functions) detect the position of the accelerator pedal with a pedal-travel sensor (potentiometer). Taking into account the current engine operating status, the ECU calculates how far the throttle valve needs to open and actuated the throttle-valve drive accordingly (see Fig.). A throttle-valve angle sensor (potentiometer) monitors exact compliance with the desired throttle-valve position."), 507 ("The most important torque demand derives from the vehicle-operator command generated by the accelerator pedal, from the position of which the engine-management system interprets the demand to be made on the IC engine for a particular torque.").

155. Moreover, Probst discloses that: "Fuel delivery requirements depend more than anything else on how much work you are asking the engine to do—on how much of a 'load' you are placing on it. To accelerate, you step down harder on the accelerator. This opens the throttle valve, increasing manifold pressure. The greater pressure difference between the manifold and the cylinders increases air

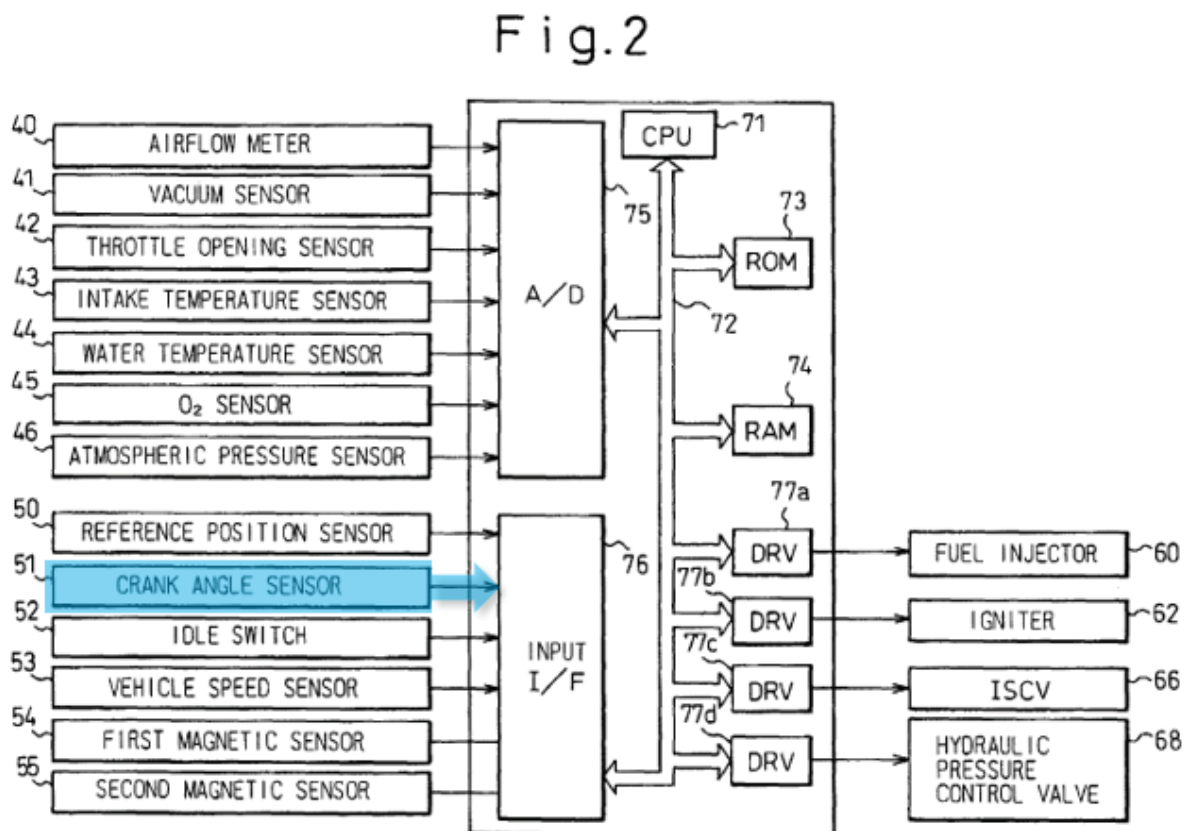
flow, and therefore fuel flow, to increase power and accelerate the car.” EX1016, 2:7. And even the ’540 patent acknowledges that the claimed “engine performance command” can be obtained from a “conventional pedal.” EX1001, 2:6-8.

156. Thus, the combination of Yoshioka and Ohkubo renders obvious claim 2.

D. Yoshioka and Ohkubo renders obvious claim 7.

157. Claim 7 depends from claim 1 and recites “said receiving an engine performance feedback includes receiving engine speed data.” Yoshioka renders obvious this limitation.

158. Yoshioka, in particular, discloses that the signal from the crank angle sensor 51 is used to calculate the engine revolution speed NE. EX1005, 7:16-24. Moreover, the ECU 70 of Yoshioka receives signals from all of the sensors and switches, including the crank angle sensor 51, to provide actuator control signals through drive control circuits 77a to 77d. *Id.*, 5:21-30. Figure 2 of Yoshioka, below, shows the ECU 70 receiving the signal from the crank angle sensor 51.



EX1005, FIG. 2 (annotated).

159. Yoshioka further discloses that the ECU 70 “uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.” EX1005, 9:22-26, 1:9-14 (“The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to **continuously and variably control the**

open/close timing of an intake or exhaust valve of the engine according to a target timing.”).

160. In my opinion, a POSA would have thus understood that the engine revolution speed NE is continuously monitored and fed back to the ECU 70. Moreover, it would have been obvious to a POSA to utilize the engine revolution speed NE already available to the ECU 70 of Yoshioka when determining the target valve timing displacement (“valve feedforward term”) to “continuously and variably” control the “open/close timing of the intake valve 24 according to operating conditions” and improve fuel efficiency. EX1005, 1:9-14, 6:6-8. Doing so would have merely been applying a known technique in a known device to yield predictable results.

161. Thus, Yoshioka renders obvious claim 7.

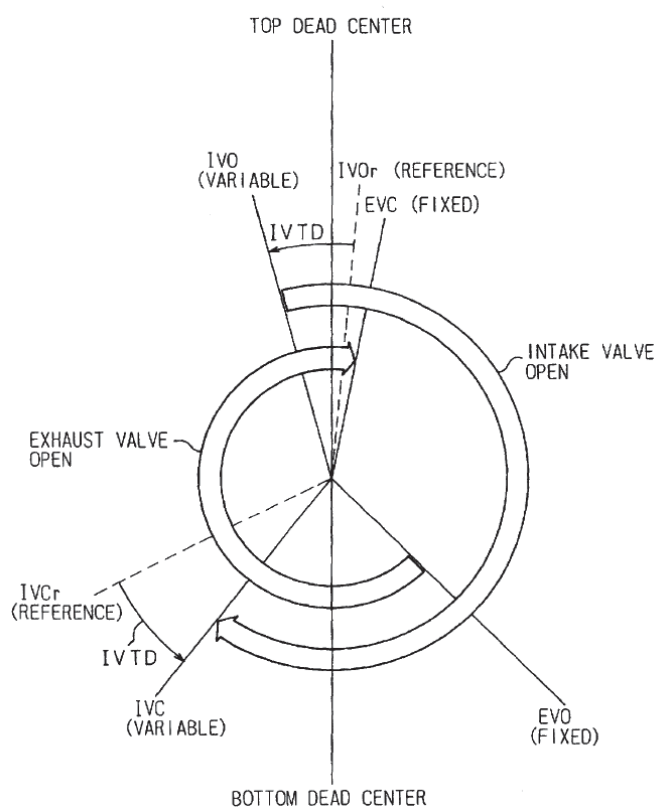
E. Yoshioka and Ohkubo renders obvious claim 10.

162. Claim 10 depends from claim 1 and recites “said calculating a valve timing command includes calculating an EVO command, an EVC/IVO command, and an IVC command.” Yoshioka, alone or in combination with Ohkubo, renders obvious this limitation.

163. Figure 3 of Yoshioka, below, illustrates a timing chart showing crank angles corresponding to the open/close timing of the intake and exhaust valves 24 and 26, including: the opening timing IVO and close timing IVC of intake valve 24

and the opening time EVO and close timing EVC of exhaust valve 26. EX1005, 6:16-37.

Fig. 3



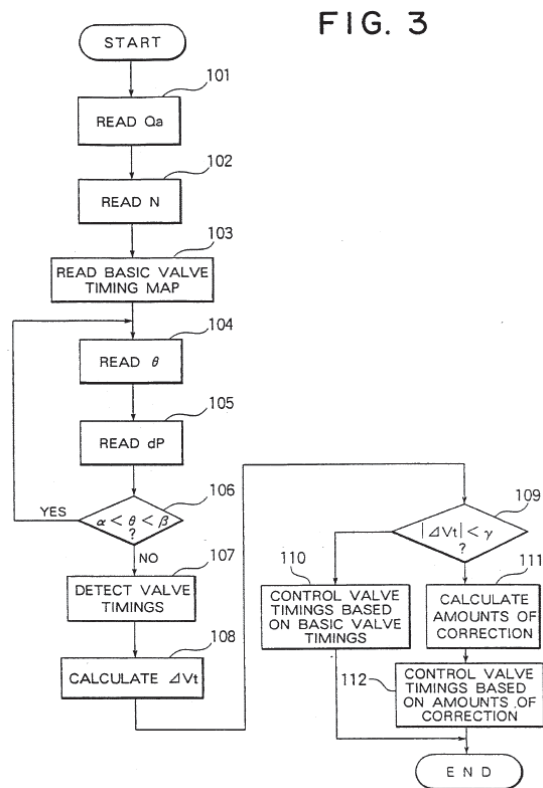
EX1005, FIG. 3.

164. Yoshioka discloses “continuously and variably controlling the open/close timing of at least one of the intake and the exhaust valves of the engine,” and in one example, that the “[v]alve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92.” *Id.*, 1:61-2:4, 6:6-8. Yoshioka further discloses that “[a]lthough the embodiments variably control the open/close

timing of the intake valve 24, the present invention is naturally applicable to control the exhaust valve 26, too.” EX1005, 9:35-37. That is, Yoshioka discloses varying the valve open/close timing for both the intake valve and exhaust valve—intake valve 24 opening timing IVO and close timing IVC and exhaust valve 26 opening timing EVO and close timing EVC.

165. In my opinion, a POSA would have thus understood that by controlling the continuously variable mechanism 92 for both the intake and exhaust valves 24 and 26, Yoshioka discloses calculating an EVO command, an EVC/IVO command, and an IVC command.

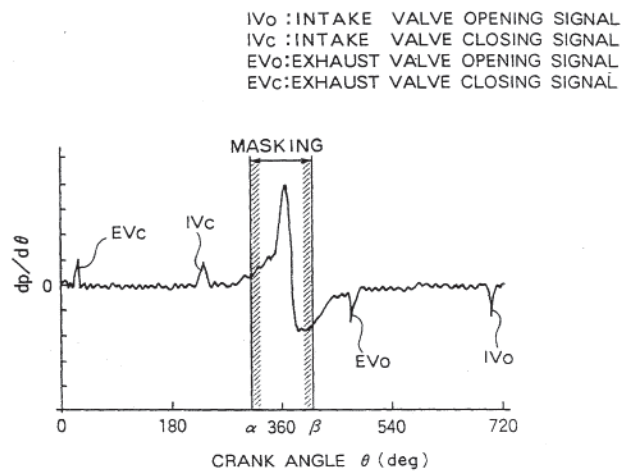
166. Should MMT argue, incorrectly, that Yoshioka does not disclose this limitation, Ohkubo also discloses calculating an EVO command, an EVC/IVO command, and an IVC command. Specifically, Ohkubo discloses first determining the values “for the basic valve timings such as IVo (intake valve opening timing), IVc (intake valve closing timing), EVo (exhaust valve opening timing) and EVc (exhaust valve closing timing)” from a two-dimensional table map which is stored in the NVM 29 (at step 103). EX1006, 5:15-22.



EX1006, FIG. 3

167. Ohkubo discloses that the EVO, EVC, IVO, and IVC valve timings are then detected (at step 107). EX1006, 5:38-39. This is shown below in Figure 4:

FIG. 4



EX1006, FIG. 4.

168. Then (at step 111), Ohkubo discloses calculating a correction for each of the EVO, EVC, IVO, and IVC valve timings, if needed. EX1006, 5:51-54 (“...the program goes to Step 111 where appropriate amounts of correction for properly correcting the valve timings are calculated.”). Based on this explicit teaching of Ohkubo, it is my opinion that it would have been obvious to a POSA for the ECU 70 of Yoshioka to calculate an EVO command, an EVC/IVO command, and an IVC command to “bring an actual displacement to the target displacement” as disclosed by Yoshioka. Doing so would have merely been applying a known technique in a similar, known device in the same way. *See supra* Section X.A.

169. Thus, Yoshioka, alone or in combination with Ohkubo, renders obvious claim 10.

F. Yoshioka and Ohkubo renders obvious claim 11.

170. Claim 11 depends from claim 1 and recites “further comprising receiving fuel conversion data.” Yoshioka renders obvious this limitation.

171. For example, Yoshioka discloses that the ECU 70 receives “an air-fuel ratio feedback correction according to a signal from the O₂ sensor 45” EX1005, 5:48-55. That is, the “exhaust valve 26 discharges an exhaust gas of the combusted air-fuel mixture into an exhaust manifold 30 connected to an exhaust pipe 34” such that the O₂ sensor 45 detects the concentration of oxygen in the exhaust gas. EX1005, 4:48-52. Accordingly, Yoshioka discloses receiving fuel conversion data (*e.g.*, the concentration of oxygen in the exhaust gas) from the O₂ sensor 45.

172. Moreover, the use of oxygen (O₂) sensors—also referred to as lambda sensors—was known in the art as described by Aird:

The ultimate sophistication is to provide an oxygen sensor (O₂ sensor, lambda sensor) in the exhaust pipe and to correct the mixture strength from moment to moment, based on that information; an excess of O₂ calls for a richer mixture, and vice versa. For all these reasons, EFI with an injector for each cylinder is unarguably the state-of-the art for fuel-air mixing and delivery.

EX1009, Aird, 33. *See also*, EX1016, 2:19 (“The source of this feedback is a lambda sensor (also called an exhaust-gas sensor, or oxygen sensor), installed in the exhaust system. The sensor generates a low-voltage signal; the signal's strength is based on the amount of unused oxygen remaining in the exhaust stream—an indirect measurement of the air-fuel ratio”).

173. Thus, the combination of Yoshioka and Ohkubo renders obvious claim 11.

G. Yoshioka and Ohkubo renders obvious claim 12.

174. Claim 12 depends from claim 11 and recites “said determining a valve feedforward term includes determining a valve feedforward term based on the fuel conversion data.” Yoshioka in combination with Ohkubo renders obvious this limitation.

175. As I discussed above, Yoshioka’s ECU 70 receives an input from the O₂ sensor 45 indicating fuel conversion data. *See supra* Section X.F. Yoshioka further discloses that the ECU 70 receives the signals from the sensors, including the O₂ sensor 45, and controls the open/close timing of the intake valve 24 according to various operating conditions by controlling the continuously variable mechanism 92. EX1005, 5:21-24; 6:6-15 (“Valve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92. ...”).

176. In my opinion, a POSA would have understood that the fuel conversion data (*e.g.*, the concentration of oxygen in the exhaust gas) from the O₂ sensor 45 is an operating condition of the engine. Thus, a POSA would have at least found it obvious for Yoshioka's ECU 70 to use the signal input from the O₂ sensor 45, an operating condition of the engine, to determine the initial valve timing target displacement IVTDt ("valve feedforward term").

177. Ohkubo similarly discloses the engine having an oxygen sensor 24 to sense the concentration of oxygen contained in the exhaust gas discharged from the engine cylinder 1 and provides that as an input to control unit 25. EX1006, 4:30-42. The control unit 25 then uses the input from the sensors, including the oxygen sensor 24, to determine optimal opening and closing timings of the intake and exhaust valves. *Id.*, 4:46-65.

178. Indeed, the use of oxygen (O₂) sensors to detect fuel conversion data was known in the art as described by Aird:

The ultimate sophistication is to provide an oxygen sensor (O₂ sensor, lambda sensor) in the exhaust pipe and to correct the mixture strength from moment to moment, based on that information; an excess of O₂ calls for a richer mixture, and vice versa. For all these reasons, EFI with an injector for each cylinder is unarguably the state-of-the art for fuel-air mixing and delivery.

EX1009, 33. *See also*, EX1016, 2:19 ("The source of this feedback is a lambda sensor (also called an exhaust-gas sensor, or oxygen sensor), installed in the

exhaust system. The sensor generates a low-voltage signal; the signal's strength is based on the amount of unused oxygen remaining in the exhaust stream—an indirect measurement of the air-fuel ratio”).

179. Thus, it would have been obvious to a POSA to utilize the fuel conversion data already available to Yoshioka’s ECU 70 to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) as taught by Ohkubo to improve fuel efficiency and reduce harmful exhaust components. Doing so would have merely been applying a known technique in a similar, known device in the same way. *See supra* Section X.F.

180. Thus, the combination of Yoshioka and Ohkubo renders obvious claim 12.

H. Yoshioka and Ohkubo renders obvious claim 13.

181. Claim 13 depends from claim 11 and recites “said calculating a valve feedback term includes calculating a valve feedback term based on the fuel conversion data.” Yoshioka, alone or in combination with Ohkubo, renders obvious this limitation.

182. As I described above, Yoshioka’s ECU 70 receives an input from the O₂ sensor 45 indicating fuel conversion data, and it would have been obvious to a POSA to utilize the fuel conversion data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve

feedforward term”) as taught by Ohkubo to improve fuel efficiency and reduce harmful exhaust components. *See supra* Section X.G.

183. Yoshioka further discloses that the ECU 70 “uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.” EX1005, 9:22-26, 1:9-14 (“The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of an intake or exhaust valve of the engine according to a target timing.”).

184. In my opinion, a POSA would have thus understood that the fuel conversion data from the O₂ sensor 45 is continuously monitored and fed back to the ECU 70 to “continuously and variably” control the “open/close timing of the intake valve 24 according to operating conditions.” EX1005, 1:9-14, 6:6-8. It would have also been obvious to a POSA to use the most current fuel conversion data from the O₂ sensor 45 along with the measured actual valve timing displacement to determine the “valve feedback term.” Indeed, by considering the input from the O₂ sensor 45 when calculating both the “valve feedforward term” and the “valve feedback term,” it would have been obvious to a POSA that the

valve open/close timings can be more accurately controlled to improve fuel efficiency and reduce harmful exhaust components. Doing so would have merely been applying a known technique in a similar, known device in the same way.

185. Thus, Yoshioka, alone or in combination with Ohkubo, renders obvious claim 13.

I. Yoshioka and Ohkubo renders obvious claim 14.

186. Claim 14 depends from claim 1 and recites “further comprising receiving engine emissions data.” Yoshioka discloses this limitation.

187. Yoshioka discloses that the ECU 70 receives “an air-fuel ratio feedback correction according to a signal from the O₂ sensor 45, and an air-fuel ratio learning correction to equalize a central feedback correction value to a theoretical air-fuel ratio.” EX1005, 5:48-55. That is, the “exhaust valve 26 discharges an exhaust gas of the combusted air-fuel mixture into an exhaust manifold 30 connected to an exhaust pipe 34” such that the O₂ sensor 45 detects the concentration of oxygen in the exhaust gas. *Id.*, 4:48-52. The quantity of oxygen present in the exhaust gas, as measured by the O₂ sensor 45 and received by the ECU 70 of Yoshioka, represents the level of emissions in the exhaust gas (“engine emissions data”).

188. Thus, the combination of Yoshioka and Ohkubo renders obvious claim 14.

J. Yoshioka and Ohkubo renders obvious claim 15.

189. Claim 15 depends from claim 14 and recites “said determining a valve feedforward term includes determining a valve feedforward term based on the engine emissions data.” Yoshioka, alone or in combination with Ohkubo, renders obvious this limitation.

190. As I discussed above, Yoshioka’s ECU 70 receives an input from the O₂ sensor 45 indicating fuel conversion data. *See supra* Section X.F. Yoshioka further discloses that the ECU 70 receives the signals from the sensors, including the O₂ sensor 45, and controls the open/close timing of the intake valve 24 according to various operating conditions by controlling the continuously variable mechanism 92. EX1005, 5:21-24; 6:6-15 (“Valve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92. ...”).

191. In my opinion, a POSA would have understood that the engine emissions data (*e.g.*, the concentration of oxygen in the exhaust gas) from the O₂ sensor 45 is an operating condition of the engine. *See* EX1016, 2:19 (“The source of this feedback is a lambda sensor (also called an exhaust-gas sensor, or oxygen sensor), installed in the exhaust system. The sensor generates a low-voltage signal; the signal's strength is based on the amount of unused oxygen remaining in the exhaust stream—an indirect measurement of the air-fuel ratio”). It would have thus

been obvious to a POSA for the ECU 70 to use the signal input from the O₂ sensor 45, an operating condition of the engine, to determine the initial valve timing target displacement IVTDt (“valve feedforward term”). Doing so would have merely been applying a known technique in a known device to yield predictable results.

192. Ohkubo similarly discloses that the engine has the oxygen sensor 24 to sense the concentration of oxygen contained in the exhaust gas discharged from the engine cylinder 1 and provide that as an input to control unit 25. EX1006, 4:30-42. The control unit 25 then uses the input from the sensors, including the oxygen sensor 24, to determine optimal opening and closing timings of the intake and exhaust valves. *Id.*, 4:46-65.

193. It is my opinion that it would have been obvious to a POSA to utilize the engine emissions data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) as taught by Ohkubo to improve fuel efficiency and reduce harmful exhaust components. Doing so would have merely been applying a known technique in a known device to yield predictable results. *See supra* Section X.G.

194. Thus, Yoshioka, alone or in combination with Ohkubo, renders obvious claim 15.

K. Yoshioka and Ohkubo renders obvious claim 16.

195. Claim 16 depends from claim 11 and recites “said calculating a valve feedback term includes calculating a valve feedback term based on the engine emissions data.” Yoshioka, alone or in combination with Ohkubo, renders obvious this limitation.

196. As I discussed above, Yoshioka’s ECU 70 receives an input from the O₂ sensor 45 indicating engine emissions data, and it would have been obvious to a POSA to utilize the engine emissions data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) as taught by Ohkubo to improve fuel efficiency and reduce harmful exhaust components. *See supra* Section X.G.

197. Yoshioka further discloses that the ECU 70 “uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.” EX1005, 9:22-27, 1:9-14 (“The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of an intake or exhaust valve of the engine according to a target timing.”).

198. In my opinion, a POSA would have thus understood that the engine emissions data from the O₂ sensor 45 is thus continuously monitored and fed back to the ECU 70 to “continuously and variably” control the “open/close timing of the intake valve 24 according to operating conditions.” EX1005, 1:9-14, 6:6-8.

Moreover, it would have been obvious to a POSA to use the most current engine emissions data from the O₂ sensor 45, along with the measured actual valve timing displacement, to determine the “valve feedback term.” Indeed, by considering the input from the O₂ sensor 45 when calculating both the “valve feedforward term” and the “valve feedback term,” it would have been obvious to a POSA that the valve open/close timings can be more accurately controlled to improve fuel efficiency and reduce harmful exhaust components. Doing so would have merely been applying a known technique in a known device to yield predictable results.

See supra Section V.A.

199. Thus, Yoshioka, alone or in combination with Ohkubo, renders obvious claim 16.

XI. Ground 2: The combination of Yoshioka, Ohkubo, and Reed renders obvious claims 3-6, 8, and 9.

A. Rationale for combining Yoshioka, Ohkubo, and Reed

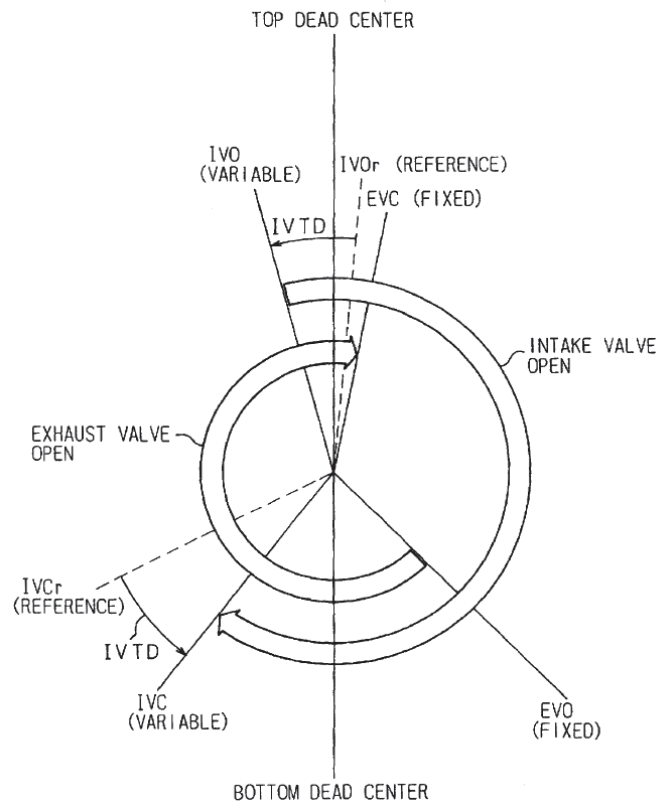
200. In my opinion, a POSA would have been motivated to combine the teachings of Yoshioka, Ohkubo, and Reed for multiple reasons, rendering the

claims obvious. I discussed the combination of Yoshioka and Ohkubo above in Section X.A. *See supra* Section X.A.

201. First, Yoshioka, Ohkubo, and Reed are from the same field of endeavor—control systems for valve timing for an internal combustion engine. Yoshioka, Ohkubo, and Reed, in particular, are directed to analogous valve timing control systems. *See supra* Section IX. Like Yoshioka and Ohkubo, Reed discloses “...a method of increasing engine torque and/or of reducing engine emissions by varying the phasing of either the intake valves or the exhaust valves or both.” EX1007, 1:11-14.

202. Reed cites to Yoshioka in the Background of the Invention section, stating that Yoshioka “purports to optimize the valve timing so as to improve the output of the engine at high altitudes while it is operating under high loads. It also purports to reduce the fuel consumption and emissions of the engine at high altitudes as it operates under low to medium loads.” EX1007, 6:54-61. However, Reed incorrectly concludes that Yoshioka is limited to controlling valve timing of only the intake valve. *Id.*, 6:61-65. For example, Yoshioka illustrates a timing chart showing crank angles corresponding to the open/close timing of the intake **and** exhaust valves 24 and 26, including: intake valve 24 opening timing IVO and close timing IVC, and exhaust valve 26 opening timing EVO and close timing EVC. EX1005, 6:16-37.

Fig. 3



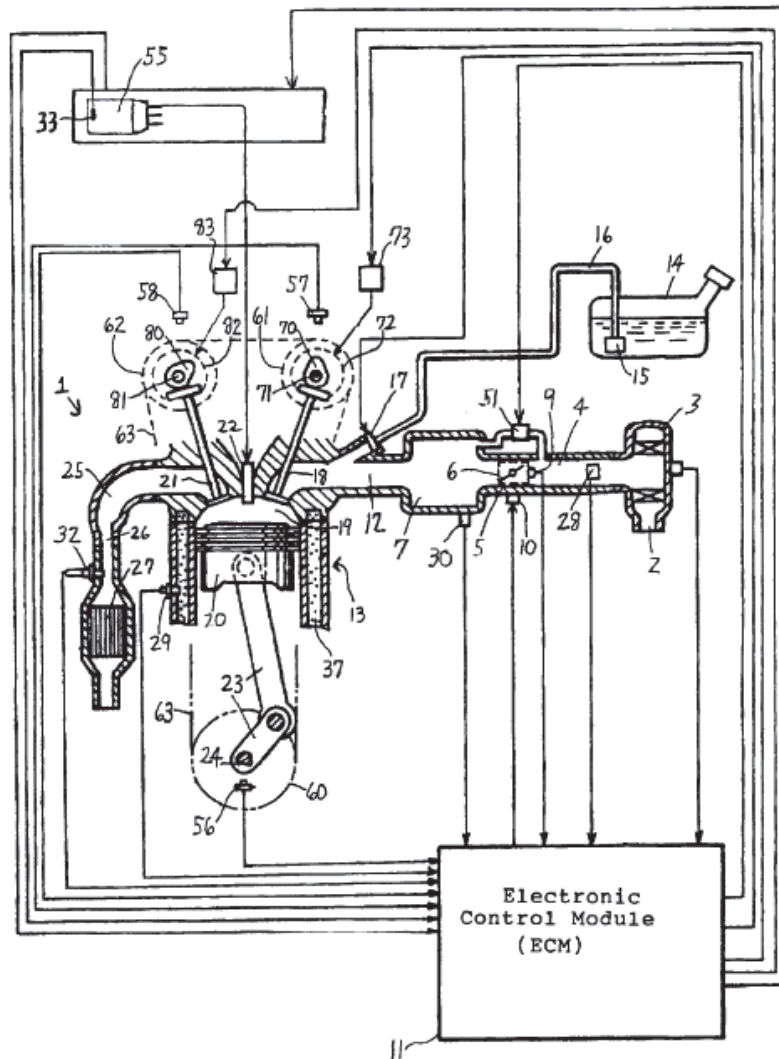
EX1005, FIG. 3.

203. Moreover, Yoshioka discloses “continuously and variably controlling the open/close timing of at least one of the intake **and** the exhaust valves of the engine,” and in one example, that the “[v]alve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92.” EX1005, 1:61-2:4, 6:6-8. Yoshioka further discloses that “[a]lthough the embodiments variably control the open/close timing of the intake valve 24, the present invention is naturally

applicable to control the exhaust valve 26 too.” EX1005, 9:35-37. *See also supra* Section X.E.

204. Reed further discloses an engine and electronic control module (ECM) 11 having the same components as Yoshioka, including: ECM 11, intake valve 18, exhaust valve 21, air flow sensor 36, and O₂ (oxygen) sensor 32. EX1007, 1:24-29, 1:64-2:10, 2:25-51.

FIGURE 1



EX1007, FIG. 1.

205. Reed discloses a method of setting target valve timing similar to that of Yoshioka, including measuring aspects of engine performance (*e.g.*, engine revolution speed), measuring environmental conditions (*e.g.*, pressure and temperature), and determining target valve timing. EX1005, 7:16-8:8; EX1007,

12:45-47. Reed further discloses controlling valve timing based on temperature, which would have been readily applicable to Yoshioka's ECU 70 controlling valve timing. Indeed, Reed discloses that the desired valve timing should be corrected for both pressure and temperature. EX1007, 12:45-47, 13:44-46.

206. While Yoshioka is silent on controlling valve timing based on temperature, it would have been obvious to a POSA to combine the teachings of Yoshioka (and Ohkubo) and Reed, which would have yielded predictable benefits and results—for example, improved valve timing control by accounting for pressure and temperature. Indeed, Reed recognized that it is **preferable** to adjust the valve timing to accommodate for the effects of both temperature and pressure to “improve[] the ability of the engine to deliver the desired torque, produce less feed gas emissions, idle more stably, and consume less fuel.” EX1007, Abstract, 12:45-47, 13:44-46.

207. In my opinion, a POSA would have had a reasonable expectation of success combining Reed with Yoshioka and Ohkubo. While Reed discloses using engine temperature to correct valve timing, a POSA would have understood that ambient temperature, like ambient pressure, also impacts engine performance.

208. For example, Jackson recognized in the 1970s that ambient temperature impacts engine performance. See EX1014, Jackson, 1:9-29 (“... internal combustion engines have several characteristics in common regarding the

effect and control of factors which influence the efficiency of fuel conversion to useable heat and/or mechanical energy and attendant effects such as noxious exhaust component emissions” that “are expected to operate and are operated, under varying conditions of load, engine operating temperature, **ambient temperature and barometric pressure**,” and “... all of these variables are known to effect the efficien[c]y of fuel conversion to mechanical energy and the attendant production of undesirable byproducts or unreacted feed constituents in the form of exhaust emissions.”); 1:30-38 (“For example, in the operation of a conventional spark ignition gasoline automotive engine, it is known that considerable reduction in air-to-fuel ratio, below that required for operation at design operating temperatures, is required when the engine is cold. This requirement is due to the relatively low volatility of the gasoline fuel at relatively low engine operating temperatures thus requiring the injection of higher amounts of fuel per unit standard volume of air to obtain adequate performance.”).

209. In my opinion, a POSA would have thus recognized that Reed’s technique of correcting valve timing based on temperature as well as pressure, just like Yoshioka, was a predictable solution readily applicable to Yoshioka’s ECU 70.

210. Moreover, a POSA would have also understood how to combine the teachings of Reed with Yoshioka and Ohkubo to control the valve timing. A POSA

would have simply used Reed's implementation details (for example, adjusting valve timing using a temperature-correction (in addition to pressure-correction) and calculating an appropriate amount of correction for the valve timing(s)) in order to accurately control Yoshioka's valve timings. And it would have been obvious to a POSA to utilize the ambient temperature data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt ("valve feedforward term") by applying a temperature correction coefficient as taught by Reed to improve fuel efficiency and reduce engine emissions. EX1007, 1:11-14 ("...reducing engine emissions by varying the phasing of either the intake valves or the exhaust valves or both.").

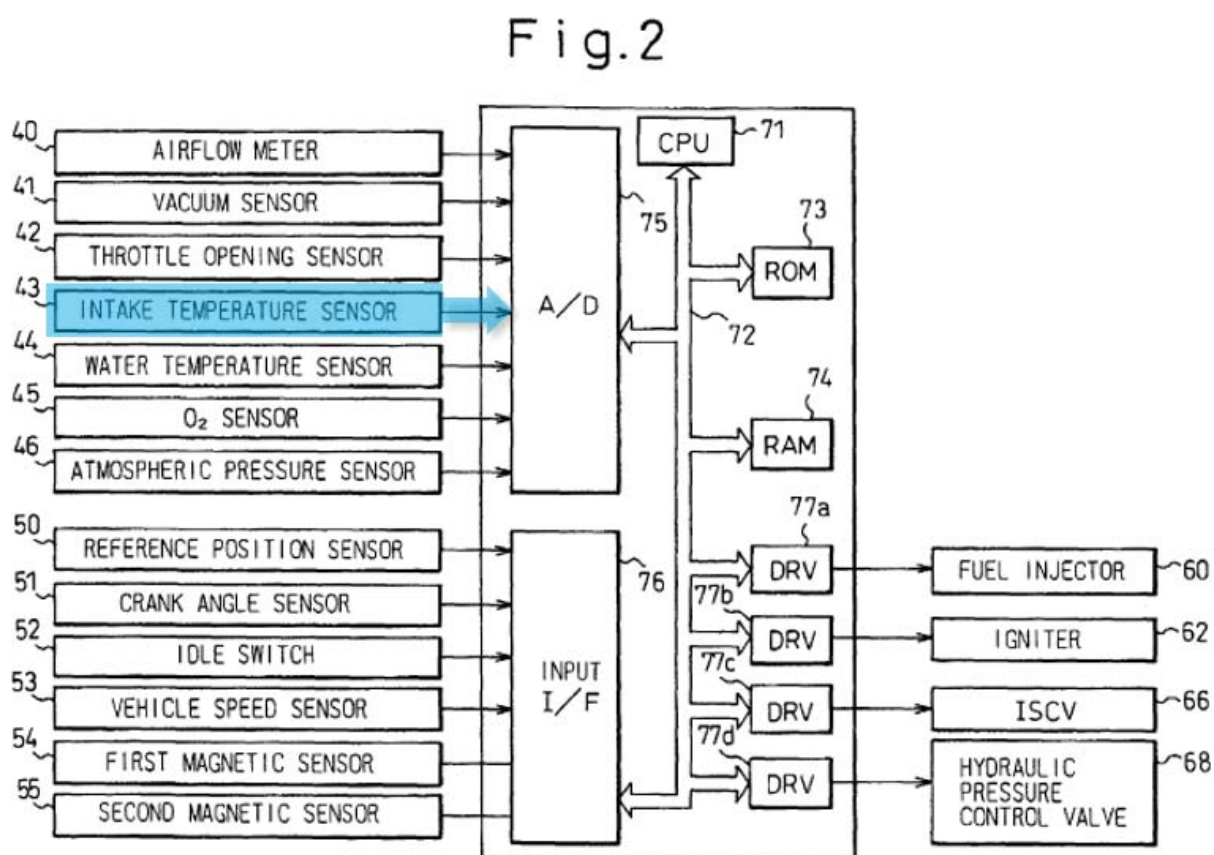
211. Furthermore, a POSA would have understood that Reed's valve timing correction technique would have been combined in a known way to provide the predictable benefits disclosed in Yoshioka to improve the accuracy of controlling the valve opening and closing timings to "optimiz[e] valve timing to maintain the output of the engine." EX1005, 2:32-34; EX1006, 6:13-22 ("the air-intake efficiency and the exhaust-discharge efficiency of the engine can be improved, thus increasing the output power of the engine."). Doing so would have merely been applying a known technique in a known device to yield predictable results.

212. Accordingly, it is my opinion that it would have been obvious to combine the teachings of Yoshioka, Ohkubo, and Reed.

B. Yoshioka, Ohkubo, and Reed renders obvious claim 3.

213. Claim 3 depends from claim 1 and recites “said receiving an environmental conditions signal includes receiving ambient temperature data.” Yoshioka, alone or in combination with Ohkubo and Reed, renders obvious this limitation.

214. Yoshioka discloses that a “temperature sensor 43 detects the temperature of intake air” (*i.e.*, ambient air temperature). EX1005, 4:15-16, FIG. 2. As shown in Figure 2, below, the ECU 70 receives the signal from the temperature sensor 43, along with the “signals from the sensors and switches through an A/D converter 75 and an interface 76, processes the signals, and provides actuator control signals through drive control circuits 77a to 77d.” *Id.*, 5:26-31. Accordingly, the signal received by the ECU 70 includes ambient temperature data detected by sensor 43. *Id.*, 4:15-16, 5:26-31.



EX1005, FIG. 2 (annotated).

215. Thus, Yoshioka discloses receiving ambient temperature data.

216. Should MMT argue that claim 3 requires using the received ambient temperature data to determine the “valve feedforward term,” it is my opinion that it would have been obvious to a POSA to use temperature data to determine the target valve timing displacement IVTDd (“valve feedforward term”) as taught by Reed.

217. As an initial matter, a POSA would have known that atmospheric pressure and temperature have a notable impact on the operation of the engine. The

number of air molecules present in an engine during combustion, for example, varies according to air density, which depends on these two environmental factors (*i.e.*, atmospheric pressure and temperature). Specifically, air density is directly related to temperature and pressure per the ideal gas law: $pV=nRT$. Here, p is the pressure, V is the volume, n is the number of moles of gas molecules, T is the temperature, and R is a constant. EX1012, 59. Thus, as the pressure increases at a constant temperature, the number of molecules n available to the engine will increase, and if the temperature increases at a constant pressure, the number of molecules n available to the engine will decrease. Both of these changes in environmental conditions impact engine performance.

218. Reed discloses that the intake valve timing is varied based on pressure and temperature using pressure and temperature-correction values. EX1007, 12:45-13:20 (“FIG. 7 illustrates an example of a temperature-correction curve T_c . It is plotted against engine temperature on the horizontal axis and against the correction factor on the vertical axis. It increases monotonically, preferably linearly, until temperature T_M is reached, at which point the curve T_c has its maximum value, unity. When the temperature of the engine 1 [is] less than or equal to temperature T_1 , intake cam phasing is not desired. When the engine temperature lies above T_1 , temperature-correction is required. The desired intake cam phaser position is then made equal to the value obtained from the table lookup multiplied by the

temperature-correction value taken from curve T_c”), 15:28-41. That is, Reed applies not only a pressure correction value (P_c) when determining valve timing, like Yoshioka’s pressure correction coefficient K_{pa} , but also applies a temperature correction value (T_c) to “improve[] the ability of the engine to deliver the desired torque, produce less feed gas emissions, idle more stably, and consume less fuel.” EX1007, Abstract.

219. While Reed discloses using engine temperature to correct valve timing, a POSA would have understood that ambient temperature, like ambient pressure, also impacts engine performance. For example, Jackson recognized in the 1970s that ambient temperature impacts engine performance. See EX1014, 1:9-29 (“... internal combustion engines have several characteristics in common regarding the effect and control of factors which influence the efficiency of fuel conversion to useable heat and/or mechanical energy and attendant effects such as noxious exhaust component emissions” that “are expected to operate and are operated, under varying conditions of load, engine operating temperature, ambient temperature and barometric pressure,” and “all of these variables are known to effect the efficien[c]y of fuel conversion to mechanical energy and the attendant production of undesirable byproducts or unreacted feed constituents in the form of exhaust emissions.”), 1:30-38 (“For example, in the operation of a conventional spark ignition gasoline automotive engine, it is known that considerable reduction

in air-to-fuel ratio, below that required for operation at design operating temperatures, is required when the engine is cold. This requirement is due to the relatively low volatility of the gasoline fuel at relatively low engine operating temperatures thus requiring the injection of higher amounts of fuel per unit standard volume of air to obtain adequate performance.”).

220. In my opinion, it would have thus been obvious to a POSA to utilize the ambient temperature data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) by applying a temperature correction coefficient as taught by Reed to improve fuel efficiency and reduce engine emissions. EX1007, 1:11-14 (“...reducing engine emissions by varying the phasing of either the intake valves or the exhaust valves or both.”). Doing so would have merely been applying a known technique—adjusting valve timing using a temperature correction coefficient—in a known device to yield predictable results—improved engine performance. *See supra* Section XI.A.

221. Thus, the combination of Yoshioka, Ohkubo, and Reed renders obvious claim 3.

C. Yoshioka, Ohkubo, and Reed renders obvious claim 4.

222. Claim 4 depends from claim 3 and recites “said determining a valve feedforward term includes determining the valve feedforward term based on the

ambient temperature data.” Yoshioka, in combination with Ohkubo and Reed, renders obvious this limitation.

223. As I discussed above, Yoshioka discloses receiving ambient temperature data, and Reed discloses determining valve timing based on temperature data. *See* EX1005, 4:15-16, 5:26-32; EX1007, 12:45-13:20, 15:28-41; *see also supra* Section XI.B. Moreover, it is my opinion that it would have been obvious to a POSA to utilize the ambient temperature data already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) as taught by Reed to improve fuel efficiency and reduce engine emissions, as described above. EX1007, 1:11-14 (“...reducing engine emissions by varying the phasing of either the intake valves or the exhaust valves or both.”). Doing so would have merely been applying a known technique—adjusting valve timing using a temperature correction coefficient—in a known device to yield predictable results—improved engine performance. *See supra* Section XI.A.

224. Thus, the combination of Yoshioka, Ohkubo, and Reed discloses claim 4.

D. Yoshioka, Ohkubo, and Reed renders obvious claim 5.

225. Claim 5 depends from claim 4 and recites “said determining a valve feedforward term includes determining the valve feedforward term based on the

engine performance feedback.” Yoshioka, in combination with Ohkubo and Reed, renders obvious this limitation.

226. Yoshioka does not explicitly state that the target valve timing displacement, IVTDt, is set using the actual valve timing displacement. However, Yoshioka discloses that the ECU 70 “uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.” EX1005, 9:22-27, 1:9-14 (“The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of an intake or exhaust valve of the engine according to a target timing.”). That is, Yoshioka’s ECU 70 continuously receives the actual valve timing displacement (“engine performance feedback”) and sets the target valve timing displacement IVTDt (“valve feedforward term”).

227. It is my opinion that a POSA would have understood that the actual valve timing displacement (“engine performance feedback”) is an operating condition of the engine. It would have thus been obvious to a POSA to use this feedback already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) to further

improve fuel efficiency and optimize valve timing. EX1005, 9:40-43 (“... the present invention provides a method of and apparatus for setting optimum valve timing according to atmospheric pressure for a continuous variable valve timing mechanism of an internal combustion engine.”). Doing so would have merely been applying a known technique in a similar, known device in the same way.

228. And as I discussed above, it would have been obvious to a POSA to utilize the engine revolution speed NE (an “engine performance feedback”) already available to the ECU 70 of Yoshioka to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) to improve fuel efficiency. Doing so would have merely been applying a known technique in a similar, known device in the same way. *See supra* Section X.D.

229. Thus, the combination of Yoshioka, Ohkubo, and Reed renders obvious claim 5.

E. Yoshioka, Ohkubo, and Reed renders obvious claim 6.

230. Claim 6 depends from claim 5 and recites “said determining a valve feedforward term includes referencing a look-up table with the engine performance command, the ambient temperature, and the engine performance feedback.”

Yoshioka, in combination with Ohkubo and Reed, renders obvious this limitation.

231. As an initial matter, it was known in the art to use look-up tables/maps in engine control systems to account for various operating parameters and variables. Probst, for example, described that:

To determine precise timing-advance requirements, each family of engines is tested to learn the best timing for each condition of speed, load, and other variables in heat and cold, on the dynamometer and in the mountains. The goal is to find the timing for best power, for best economy, all the while meeting emission limits.

The result of these tests is a series of data “maps”, as shown in Fig. 4-4. Literally thousands of data points from these tests are stored in the computer memory of the Motronic control unit ROM (Read Only Memory) for readout during engine operation. As you may know, a ROM cannot be changed. For any combination of engine load and rpm, the control unit can supply the best ignition timing. For example, for an rpm-input signal of 2000 rpm, at a load signaled by the air-flow sensor, the computer would look up the timing advance angle, let's say it should be 22 degrees BTDC.

EX1016, 3:29.

232. Moreover, Probst disclosed that:

The main map in the control unit is a set of lambda memory points used to determine the desired air-fuel ratio for the fuel injection pulses. The lambda memory map, shown in Fig. 4-8 is derived from lab and road tests, and modified for the requirements of the vehicle and the country where it will operate:

Air quantity and the requirements for control of injected fuel can vary due to many factors such as manifold-pressure variations resulting from individual piston pumping, and intake-valve opening/closing.

EX1016, 3:31.

233. Figure 4-8 of Probst illustrates one such map:

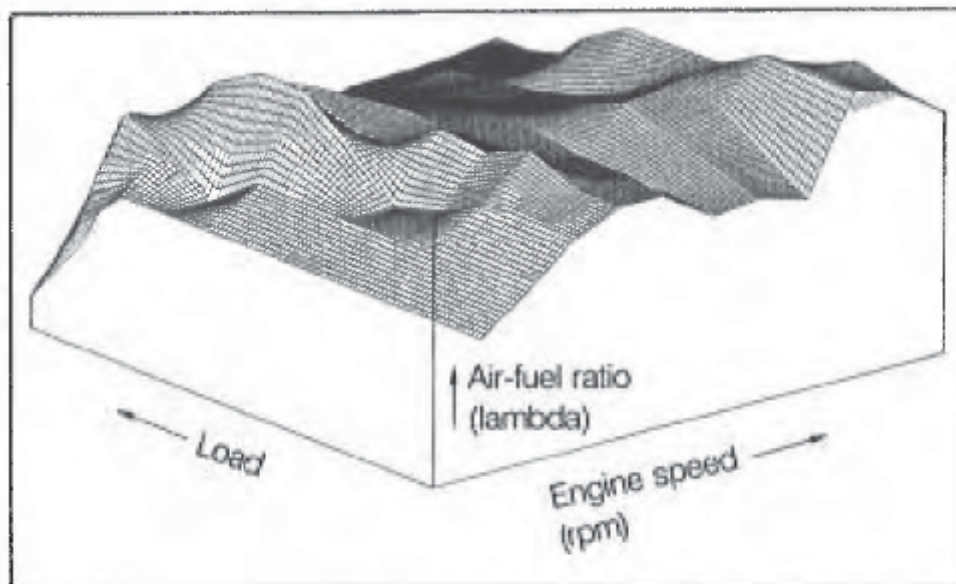


Fig. 4-8. Lambda "map" establishes lambda, or air-fuel ratio for each load/speed point.

EX1016, FIG. 4-8.

234. Yoshioka, in particular, discloses that the ECU 70 "uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high [loads], thereby smoothing the torque characteristics with respect to the engine load and the

throttle opening.” EX1005, 5:25-26 (“A read-only memory (ROM) 73 stores programs and maps.”), 7:36-40, 9:22-27.

235. Ohkubo similarly discloses a basic valve timing map 7 used to determine valve opening and closing times. EX1006, 3:33-40, 4:63-65 (“In this regard, predetermined optimal valve opening and closing timings are stored in the NVM 29 in an appropriate form such as a map.”), 5:15-22.

236. And Reed also discloses the use of lookup tables to determine desired valve timing based on at least two variables such as engine speed and engine load. EX1007, 12:27-44.

237. As I discussed above, it is my opinion that it would have been obvious to a POSA to use the feedback (*e.g.*, actual valve timing or engine revolution speed NE) already available to Yoshioka’s ECU 70 to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) to further improve fuel efficiency and optimize valve timing. *See supra* Section XI.D. Moreover, it would have been obvious to a POSA to use the existing map/lookup table to determine the initial valve timing target displacement IVTDt (“valve feedforward term”) with the ambient temperature measurement and engine performance feedback (*e.g.*, actual valve timing or engine revolution speed NE). Doing so would have simply required a modification of the pre-existing map/lookup table to account for additional feedback when determining the initial valve timing target displacement

IVTDt, which would have been well within the skill of a POSA at the time. Indeed, this would have simply been the use of a known technique (map/lookup tables) in a similar device (engine control unit) in the same way.

238. Thus, the combination of Yoshioka, Ohkubo, and Reed renders obvious claim 6.

F. Yoshioka, Ohkubo, and Reed renders obvious claim 8.

239. Claim 8 depends from claim 1 and recites “said obtaining an engine performance command includes obtaining an engine torque command; and wherein receiving an engine performance feedback includes receiving engine torque data.” Yoshioka, in combination with Ohkubo and Reed, renders obvious this limitation.

240. Yoshioka discloses that the ECU 70 calculates the engine load GN based on the mass flow rate of the intake air detected by the airflow meter 40 and the engine revolution speed NE and ultimately determines the target valve timing displacement, IVTDt, using these values. *See supra* Section X.B. Reed similarly discloses calculating engine load and setting the desired valve timing accordingly. EX1007, Abstract, 9:51-53, 10:35-41, 12:27-44, 15:62-16:5. In particular, Reed discloses that the engine load is derived from engine torque. EX1007, 14:19-21.

241. Based on the teaching of Reed, it is my opinion that it would have been obvious to a POSA for Yoshioka’s ECU 70 to calculate the engine load GN based on desired engine torque. Doing so would have simply been the use of a

known technique (determining engine load based on desired engine torque) in a similar device (engine control unit) in the same way. Yoshioka, in combination with Ohkubo and Reed, thus discloses “obtaining an engine performance command includes obtaining an engine torque command.”

242. Moreover, it would have been obvious to a POSA, and in fact a natural and logical extension, to monitor the engine torque data as feedback for continuous control of the open/close timing of the valve(s) to “thereby smooth[] the torque characteristics with respect to the engine load and the throttle opening” as described by Yoshioka. EX1005, 9:22-26. Doing so would have merely been applying a known technique in a similar, known device in the same way. *See supra* Section X.B.

243. Thus, the combination of Yoshioka, Ohkubo, and Reed renders obvious claim 8.

G. Yoshioka, Ohkubo, and Reed renders obvious claim 9.

244. Claim 9 depends from claim 8 and recites “calculating a valve feedback term includes calculating a valve feedback term based on the engine torque command and the engine torque data.” Yoshioka, in combination with Ohkubo and Reed, renders obvious this limitation.

245. As I discussed above, it would have been obvious to a POSA to obtain an engine torque command (“engine performance command”) and engine torque data (“engine performance feedback”). *See supra* Section XI.F.

246. Yoshioka further discloses that the ECU 70 “uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.” EX1005, 9:22-27, 1:9-14 (“The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of an intake or exhaust valve of the engine according to a target timing.”).

247. It is my opinion that it would have thus been obvious to a POSA to use desired engine torque and the engine torque data to determine the “valve feedback term.” Indeed, by considering the desired engine torque and the engine torque data when calculating the “valve feedback term,” the valve open/close timings can be more accurately controlled to improve fuel efficiency and reduce harmful exhaust components. Doing so would have merely been applying a known technique in a similar, known device in the same way.

248. Thus, the combination of Yoshioka, Ohkubo, and Reed renders obvious claim 9.

XII. Objective indicia do not support patentability.

249. I understand that the obviousness analysis includes considering evidence of secondary indicia of nonobviousness (if available). I understand that any secondary indicia of nonobviousness must have a nexus, or a close connection, to the novel aspects of the claimed invention. I understand that secondary indicia of nonobviousness include the following: commercial success, long-felt but unmet need, failure of others, copying, praise in the industry, and unexpectedly superior results as compared with the closest prior art.

250. Considering the information I have reviewed, my general knowledge of the field of engine control systems, and publications within that and related fields, I am not aware of any objective evidence that might support the patentability of the claims of the '540 patent. Further, from my review of the file history (EX1002) of the '540 patent, I did not see that the applicant raised any secondary indicia of nonobviousness during prosecution. I am not otherwise aware of any publicly available evidence of secondary indicia of nonobviousness. Should any alleged secondary indicia of nonobviousness be presented, I reserve the right to provide an opinion thereon.

XIII. Conclusion

In signing this Declaration, I recognize that this Declaration will be filed as evidence in an *inter partes* review before the Patent Trial and Appeal Board of the United States Patent and Trademark Office. I also recognize that I may be subject to cross-examination in the case and that cross-examination will take place within the United States. If cross-examination is required, I will appear for cross-examination within the United States during the time allotted.

I hereby declare that all statements made herein are made of my own knowledge and are true and that all statements made on information and belief are believed to be true; and further that these statements were made with the knowledge that willful false statements and the like so made are punishable by fine or imprisonment, or both, under Section 1001 of Title 18 of the United States Code.

Executed on this 24th day of November, 2019.



Mark Ehsani, Ph.D., P.E.

M. Ehsani



M. Ehsani received the B.S. and M.S. degrees from the University of Texas at Austin in 1973 and 1974, respectively, and the Ph.D. degree from the University of Wisconsin-Madison in 1981, all in electrical engineering.

From 1974 to 1977 he was with the Fusion Research Center, University of Texas, as a Research Engineer. From 1977 to 1981 he was with Argonne National Laboratory, Argonne, Illinois, as a Resident Research Associate, while simultaneously doing the doctoral work at the University of Wisconsin-Madison in energy systems and control systems. Since 1981 he has been at Texas A&M University, College Station, Texas where he is now a Professor of electrical engineering and Director of Advanced Vehicle Systems Research Program and the Power Electronics and Motor Drives Laboratory. He is the author of over 400 publications in pulsed-power supplies, high-voltage engineering, power electronics, motor drives, and advanced vehicle systems and is the recipient of the Prize Paper Awards in Static Power Converters and motor drives at the IEEE-Industry Applications Society 1985, 1987, and 1992 Annual Meetings, as well as over 140 other honors and recognitions. In 1984 he was named the Outstanding Young Engineer of the Year by the Brazos chapter of Texas Society of Professional Engineers. In 1992, he was named the Halliburton Professor in the College of Engineering at A&M. In 1994, he was also named the Dresser Industries Professor in the same college. In 2001 he was selected for Ruth & William Neely/ Dow Chemical Faculty Fellow of the College of Engineering for 2001-2002, for “contributions to the Engineering Program at Texas A&M, including classroom instruction, scholarly activities, and professional service”. In 2003 he was selected for BP Amoco Faculty Award for Teaching Excellence in the College of Engineering. He was also selected for the IEEE Vehicular Society 2001 Avant Garde Award for “Contributions to the theory and design of hybrid electric vehicles”. In 2003 he was selected for IEEE Undergraduate Teaching Award “For outstanding contributions to advanced curriculum development and teaching of power electronics and drives.” In 2004 he was elected to the Robert M. Kennedy endowed Chair in Electrical Engineering at Texas A&M University. In 2005 he was elected as the Fellow of Society of Automotive Engineers (SAE). He is the co-author of 19 books on power electronics, motor drives and advanced vehicle systems, including Vehicular Electric Power Systems, Marcel Dekker, Inc. 2003 and “Modern Electric Hybrid Vehicles and Fuel Cell Vehicles – Fundamentals, Theory, and Design”, CRC Press, 2004. He is also the co-author of over 400 publications and has over 30 granted or pending US and EC patents. He has also been consultant to over 60 companies, US government agencies and international organizations. His current research work is in energy systems, power electronics, motor drives, hybrid vehicles and their control systems, and sustainable energy engineering.

Dr. Ehsani has been a member of IEEE Power Electronics Society (PELS) AdCom, past Chairman of PELS Educational Affairs Committee, past Chairman of IEEE-IAS Industrial Power Converter Committee and past chairman of the IEEE Myron Zucker Student-Faculty Grant program. He was the General Chair of IEEE Power Electronics Specialist Conference for 1990. He is the founder of IEEE Power and Propulsion Conference, the founding chairman of the IEEE VTS Vehicle Power and Propulsion and chairman of Convergence Fellowship Committees. In 2002 he was elected to the Board of Governors of VTS. He has also served on the editorial board of several technical journals and was the associate editor of IEEE Transactions on Industrial Electronics and IEEE Transactions on Vehicular Technology. He is a Life Fellow of IEEE, a past IEEE Industrial Electronics Society and Vehicular Technology Society Distinguished Speaker, IEEE Industry Applications Society and Power Engineering Society Distinguished Lecturer. He is also a registered professional engineer in the State of Texas.

BIOGRAPHICAL DATA

- Ehsani, Mehrdad (Mark)
- Professor, Electrical Engineering
- Birth date: 10/9/50
- Citizenship: U. S.
- Marital Status: Married
- Number of Children: Three
- Last Security Clearance: Secret

ADDRESS

Work: Texas A&M University
Department of Electrical Engineering
College Station, Texas 77843

PROFESSIONAL INTERESTS

- Electronics
- Solid State Power Systems
- Power Electronics
- Motor Drives
- Specialized Power Systems
- Control Systems
- Energy Storage Systems
- High Voltage Direct Current (HVDC) Power Transmission
- Applications of Microcomputers to Power Control
- Pulsed Power Systems
- Electric Hybrid Vehicles
- High Voltage Engineering
- Electrical Failures and Hazards
- Advanced Vehicle Power and Propulsion Systems
- Novel Electromagnetic Machines
- Sustainable Energy and Transportation

EDUCATION

- Ph. D., Electrical Engineering, University of Wisconsin, Madison, 1981
- M. S., Electrical Engineering, University of Texas, Austin, 1974
- B. S., Electrical Engineering, University of Texas, Austin, 1973

EXPERIENCE

- **Educational**
 1. Assistant Professor, Electrical Engineering, Texas A&M University, August 1981-1987

2. Associate Professor of Electrical Engineering, Texas A&M University, September 1987-1992
3. Professor, Electrical Engineering, Texas A&M University, 1992-present
4. Director, Texas Applied Power Electronics Center, Department of Electrical Engineering, Texas A&M University, 1982-present
5. Director of Advanced Vehicle Systems Research Program, College of Engineering, Texas A&M University, 1992-present

- **Industrial**

1. Research Engineer, Fusion Research Center, Austin, Texas, 1974-1977
2. Research Engineer, Argonne National Laboratory, 1977-1981
3. Consultant to over 65 U.S. and International Companies and Government Agencies

PROFESSIONAL SOCIETY MEMBERSHIPS

1. Institute of Electrical and Electronics Engineers (IEEE), since 1970
2. IEEE Industry Applications Society (IAS)
3. IEEE Industrial Electronics Society (IES)
4. IEEE Power Electronics Society (PELS)
5. IEEE Vehicular Technology Society (VTS)
6. Society of Automotive Engineers (SAE)
7. Member, American Energy Society
8. Registered Professional Engineer, Texas No. 57178

HONORS/AWARDS

1. Life Fellow of IEEE
2. Fellow of SAE
2. Outstanding Young Engineer of the year, 1984, Brazos Chapter, Texas Society of Professional Engineers.
3. Prize Paper First Place Award in Power Electronics, IEEE Industry Applications Society, 1985 and 1987 Annual Meetings.
4. Engineering Excellence Award (\$24,000), College of Engineering, Texas A&M University, 1986 and 1987.
5. Chief Editor for Power Systems Series, CRC Press, 1989.
6. General Chair, IEEE-PELS Power Electronics Specialists Conference, 1990.
7. Member of Editorial Board of Electric Machines and Power Systems Journal.
8. IEEE-Industrial Electronics Society Distinguished Speaker.
9. Plenary Session Author in IEEE Power Electronics Specialists Conference for 1990 and 1991.

10. Invited Author at IEEE Applied Power Electronics Conference and Exposition, Boston, February 1992.
11. Invited Author at 1992 International Symposium on Power Electronics, Seoul, Korea, April 1992.
12. Invited Author at International Aegean Conference on Electrical Machines and Power Electronics, Kusadasi, Turkey, May 1992.
13. Prize Paper Third Place Award in Motor Drives, IEEE Industry Applications Society 1992 Annual Meeting.
14. Invited Author and Panelist at 2nd International Power Electronics Congress, Cuernavaca, Mexico, August 1993.
15. IEEE Industry Applications Society Distinguished Speaker and Invited Author at II Brazilian Power Electronics Conference, Umberlandia, Brazil, December 1993.
16. Plenary Session Author in IEEE Power Electronics Specialists Conference for 1993.
17. Winner of IEEE IAS 1993 Annual Meeting Prize Paper Award from the Motor Drives Committee.
18. Invited Author at Southcon Technical Conference, Orlando Florida, March 1994.
19. Member of the Scientific Committee of the 1st International Power Electronics and Motion Control Conference, Beijing, China, June 1994.
20. Invited Author in 1st International Power Electronics and Motion Control Conference, Beijing, China, June 1994.
21. Member of the Advisory Committee of Second International Workshop on “The Future of Electronic Power Processing and Conversion,” Berg-en-dal, South Africa, August 1994.
22. General Chairman of 3rd International Power Electronics Congress, Puebla, Mexico, August 1994.
23. Listed in Who’s Who in America, 49th through current Editions.
24. Listed in American Men and Women of Science.
25. Listed in Who is Who in the South and Southwest.
26. Invited Short Course in Tel Aviv University, Israel, May 1995.
27. Invited Paper in International Aegean Conference on Electrical Machines and Power Electronics, Kusadasi, Turkey, June 1995.
28. Member of Steering Committee, World Congress of Industry Leaders and Educators, Fair of Engineering Innovations and UNESCO-UNISPAR Seminar, sponsored by Jozef Zych, Speaker of Polish Parliament, October 1996.
29. Invited Short Course on “Sensorless variable reluctance machines,” IEEE Industry Applications society 1996 Annual Meetings, October 1996.

30. Member of Steering Committee of the 1st International Congress in Israel on Energy, Power & Motion Control, Tel-Aviv, Israel, November 1996.
31. Honorary Professor of Electrical Engineering, The University of Hong Kong, 1996.
32. Blue Ribbon University Lecture at US Department of Transportation, Research and Special Programs Administration, "Electrically Peaking Hybrid (ELPH) Vehicles: A Sustainable Technology for the 21st Century," Washington, DC, April 25, 1997.
33. Mahdavi, A. Emadi, M. D. Bellar and M. Ehsani, "Analysis of Power Electronic Converters Using the Generalized State Space Averaging Approach," Invited Paper for Special Issue of IEEE Transactions on Circuits and Systems on Simulation, Theory and Design of Switched Analog Networks, IEEE Trans. on Circuit s and Systems, vol. 44, no. 8, August 1997.
34. Invited Paper on "Soft Switching Motor Drive Inverters for Electric and Hybrid Vehicles," in IEEE Industrial Electronics Annual Meeting, IECON'97, New Orleans, Louisiana, November 1997.
35. Invited Panelist on "Trends in Power Electronics and Motor Drives," Invited Paper in IEEE Industrial Electronics Annual Meeting, IECON'97, New Orleans, Louisiana, November 1997.
36. Elected IEEE Industry Applications Society Distinguished Lecturer, 1998-99.
37. Invited Lecture, entitled "Modern Motor Drives for Industrial and Product Applications," IEEE Chapter Meeting, Little Rock Arkansas, January 22, 1998.
38. National Science Foundation Site Reviewer for NSF Center Proposal, Blacksburg, Virginia, Jan. 19-22, 1998.
39. Invited Speaker, IEEE Power Engineering Society Section Meeting, Columbus, Ohio, March 31, 1998.
40. Key Note Speaker, Southeast Michigan Section Spring Meeting, April 2, 1998, Dearborn, Michigan.
41. Invited Key Note Speaker on "State of The Art in Power Electronics and Motor Drives," Vicenza Trade Fair and Workshop, Vicenza, Italy, May 15, 1998.
42. Invited Seminar at Toshiba Small Motor Development Center, "Advanced Switched Reluctance Motor Drives for Industrial and Traction Applications," July 14, 1998.
43. M. Ehsani, R. Velayutham, S. Gopalakrishnan, and B. Fahimi, "Sensorless Control of Switched Reluctance Motor: a Technology Ready for Applications," Invited Paper, International Conference on Electrical Machines, Istanbul, Turkey, September 2-4, 1998.
44. Member of Program Committee, International Conference on Electronics, October 1998, Oran, Algeria.
45. Invited Paper, entitled "Switched Reluctance Motor Drives: State of the Art and Applications," 1998 Special Issue of Indian Academy of Sciences.
46. Member International Steering Committee, The IEEE International Symposium on Diagnostics for Electrical Machines, Power Electronics and Drives, Gijon, Spain, September, 1999.

47. Organizer and Plenary Paper Presenter for Invited Session at 1999 International Federation of Automatic Control (IFAC) World Congress in Beijing, China: Advances in Real-Time DSP Control of Stiff Systems.
48. Organizer Presenter for Tutorial Session at 1999 American Control Conference (ACC) in San Diego, California: Advances in Real-Time Control of Motor Drive Systems.
49. Listed in Dictionary of International Biography, Cambridge, England, 27th Edition, 1999.
50. Member of International Steering Committee of IEEE International Power Electronics Congress, Acapulco, Mexico, 2000.
51. Member of Technical Program Committee of 4th International Power Electronics Conference (IPEC-Tokyo 2000).
52. Member of Technical Program Committee of 3rd International Power Electronics and Motion Control Conference (IPEMC, 2000).
53. Appointed as the Distinguished Lecturer of IEEE Power Engineering Society, 1999-2000.
54. Member of International Steering Committee of the 1999 IEEE International Symposium on Diagnostics for Electrical Machines, Power Electronics and Drives, September 1-3, Spain.
55. Invited Seminar to the Local Chapter of IEEE in Istanbul, Turkey on “State of the Art in Power Electronics and Motor Drives,” August 27, 1998.
56. Invited Seminar to the Local Chapter of IEEE in Paris, France on “An Overview and State of the Art in Hybrid Electric Vehicles,” September 23, 1999.
57. Invited Seminar to the Local Chapter of IEEE in Heidelberg, Germany on “State of the Art and Recent Applications of Switched Reluctance Motor Drives,” November 19, 1999.
58. Invited Seminar at the Swiss Federal Institute of Technology on “State of the Art on Sensorless Switched Reluctance Motors,” Zurich, Switzerland, May 9, 2000.
59. Invited Short Course at Tel Aviv University on “Hybrid Electric Vehicles,” Tel Aviv, Israel, May 23-25, 2000.
60. Invited Seminar at Technion University on “State of the Art on Electric and Hybrid Vehicles,” Haifa, Israel, May 29, 2000.
61. Invited Presentation in First Sned Research Institute Workshop on Fuel Cell Hybrid Track Vehicles, August 1-3, 2000, Georgetown, Texas.
62. Advisor to the Board of Directors of Sned Research Institute.
63. Listed in International Who’s Who of Professionals, 2001.
64. Member of International Steering Committee of VII IEEE Power Electronics Congress, Acapulco, Mexico, October 2000.

65. Reference Publications Listed in Infography Publishers' Fields Of Knowledge.com, in Switched Reluctance Motor Drives, Solid State Power Systems, Brushless DC Motor Drives, Hybrid Electric Vehicles, and Synchronous Reluctance Motor Drives.
66. Appointed as a Member of Committee on Next Generation Materials and Processes for Advanced Hybrid Power Systems, National Materials Advisory Board, National Research Council, National Academy of Engineering, September 2000.
67. Member of International Steering Committee of CIEP'02, Guadalajara, Mexico.
68. Panelist on hybrid electric vehicles, National Academy of Engineering Mini-Symposium, March 19, 2001, College Station, Texas.
69. Selected for Ruth & William Neely/ Dow Chemical Faculty Fellow of the College of Engineering for 2001-2002, for "contributions to the Engineering Program at Texas A&M, including classroom instruction, scholarly activities, and professional service".
70. Member of Steering Committee of International Aegean Conference on Electrical Machines and Power Electronics, Kusadasi, Turkey, June 2001.
71. Invited paper at the SAE 2001 *Future Transportation Technology Conference*, Costa Mesa, CA, August 2001, "42V Automotive Power Systems".
72. Winner of 2001 Avant Garde Award from IEEE Vehicular Technology Conference for "Contributions to the Theory and Design of Hybrid Electric Vehicles."
73. Listed in Cambridge International Biographical Center, Who's Who in the 21st Century.
74. Invited external Ph. D. examiner on hybrid vehicle technology at the University of Vallenciennes, France, Jan. 4, 2002.
75. Guest Editor, Special Issue on Switched Reluctance Motor Drives, IEEE Transactions on Industrial Electronics, Vol. 49, No. 1, Feb. 2002.
76. Winner of 2003 IEEE Undergraduate Teaching Award, with the following citation: "For outstanding contributions to advanced curriculum development and teaching of power electronics and drives."
77. Member of International Steering Committee of 2003 International Power Electronics and Motion Control Conference, Xi'an, China.
78. Elected Distinguished Speaker of IEEE Vehicular Technology Society, 2003-2004.
79. Winner of Texas A&M Technology Licensing Office "Spirit of Innovation Award for the 2000th Disclosure", 2003.
80. Listed in the International Biographical Center of Cambridge, England Outstanding People of 20th Century.
81. Selected as an International Educator of the Year for 2003 by International Biographical Center of Cambridge, England.
82. Appointed to Research Visionary Board, Chief Technology Office, Motorola Corporation, 2003.

83. Invited for Distinguished Lecture in the Department of and Computer Engineering Illinois Institute of Technology, Chicago, Ill. August 29, 2003.
84. Selected for Texas A&M College of Engineering BP Amoco Faculty Award for Teaching Excellence, 2003.
85. Associate Editor of International Journal of Electrical Engineering in Transportation
86. “Integrated Biofuel-Vehicle System”, Key Note address given at the opening of the 2004 IEEE-VTS Vehicle Power and Propulsion Symposium in Paris, France, October 4th, 2004
87. Elected to the Robert M. Kennedy endowed Chair of Electrical Engineering in 2004.
88. Served on the National Research Council Review Committee for NASA LEAP Program proposals, 2005.
89. Elected as Fellow of Society of Automotive Engineers (SAE).
90. Served on Motorola Corporation Research Visionary Board, 2005.
91. Listed in Who’s Who in American Education.
92. Listed in Who’s Who in Science and Engineering.
93. Listed in Who’s Who in Finance and Business
94. Listed in Who’s Who in South and Southwest
95. Listed in Who’s Who in America
96. Listed in Covington Who's Who
97. Member of international steering committee, 5th International Power Electronics and Motion Control Conference (IPEMC2006), shanghai,China , August of 2006.
98. Key Note Speech Presenter, IEEE Vehicle Power and Propulsion Conference/ SAE Future Transportation Technology Conference Joint session, “Integrated Biofuel Vehicle Systems,” September 7, 2005, Chicago, Illinois.
99. Member of National Research Council Panel for Decadal Survey of Civil Aeronautics, 2005-2006.
100. Key Note Speaker, ELECO 2005, “A Vision of Sustainable Vehicles and Fuel Technologies,” Istanbul, Turkey.
101. Key Note Presentation, IEEE Vehicle Power and Propulsion Conference, VPPC, “ A Vision of Renewable Energy and Sustainable Transportation,’ Windsor, England, September, 2006.
102. Member of the Advisory Committee and Key Note Speaker of Automotive Research Association of India Electric and Hybrid Electric Vehicles Conference, December 18-20, 2006, Pune, India.
103. Member of Italian Ministry for University and Research International Review Panel.

104. Member of Hong Kong Scientific and Engineering Review Panel.
105. Recipient of International Biographical Center, Cambridge, England, 21st Century Award for Achievement.
106. Listed in Who's Who in America, 61st Edition, 2007.
107. Member of Advisory Board of Automotive X Prize Foundation, Inc.
108. Member of the Steering Committee of International Conference on Electrical and Electronics Engineering, ELECO'2006 & ELECO'2007.
109. 2007 Innovation Award, Texas A&M Technology Lisencieng Office.
110. Key Note Speaker, MIT Automotive Consortium meeting, Seattle, WA, April, 2007.
111. Key Note Speaker, IEEE Vehicle Power and Propulsion Conference, VPPC07, "Sustainable Fuel and Vehicle Technologies: Update on Progress & Problems," Arlington, Texas, September 10-12, 2007.
112. Key Note Speaker, IEEE Vehicle Power and Propulsion Conference, VPPC 08, "Issues on Sustainable Energy and Transportation," Harbin, China, September 3, 2008.
113. Member of Steering Committee, 6th IEEE International Power Electronics and Motion Control Conference (IPEMC 2009), Wuhan, China, during May 17-20, 2009.
114. Listed in Who's Who in Finance and Business.
115. IEEE Vehicular Technology Society 2008 VTS Outstanding Service Award.
116. Member, Automotive X PRIZE Development Advisory Board.
117. Key Note Speaker at IEEE Vehicle Power and Propulsion Conference, VPPC 09 "Sustainable Energy Engineering," Dearborn, Michigan, September 2009.
118. Invited short Course, Technological University of Panama, "Sustainable energy engineering,' June, 2010.
119. Winner of 2011 Texas A&M University System Patent and Innovation Award.
120. Inaugural ABB Distinguished Lecturer, North Carolina State University, March 2012.
121. Member of the organizing committee, Power Train Modeling and Control Conference, PMC2012, September 4-6, 2012.
122. Winner of Texas A&M University Office of Technology Commercialization Patent and Innovation 2012 Award.
123. Technical Advisor to the government panel on National Integrated Sustainable Power System of Pananam-2050.
124. Panelist: Aspen Clean Energy Round Table, June 4-6, 2012.
125. Member of Organizing Committee, TMC 2012 (Testing, Mapping and Calibration Conference) 4th to 6th of September 2012, University of Bradford, West Yorkshire, UK.

126. Key note speaker and Program Co-Chair, 2012 International Conference on Connected Vehicles & Expo (ICCVE 2012), December 2012, Beijing, China.
127. Key Note speaker at IEEE Green Technologies Conference, Denver, Colorado, April 4-5, 2013.
128. Invited Distinguished Lecture, “Sustainable Energy Engineering,” University of Colorado, Boulder, September 19, 2013.
129. Listed in "Who's Who in North American Education (2014-2015 Edition)."
130. Key note on “Sustainable Energy and Transportation Engineering” at IEEE Green Tech’14, Corpus Christi, Texas, April, 2014.
131. Keynote on “Sustainable Energy Engineering” at 3rd International Conference in Renewable Energy Research and Applications, Milwaukee, October 2014.
132. IEEE Vehicular Technology Distinguished Lecture “ Evolution of Electric Drivetrain,” McGill-IEEE Symposium on Advanced Electric Vehicle Drivetrains, November 14, 2014, Montreal, Canada.
133. Invited seminar at University of Texas at Austin, Energy Institute, “Engineering and Socio-economic aspects of sustainable energy,” January 22, 2015.
134. M. Ehsani, “Confluence of Fossil and Sustainable Energy Sources,” keynote at IEEE Green Tech’15, New Orleans, April, 2015.
135. Elected to IEEE Life Fellow, 2015.
136. IEEE Power Engineering Society Distinguished Lecture on sustainable energy, Austin, Texas chapter, February 23, 2016.
137. Invited keynote at 3rd Biennial International Conference on Powertrain Modelling and Control Testing, Mapping and Calibration, Loughborough University, UK, September 7, 2016.
138. Invited honorary lectures on “Electrification of Vehicles” at the Gordon Institute, Tel Aviv University, Israel, June 2017.
139. Guest Editor of IEEE Transactions on Energy Conversion Special Edition on “Emerging Electric Machines and Drives for Smart Energy Conversion,” 2017.
140. Invited IEEE Power Engineering Society Distinguished Speaker on “An Over View of High Voltage DC Power Transmission,” Austin, Texas, March 2018.
141. Invited IEEE Power Engineering Society Distinguished Speaker on “Engineering and Socio-Economic Aspects of Sustainable Energy and Transportation,” San Diego, California, April 2018.
142. Recipient of IEEE Vehicular Society Outstanding Contribution Recognition certificate, 2015-2018.
143. Honorary Lecture on “Engineering and Socio-Economic Aspects of Sustainable Energy and Transportation”, Texas A&M University, Kingsville, October, 2018.
144. Listed in Who’s Who in America, 2019 edition.
145. Best Paper Award, “A Review of the More Electric Aircraft Power Electronics,” Texas Power and Energy Conference (TPEC), College Station, Texas, February, 2019.

146. Invited keynote “ Engineering and Socio-Economic Aspects of Sustainable Energy and Transportation,” 2019 MIT Applied Energy Symposium, May 22-24, 2019.

ACADEMIC CHAIRS

- Halliburton Professorship-1992.
- Dresser Industries Professorship-1994.
- Robert M. Kennedy endowed Chair of Electrical Engineering, since 2004.

SAMPLE SHORT COURSES

1. Invited Short Course offered by Electric Power Research Institute (EPRI), Power Electronic Applications Center (PEAC), June 1989.
2. Invited Short Course in IEEE Applied Power Electronics Conference Exposition, Dallas, March 1991: “Pulse Width Modulation: Fundamentals and Implementation.”
3. Invited Short Course in IEEE Applied Power Electronics Conference Exposition, San Diego, March 1993: “Technology Simplification in Switched Reluctance Motor Drives.”
4. Invited Short Course at Tel Aviv University, November 1993: “Super Conductive Magnet Power Supplies.”
5. Invited Short Course in IEEE Applied Power Electronics Conference and Exposition, Orlando, Florida, February 1994: “High Power Soft Switched Converters.”
6. Invited Short Course in IEEE Applied Power Electronics Conference and Exposition, Dallas, Texas, March 1995: “Electric Drives in Electric and Hybrid Vehicles.”
7. Invited Short Course at GM Proving Grounds in Milford, MI and Mesa, AZ, September/October, 1995: “Electric Drives in Electric and Hybrid Vehicles.”
8. Invited Short Course at LeTourneau, Longview, Texas and Texas Instruments, Dallas Texas, January 12 & 26, 1996: “Design and Control of Switched Reluctance Motor Drives.”
9. Invited Short Course at IEEE Industry Applications Society 1996 Annual Meeting, San Diego, California, October 6, 1996: “Sensorless Variable Reluctance Motors.”
10. Invited Short Course at Tel Aviv University, May 22-25, 2000: “Design and Simulation of Electric and Hybrid Electric Vehicles.”
11. Invited Short Course at Hanyang University, Seoul, Korea, June 26, 2000: “State of the Art of Brushless DC Motor and Switched Reluctance Motor Drives.”
12. Invited Short Course at LG Electronics Company, Seoul, Korea, June 27, 2000: “Sensorless Controls and Reduced Parts Converters for the Advanced BLDC and SRM Drives.”
13. Invited Short Course at Hyundai Motor Company, Suwon, Korea, June 28, 2000: “Design and Control of Electric and Hybrid Electric Vehicles.”
14. Invited Short Course at Korean Electromagnetic Research Institute (KERI), Changwon, Korea, June 29, 2000: “Design and Control of Switched Reluctance Motors and Electric and Hybrid Electric Vehicle”

15. Sensorless Brushless DC Motor Drives for Integrated in-line automotive Pump Applications, Short course given at EMP Corp. Escenaba, Michigan, August, 2004.
16. “Control of BLDC Machines with Improved Performance”, U.S. Army Vetronics Institute 3rd Annual Winter Workshop, Jan 13, 2004, U.S. Army Tank-Automotive RD&E Center Warren, MI
17. “Control of the BLDC machine with improved performance”, Short Course, June 2004, Tel Aviv University, Israel.
18. “ Advanced Mobile Integrated Power System (AMPS)”, Short Course, January 2005, US Army Tank Automotive Command (TACOM), Warren, Michigan.
19. “ Vehicle Power systems”, Short Course, January 2006, US Army Tank Automotive Command (TACOM), Warren, Michigan.
20. “ Short Course on Hybrid Electric Vehicle”, **TUBITAK Marmara Research Center, TURKE**, Dec. 2005.
21. IEEE Vehicular Technology Distinguished Lecture, Isreal Chapter, Tel Aviv, Dec. 2005.
22. “ Short Cours on Advanced Controls for Brushless DC Motor Drivres,” IEEE Applied Power Electroncis Conference, Dallas, Texas, March, 2006.
23. M. Ehsani, “Short Course on Sustainable Energy Engineering,” University of Tel Aviv, December 2010, invited short course.
24. M. Ehsani “Short Course on Sustainable Energy and Transportation,” Technical University of Panama, June 2010, invited short course.

PROFESSIONAL COMMITTEE MEMBERSHIPS (past or present)

- IEEE Power Electronics Society (PELS) Administrative Committee
- IEEE Industry Applications Society (IAS) Executive Committee.
- IEEE Myron Zucker Faculty-Student Grant Program Chairman.
- IEEE IAS Industrial Power Converter Committee (IPCC), Chairman.
- IEEE-PELS Meetings Committee Member.
- IEEE-IAS Industrial Drives Committee Member.
- IEEE IAS Education Committee Member.
- IEEE-PELS Power Electronics Specialists Conference Program Committee Member.
- IEEE 1994 Power Electronics Specialists Conference Topic Chairman in “Motor Drives”
- IEEE Power Electronics Society Energy Committee Chairman
- Editorial Board of Electric Machines and Power Systems Journal Member
- IEEE Power Electronics Society Chairman of Education Committee
- IEEE Power Electronics Society Intersociety Liaison
- IEEE Transactions on Industrial Electronics, Associate Editor
- IEEE Transactions on Vehicular Technology, past Associate Editor
- IEEE Vehicular Technology Society Chairman of Vehicle Power and Propulsion Committee
- IEEE Vehicular Technology Society Chairman of Convergence Fellowship Committee
- Member of IEEE Vehicular Technology Society Board of Governors
- Member of IEEE Vehicular Society Fellows Nominations Committee
- Chair of IEEE Vehicular Technology Society, Vehicles Committee

UNIVERSITY SERVICE

1. Member, Graduate Studies Committee, since 1982.
2. Member, Electrical Engineering Administrative Council, Hiring Committee, 1986.
3. Member, Doctor of Engineering Program Committee, Electrical Engineering Department, 1989.
4. Founding Member, Deans Committee on Engineering Scholars Program, College of Engineering, 1988, 1989, 1990, 1991.
5. Electric Power Institute (EPI) member.
6. Texas A&M Metermens School Lecturer, 1982-present.
7. Member, Department Awards Committee, 1992-present.
8. Chairman of the University Council of Principle Investigators, 1994-1995.
9. Member of Interdisciplinary Research Task Force, 1994.
10. Member of University Research Council Subcommittee on Conflict of Interest, 1994-1995.
11. Member of Departmental Awards Committee, 1995.
12. Member of Departmental Tenure and Promotion Committee, 1996- 1997.
13. Faculty Advisor to IEEE Student Chapter of Power and Power Electronics at Texas A&M University.
14. Member of Departmental Advisory Committee, 1999-2005.
25. Member Departmental Tenure and Promotion Committee, 2000-2002.
26. Member Departmental Distinguished Lecturer Selection Committee, 2003- 2005.
27. Member Departmental Tenure and Promotion Committee, 2008-present.
28. Chair of Department Awards commit 2011.
29. Departmental Representative at the Engineering Faculty Advisory Committee, EFAC, College of Engineering.
30. Member, Departmental Graduate and Admissions Committee.

PUBLICATIONS

- **Books**

1. Co-author: IEEE Guide for Self-Commutated Converters, ANSI/IEEE Std. 936, 1987.
2. Converter Circuits for Superconductive Magnetic Energy Storage, Co-Author: R. L. Kustom, Texas A&M University Press, 1988.
3. Contributor of Chapter on “Switched Reluctance Motor Drives” to Encyclopedia of Electrical and Electronics Engineering, John Wiley & Sons.
4. Contributor of Chapter on “Harmonic and Power Factor Control” to Encyclopedia of Electrical and Electronics Engineering, John Wiley & Sons.
5. Modern Electrical Drives, Co-Author: H. B. Ertan, et al., Kluwer Academic Publishers, 2000, printed in the Netherlands.
6. Contributor of a chapter on “ More Electric Vehicles” to CRC Handbook of Power Electronics, 2002.
7. Contributor of a chapter on “More Electric Aircraft” to CRC Handbook of Power Electronics, 2002.
8. Contributor to SAE book “Hybrid Electric Vehicles,” SAE SP-1633, published in 2001.
9. Co-author, “Combat Hybrid Power Systems Technologies, Technical Challenges and Research Priorities,” a report of National Research Council of the National Academies, 2003.
10. Vehicular Electric Power Systems, my Co-Authors: A. Emadi & JM Miller, Marcel Dekker, Inc. 2004.
11. Chapters contributor, “The 42-Volt Electrical System,” Book, Society of Automotive Engineers, Inc. PT-99, ISBN 0-7680-1297-X, 2003.
12. “Modern Electric, Hybrid Electric, and Fuel Cell Vehicles – Fundamentals, Theory, and Design”, M. Ehsani, Y. Gao, S. E. Gay, A. Emadi, CRC Press, 2005.
13. Contributor of chapter on “Hybrid Drive Trains” to “Handbook of Automotive Power Electronics and Motor Drives” CRC Press, 2005.
14. “Modern Electric, Hybrid Electric, and Fuel Cell Vehicles – Fundamentals, Theory, and Design”, M. Ehsani, Y. Gao, A. Emadi, CRC Press, Second Edition, 2010.
15. “Encyclopedia of Sustainability Science and Engineering” ISBN 978-0-387-89469-0, Springer, 2012.
16. “Transportation Technologies for Sustainability: Selected Entries from the Encyclopedia of Sustainability Science and Technology”, ISBN 978-1-4614-5843-2, Springer, 2013.
17. “Alternative Fuels and Advanced Vehicle Technologies for Improved Environmental Performance: Towards Zero Carbon Transportation,” ISBN: 978-0-85709-522-0 Woodhead Publishing, March, 2014.

18. “Modern Electric, Hybrid Electric, and Fuel Cell Vehicles – Fundamentals, Theory, and Design”, M. Ehsani, Y. Gao, S. Longo, K. Ebrahimi, CRC Press, 3rd Edition, 2018.

- **Patents**

1. “Position Sensor Elimination Technique for the Switched Reluctance Motor Drive,” Inventor: M. Ehsani, US Patent No. 5,072,166, December 1991.
2. “Inverse Dual Converter for High-Power Applications,” Inventor: M. Ehsani, US Patent No. 5,208,740, May 1993.
3. “Power Conversion Using Zero Current Soft Switching,” Inventor: M. Ehsani, US Patent No. 5,287,261, February 1994.
4. “Phase and Amplitude Modulation Techniques for Rotor Position Sensing in Switched Reluctance Motors,” Inventor: M. Ehsani, US Patent No. 5,291,115, March 1994.
5. “Method and Apparatus for Sensing the Rotor Position of a Switched Reluctance Motor,” Inventor: M. Ehsani, US Patent No. 5,410,235, April 1995.
6. “Indirect Rotor Position Sensor for A Sinusoidal Synchronous Reluctance Machine,” Inventors: M. Ehsani, M. S. Arefeen, and T. A. Lipo, U. S. Patent No. 5,448,149, September 1995.
7. “The Electrically Peaking Hybrid System and Method,” Inventor: M. Ehsani, US Patent No. 5,586,613, December 24, 1996.
8. “Capacitive Power Circuit,” M. Ehsani, US Patent No. 5,852,358, December 22, 1998.
9. “Switched Reluctance Motor Drive System,” M. Ehsani, US Patent Pending, Filing Date: January 1997, Serial Number 60/061,087.
10. “Method and Apparatus for Sensing the Rotor Position of a Switched Reluctance Motor without a Shaft Position Sensor,” M. Ehsani, European Patent No. 93923117.1, April 1997.
11. “Self-Tuning Control of Switched Reluctance Motor Drives System,” M. Ehsani, US Patent No. 6,472,842, October 29, 2002.
12. “Method and Apparatus of Indirect Torque Estimation in Switched Reluctance Motor Drives,” US Patent Pending.
13. “An Efficient Power Circuit for Capacitor Charging and Discharging,” US Patent Pending.
14. “Sensing of Rotor Position of a Switched Reluctance Motor without a Shaft Position Sensor,” European Patent No. EP0662265, August 1998.
15. “Self-Tuning Control of Switched Reluctance Motor Drive System,” Patent Pending.
16. “Inductance Model based Indirect Rotor Position Sensing Scheme for Switched Reluctance Motor (SRM) Drive,” Patent Pending.

17. “ Method and Apparatus for Reducing Noise and Vibration in Switched Reluctance Motor Drives,” US patent pending.
18. “Application of Active Composite Wheels to All Terrain Vehicle,” US patent disclosure.
19. “The Series-Parallel HEV with a Transmotor,” US patent disclosure.
20. “Fuel Cell Powered Hybrid Drive Train and its Control Strategy,” US patent disclosure.
21. “A New type of Transmission for Hybrid Vehicle with Speed and Torque Summation,” US patent disclosure.
22. “Method and Apparatus for Self Tuning Control of Switched Reluctance Motor Drives,” US Patent No. is 6,472,842, Oct 29, 2002.
23. “Virtual-Actual Laboratory for Power Electronics and Motor Drives with Remote Access”, patent pending, 2/25/04.
24. M. Ehsani and Taehyung Kim, “Advanced Sensorless Drive Technique for Brushless DC Motors,” US patent pending, 60/438,949, Jan. 2004
25. M. Ehsani, et al, “ High Torque Switched Reluctance Motor,” U.S. Patent Application Serial No. 11/379,174.
26. M. Ehsani, et al, “ Generator Apparatus for a Quasi-Isothermal, Brayton Cycle Engine, U.S. Patent No. 7,008,200, March 7, 2006.
27. M. Ehsani, “System and Method for Inductance Based Position Encoding Sensorless SRM Drive,” U.S. Patent No. 7,095,205, Aug. 22, 2006.
28. M. Ehsani, “ Methods and Apparatus for Operating Hydrogen Enriched Spark Ignition Internal Combustion Engine Vehicle Propulsion System with On Board Plasma Reformer,” patent disclosure to University Office of Technology Licensing.
29. M. Ehsani, “Method and Apparatus for Hybrid Vehicle Drive Train Using Cold Plasma Reformer and Spark Ignition Internal Combustion Engine Fueled Partial Hydrogen Enriched Natural Gas, Ethanol, E85, or other Hydrocarbon fuels,” patent disclosure to University Office of Technology Licensing.
30. M. Ehsani, “ Method and Apparatus for a Cold Plasma Reformer-Compression Internal Combustion Engine Vehicle Propulsion System,” patent disclosure to University Office of Technology Licensing.
31. M. Ehsani, “ Methods and Apparatus for Generating Hydrogen Rich Gas From Hydrocarbon Fuels by Cold Microwave Plasma Reforming,” patent disclosure to University Office of Technology Licensing.
32. M. Ehsani,” Plug-In Hydraulic-Electric Hybrid Vehicles,” patent disclosure to University Office of Technology Licensing.
33. M. Ehsani, “ Power Stage and Control Algorithm Design for Balance of Plant Operation in Fuel Cell-Battery Off-Line UPS System,” patent disclosure to University Office of Technology Licensing.

34. M. Ehsani, “Plug in Hybrid Drive Train with Partial Hydrogen Boosted Natural Gas Engine and Full Recovery of Potential Energy in Highly Compressed Natural Gas,” patent disclosure to University Office of Technology Licensing.
35. M. Ehsani, et. al., “Electric Machine Having A High-Torque Switched Reluctance Motor,” US patent No. 7,663,283,B2, February 2010.
36. M. Ehsani, et. al., “Electric Machine Having Rotor and Stator Configurations,” US patent No. 8,022,586 B2, September 2011.

(Other new patent disclosures in progress)

□ **Journal Papers**

1. R. E. Fuja, R. L. Kustom, and M. Ehsani, “Three-Phase Energy Transfer Circuit with Superconducting Energy Storage Coils,” *IEEE Trans. on Industry Applications*, Vol. IA-16, No. 3, May/June 1980, pp. 438-444.
2. M. Ehsani, R. L. Kustom, and R. E. Fuja, “Microcomputer Control of a Current Source DC-DC Converter,” *IEEE Trans. on Industry Applications*, Vol. IA-19, No. 5, September/October 1983, pp. 690-698.
3. M. Ehsani, R. L. Kustom, and R. W. Boom, “A One-Phase Dual Converter for Two Quadrant Power Control of Superconducting Magnets,” *IEEE Trans. on Magnetics*, Vol. MAG-21, No. 2, March 1985, pp. 1115-1118.
4. M. Ehsani, et al., “High MVA Interrupters Using the GTO,” *Bulletin of American Physical Society*, Abstract, May 1985.
5. M. Ehsani, et al., “Converter Networks for Superconducting Magnetic Systems,” *Bulletin of American Physical Society*, May 1985.
6. M. Ehsani and R. L. Kustom, “A Bilateral Power Supply with Energy Storage Buffer for the Superconductive Coils of Large Particle Accelerators,” *IEEE Trans. on Nuclear Science*, Vol. NS-32, No. 5, October 1985, pp. 3781-3783.
7. M. Ehsani and W. H. Kernaghan, “Evaluation of the GTO for 270 MVA Interrupter Applications,” *IEEE Trans. on Nuclear Science*, Vol. NS-32, No. 5, October 1985, pp. 3784-3786.
8. J. T. Bass, M. Ehsani, and T. J. E. Miller, “Robust Torque Control of Switched-Reluctance Motors without a Shaft Position Sensor,” *IEEE Trans. on Industrial Electronics*, Vol. IE-33, No. 3, August 1986, pp. 212-216.
9. M. Ehsani, A. Hozhabri, and R. L. Kustom, “Decoupled Control Techniques for Dual Flying Capacitor Bridge Power Supplies of Large Superconductive Magnets,” *IEEE Trans. on Magnetics*, Vol. MAG-23, No. 2, March 1987, pp. 591-594.
10. J. T. Bass, M. Ehsani, and T. J. E. Miller, “Simplified Electronics for Torque Control of Sensorless Switched-Reluctance Motor,” *IEEE Trans. on Industrial Electronics*, Vol. IE-34, No. 2, May 1987, pp. 234-239.
11. M. Ehsani, J. T. Bass, T. J. E. Miller, and R. L. Steigerwald, “Development of a Unipolar Converter for Variable Reluctance Motor Drives,” *IEEE Trans. on Industry Applications*, Vol. IA-23, No. 3, May/June 1987, pp. 545-553.

12. Y. H. Kim and M. Ehsani, "An Algebraic Algorithm for Microcomputer Based (Direct) Inverter Pulse-Width Modulation," *IEEE Trans. on Industry Applications*, Vol. IA-23, No. 4, pp. 653-660, July/August 1987; For a detailed discussion of this article, see - Bowes, Sidney, Y. H. Kim and M. Ehsani, "Discussion on an Algebraic Algorithm for Microcomputer Based (Direct) Inverter Pulse-Width Modulation." *IEEE Trans. on Industry Applications*, Vol. IA-24, No. 6, November/December 1988, pp. 998-1004.
13. M. Ehsani, J. T. Bass, and T. J. E. Miller, "High Speed Torque Control of Permanent Magnet Brushless DC Motors," *IEEE Trans. on Industrial Electronics*, Vol. IE-37, No. 3, August 1988, pp. 402-406.
14. M. O. Bilgic and M. Ehsani, "Time Averaged Behavior of Single and Dual Flying Capacitor Converter of Superconductive Magnets," *International Journal of Electronics*, Vol. 66, No. 4, 1989, pp. 655-663.
15. T. M. Jahns, R. C. Becerra, and M. Ehsani, "Integrated Current Regulation for a Brushless ECM Drive," *IEEE Trans. on Power Electronics*, Vol. 6, No. 1, January 1991, pp. 118-126.
16. M. Ehsani, I. Husain, and A. B. Kulkarni, "Elimination of Discrete Position Sensor and Current Sensor in Switched Reluctance Motor Drives," *IEEE Trans. on Industry Applications*, Vol. IA-28, No. 1, January/February 1992, pp. 128-135; also cited in K. Rajashekara, et al., eds. *Sensorless Control of AC Motor Drives: Speed and Position Sensorless Operation*, IEEE Press, 1996, pp. 463-470.
17. A. B. Kulkarni and M. Ehsani, "A Novel Position Sensor Elimination Technique for the Interior-Permanent Magnet Synchronous Motor Drive," *IEEE Trans. on Industry Applications*, Vol. 28, No. 1, January/February 1992, pp. 144-150; also cited in K. Rajashekara, et al., *Sensorless Control of AC Motor Drive: Speed and Position Sensorless Operation*, IEEE Press, 1996, pp. 366-372.
18. R. C. Becerra, M. Ehsani, and T. M. Jahns, "Four-Quadrant Brushless ECM Drive with Integrated Current Regulation," *IEEE Trans. on Industry Applications*, Vol. IA-28, No. 4, July/August 1992, pp. 833-841.
19. M. Ehsani, I. Husain, K. R. Ramani, and J. H. Galloway, "Dual Decay Converter for Switched Reluctance Motor Drives in Low Voltage Applications," *IEEE Trans. on Power Electronics*, Vol. 8, No. 2, April 1993, pp. 224-230.
20. M. Ehsani, O. H. Stielau, J. D. Van Wyk, and I. J. Pitel, "Integrated Reactive Components for Power Electronic Circuits," *IEEE Trans. on Power Electronics*, Vol. 8, No. 2, April 1993, pp. 208-215.
21. M. Ehsani, I. Husain, and M. O. Bilgic, "Inverse Dual Converter (IDC) for High Power DC-DC Applications," *IEEE Trans. on Power Electronics*, Vol. 8, No. 2, April 1993, pp. 216-223.
22. R. C. Becerra, M. Ehsani, and T. J. E. Miller, "Commutation of SR Motors," *IEEE Trans. on Power Electronics*, Vol. 8, No. 3, July 1993, pp. 257-263.
23. M. Ehsani and M. O. Bilgic, "Power Converters as Natural Gyrators," *IEEE Trans. on Circuits and Systems*, Vol. 40, No. 12, December 1993, pp. 946-949.
24. M. Ehsani, I. Husain, S. Mahajan, and K. R. Ramani, "New Modulation Encoding Techniques for Indirect Rotor Position Sensing in Switched Reluctance Motors," *IEEE Trans. on Industry Applications*, Vol. 30, No. 1, January/February 1994, pp. 85-91.

25. I. Husain and M. Ehsani, "Error Analysis in Indirect Rotor Position Sensing of Switched Reluctance Motors," *IEEE Trans. on Industrial Electronics*, Vol. 41, No. 3, June 1994, pp. 301-307.
26. M. Ehsani and I. Husain, "Rotor Position Sensing in Switched Reluctance Motor Drive by Measuring Mutually Induced Voltages," *IEEE Trans. on Industry Applications*, Vol. 30, No. 3, May/June 1994, pp. 665-672.
27. M. S. Arefeen, M. Ehsani, and T. A. Lipo, "An Analysis of the Accuracy of Indirect Shaft Sensor for Synchronous Reluctance Motor," *IEEE Trans. on Industry Applications*, Vol. 30, No. 5, September/October 1994, pp. 1202-1209.
28. M. S. Arefeen, M. Ehsani and T. A. Lipo, "Sensorless Position Measurement in Synchronous Reluctance Motor," *IEEE Trans. on Power Electronics*, Vol. 9, No. 6, November 1994, pp. 624-630.
29. T. S. Wu, I. Husain, and M. Ehsani, "Switched-Mode Converters for High-Power DC-DC Applications," *International Journal of Electronics*, Vol. 77, No. 5, November 1994, pp. 583-599.
30. M. C. Smit, J. A. Ferreira, J. D. Van Wyk, and M. Ehsani, "An Ultrasonic Series Resonant Converter with Integrated LCT," *IEEE Trans. on Power Electronics*, Vol. 10, No. 1, January 1995, pp. 25-31.
31. A. Ludbrook and M. Ehsani, "Burndown Prevention in Static Power Converter Equipment," *IEEE Industry Applications Magazine*, Vol. 1, No. 2, March/April 1995, pp. 46-53.
32. D. H. Jang, G. H. Choe, and M. Ehsani, "Asymmetrical PWM Technique with Harmonic Elimination and Power Factor Control in AC Choppers," *IEEE Trans. on Power Electronics*, Vol. 10, No. 2, March 1995, pp. 175-184.
33. M. Ehsani, M. O. Bilgic, S. Khan, L. Laskai, and S. G. Jeong, "Capacitor Coupled converter (C^3) for High Power DC Conversion," *IEEE Trans. on Power Electronics*, Vol. 10, No. 4, July 1995, pp. 511-518.
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35. M. Ehsani, M. O. Bilgic, A. D. Patton, and J. Mitra, "Magnetically Inflatable SPS with Energy Storage Capability," *IEEE Aerospace and Electronics Systems Magazine*, Vol. 10, No. 8, August 1995, pp. 9-14.
36. M. Ehsani, P. Le Polles, M. S. Arefeen, I. Pitel, and J. D. Van Wyk, "Computer Aided Design and Application of Integrated LC Filters," *IEEE Trans. on Power Electronics*, Vol. 11, No. 1, January 1996, pp. 182-190.
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39. S. Moore and M. Ehsani, "An Empirically Based Electrosource Horizon Lead-Acid Battery Model," *SAE Journal* SP-1156, No. 960448, February 1996.
40. R. C. Becerra, T. M. Jahns, and M. Ehsani, "Four-Quadrant Sensorless Brushless ECM Drives," cited by K. Rajashekara, et al., eds. *Sensorless Control of AC Motor Drives: Speed and Position Sensorless Operation*, IEEE Press, 1996, pp. 283-290.
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44. Y. Gao, K. Rahman, and M. Ehsani, "The Energy Flow Management and Battery Energy Capacity Determination of the Drive train of the Electrically Peaking Hybrid Vehicle," *SAE Journal Publication*, No. SP 1284, No. 972647, August 1997, pp. 43-48.
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46. J. Mahdavi, A. Emadi, M. D. Bellar, and M. Ehsani, "Analysis of Power Electronic Converters Using the Generalized State Space Averaging Approach," Invited Paper for Special Issue of IEEE Transactions on Circuits and Systems on Simulation, Theory and Design of Switched Analog Networks, *IEEE Trans. on Circuit s and Systems*, Vol. 44, No. 8, August 1997.
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48. M. Ehsani, J. Mahdavi, I. Pitel, J. E. Brandenburg, and F. E. Little, "Development of an Efficient Power Supply for the Microwave Electrothermal Thruster," *IEEE Aerospace and Electronic System Magazine*, Vol. 13, No. 5, May 1998, pp. 37-42.
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51. M. Ehsani, M. D. Bellar, K. M. Khwaja, and A. Severinsky, "Evaluation of Soft Switching Inverters for EV and HEV Motor Drives," *IEEE Trans. on Industrial Electronics*, Vol. 48, No. 1, Feb. 2001.
52. Z. Rahman, K. Butler, and M. Ehsani, "A Study of Design Issues on Electrically Peaking Hybrid Electric Vehicles for Diverse Urban Driving Patterns," *Advances in Electric*

- Vehicle Technologies*, SP-1417, Paper #: 1999-01-1151, Society of Automotive Engineers, March 1999, pp. 1-9.
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 58. B. Fahimi, G. Suresh, J. Mahdavi, and M. Ehsani, "A New Approach to Model Switched Reluctance Motor Drive: Application to Dynamic Performance Prediction, Control and Design," submitted to the *IEEE Trans. on Power Electronics*, 1999.
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 62. B. Fahimi, G. Suresh, K. M. Rahman, and M. Ehsani, "Mitigation of Acoustic Noise and Vibration Cancellation in Switched Reluctance Motor Drive Using Neural Network Based Current Profiling," submitted to the *IEEE Trans. on Industrial Electronics*.
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5. "Theoretical Analysis of the Dynamics of Inductor-Converter Bridge," Wisconsin Superconductive Energy Storage Project Report, Volume IV, September 1983.

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9. "Microcomputer Simulation of Electrified Drilling Rig Systems," Research Project Report, Center for Energy and Mineral Resources, Texas A&M University, May 1986.
10. "Power Electronics Research and Development for High Performance Aircraft," Research Project Report, General Dynamics, Fort Worth Division, December 1986.
11. "Force Commutated Cycloconverters for 400 Hz Generator Systems," Research Project Report, Wright Patterson Air Force Base, October 1986.
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13. "Optimized Power Conditioning Alternatives for Nuclear Spacecraft," General Electric, Valley Forge, November 1988.
14. "Conceptual Design of the Power Conditioning System for Thermoelectric and Dynamic Power Conversion in a Space Nuclear Power System," NASA Center for Space Power, Texas A&M University, December 1988.
15. "New Architectures for Superconductive and Normal Power Systems," US Interagency Advanced Power Group, Systems Working Group Meeting Report, April 1989.
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20. Co-author, "Combat Hybrid Power Systems Technologies, Technical Challenges and Research Priorities," a report of National Research Council of the National Academies, 2003.
21. Co-author, Motorola Corporation Research Visionary Board Final report, 2003.
22. Co-author, "Failure Modes and Effects Analysis of Series Hybrid Combat Vehicle" reported to US Army Research Lab, April 2004.
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SAMPLE LECTURES

1. "Power Electronics and Motor Drives," One-Week Short Course, Polytechnic Institute of Guayaquil, Equator, November 1986.
2. "A Force-Commutated Direct Frequency Changer for the Aircraft VSCF Systems," Invited Seminar, Wright Patterson Air Force Base, Aeropropulsion Laboratory, November 1986.
3. "A Solid State DC Breaker for the High Performance Aircraft Power System," Invited Seminar, General Dynamics, Fort Worth Division, December 1986.
4. "Application of the Matrix Converter to the Aircraft VSCF System," Invited Seminar, Sundstrand Corporation, Rockford, Illinois, February 1987.
5. "A Short Course in Power Electronics and Motor Drives," Fisher Controls International, Marshalltown, Iowa, July 1988.
6. "New Concepts in High Power DC-DC Converters, Electromechanical Activators and Pulsed Power Systems," Invited Seminar, Wright Patterson Air Force Base, March 1989.
7. "Power Electronics and Drives," A Short Course Offered by Electric Power Research Institute (EPRI) Power Electronics Applications Center, June 1989.
8. "Comparison of Various Energy Storage Schemes for Military Applications," Invited Half a Day Seminar at US Army Electronics Technology and Devices Laboratory, Fort Monmouth, NJ, January 1989.
9. "Pulse width Modulation (PWM) Techniques," Short Course at IEEE Applied Power Electronics Conference and Exposition (APEC'91), Dallas, March 1991.
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14. Numerous other IEEE Distinguished Lectures, for the Industry Applications and Industrial Electronics Societies, all over the US and worldwide.
15. Short Cycle Time Design of Advanced Motor Drives by the Real Time Simulation and Hardware in the Loop technologies, a two day short course offered to the automotive industry in Detroit, Michigan, Nov. 3-4, 2003.

16. "Integrated Biofuel-Vehicle System", Key Note address given at the opening of the 2004 IEEE-VTS Vehicle Power and Propulsion Symposium in Paris, France, October 4th, 2004

SIGNIFICANT ADDITIONAL INFORMATION

1. Founder of power electronics education and research program at Texas A&M University including a curriculum of two undergraduate and six graduate courses and three laboratories for teaching and research: presently the Power Electronics Group at Texas A&M consists of three faculty members and three visiting faculty members. The program enrolls over 100 undergraduate students annually and has had over 40 Ph.D. and 70 M.S. students up to the present.
2. Founder and director, Texas Applied Power Electronics Center TAPC in Electrical Engineering Department, 1982, funded by Texas Engineering Station, Texas A&M Office of University Research, Texas Higher Education Coordinating Board and industrial companies. (A Multidisciplinary center working on the ELPH hybrid vehicle project. Six professors, Texas Transportation Institute and ten industrial companies are collaborating on ELPH, under TAPC, at the present time.)
3. Founder of Advanced Vehicle Systems Research Program, College of Engineering, Texas A&M University, 1992, funded by Texas Engineering Experiment Station, Department of Electrical Engineering, and Texas Transportation Institute.
4. Technical expert for proposal reviews for the US National Science Foundation, Department of Energy, Environmental Protection Agency; Department of Commerce, Canadian National Science and Engineering Council; South African Foundation for Research and Development.
5. World Bank expert for academic program development in Polytechnic Institute of Guayaquil, Equator.
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8. IEEE Standards book reviewer for Power System Harmonics (IEEE Std. 519-199 X) and for the revision of Standards for Practices and Requirements for General Purpose DC Drives.
9. Reviewer for journals and conferences of IEEE Industry Applications Society, Power Electronics Society, Industrial Electronics Society, Control Systems Society, IEEE Proceedings and a few others.

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12. Guadalupe Gonzalez	Technological University of Panama (UTP)



US005713317A

United States Patent [19]
Yoshioka

[11] Patent Number: 5,713,317
[45] Date of Patent: Feb. 3, 1998

[54] METHOD OF AND APPARATUS FOR
CONTINUOUSLY AND VARIABLY
CONTROLLING VALVE TIMING OF
INTERNAL COMBUSTION ENGINE

FOREIGN PATENT DOCUMENTS

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3-99844 10/1991 Japan .
4-175430 6/1992 Japan .

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Primary Examiner—Weilun Lo
Attorney, Agent, or Firm—Kenyon & Kenyon

[21] Appl. No.: 684,592

[22] Filed: Jul. 15, 1996

[30] Foreign Application Priority Data

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[51] Int. Cl.⁶ F02D 13/02; F01L 1/34

[52] U.S. Cl. 123/90.15; 123/90.17

[58] Field of Search 123/90.15, 90.16,
123/90.17, 90.18, 90.31

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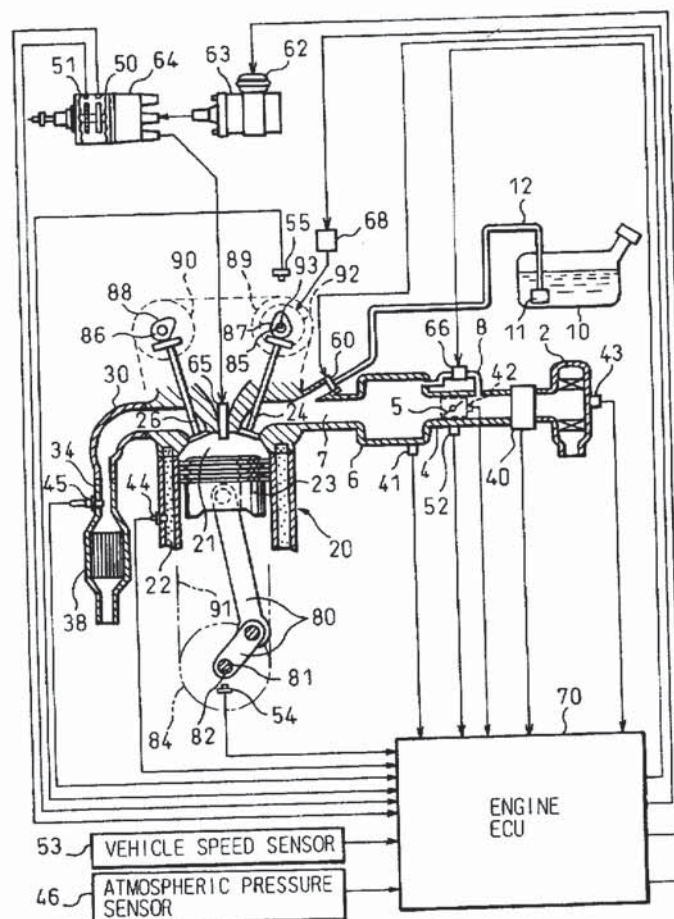
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[57] ABSTRACT

A method or an apparatus controls a continuously variable valve timing mechanism of an internal combustion engine to continuously and variably control the timing of an intake valve according to atmospheric pressure. The method or apparatus reads an engine revolution speed NE, an engine load GN, and an atmospheric pressure PA, refers to a map according to the atmospheric pressure PA, to find an atmospheric pressure correction coefficient Kpa, which is 1.0 at low altitudes with PA=760 mmHg and decreases as the altitude increases, i.e., as the atmospheric pressure PA decreases, calculates a corrected engine load GNpa as $GNpa = GN/Kpa$, and refers to a map according to the corrected engine load GNpa and engine revolution speed NE, to find a target displacement to be imposed on the open/close timing of the intake valve.

6 Claims, 15 Drawing Sheets



VW EX1005
U.S. Patent No. 6,557,540

Fig. 1

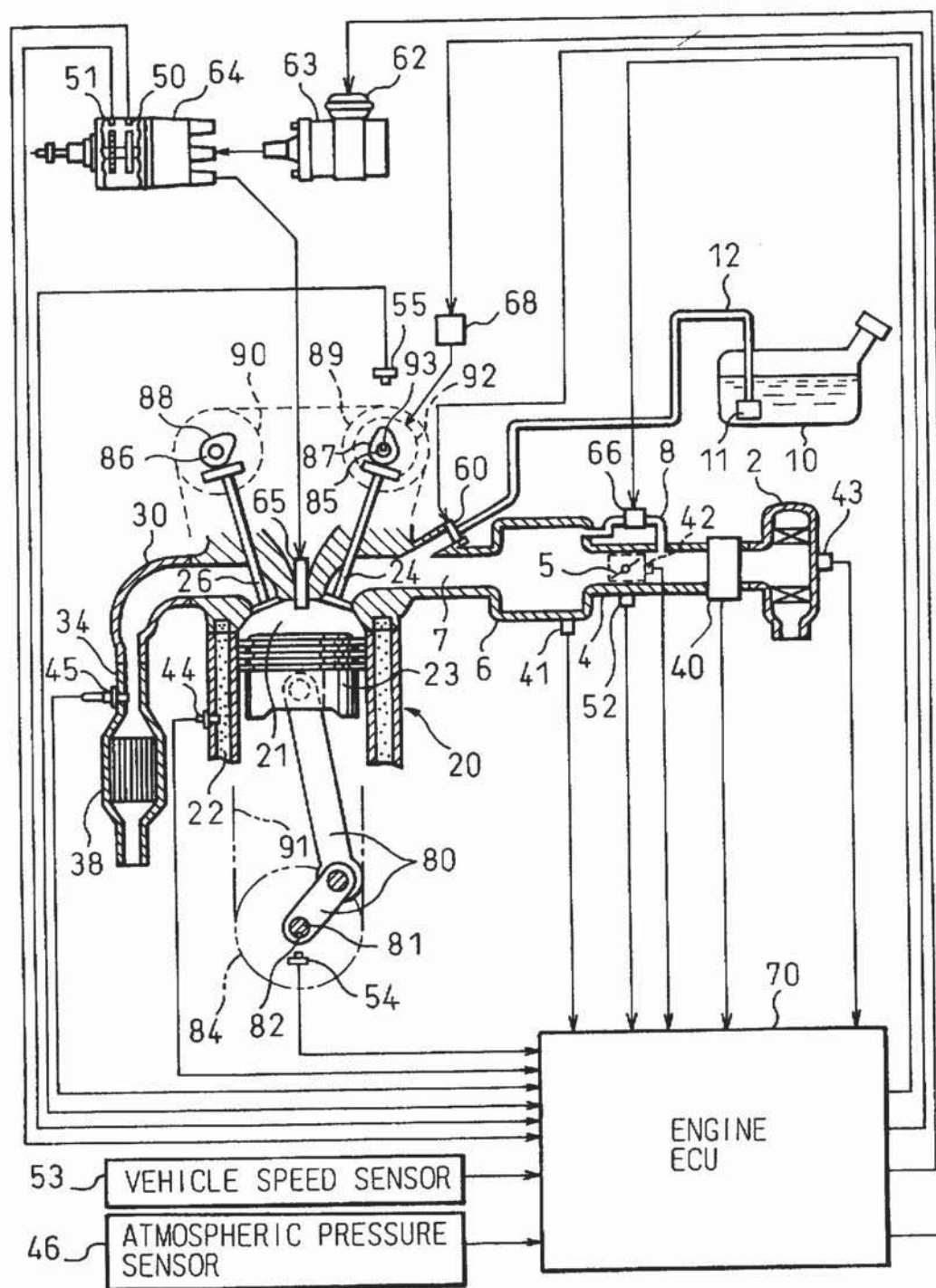


Fig.2

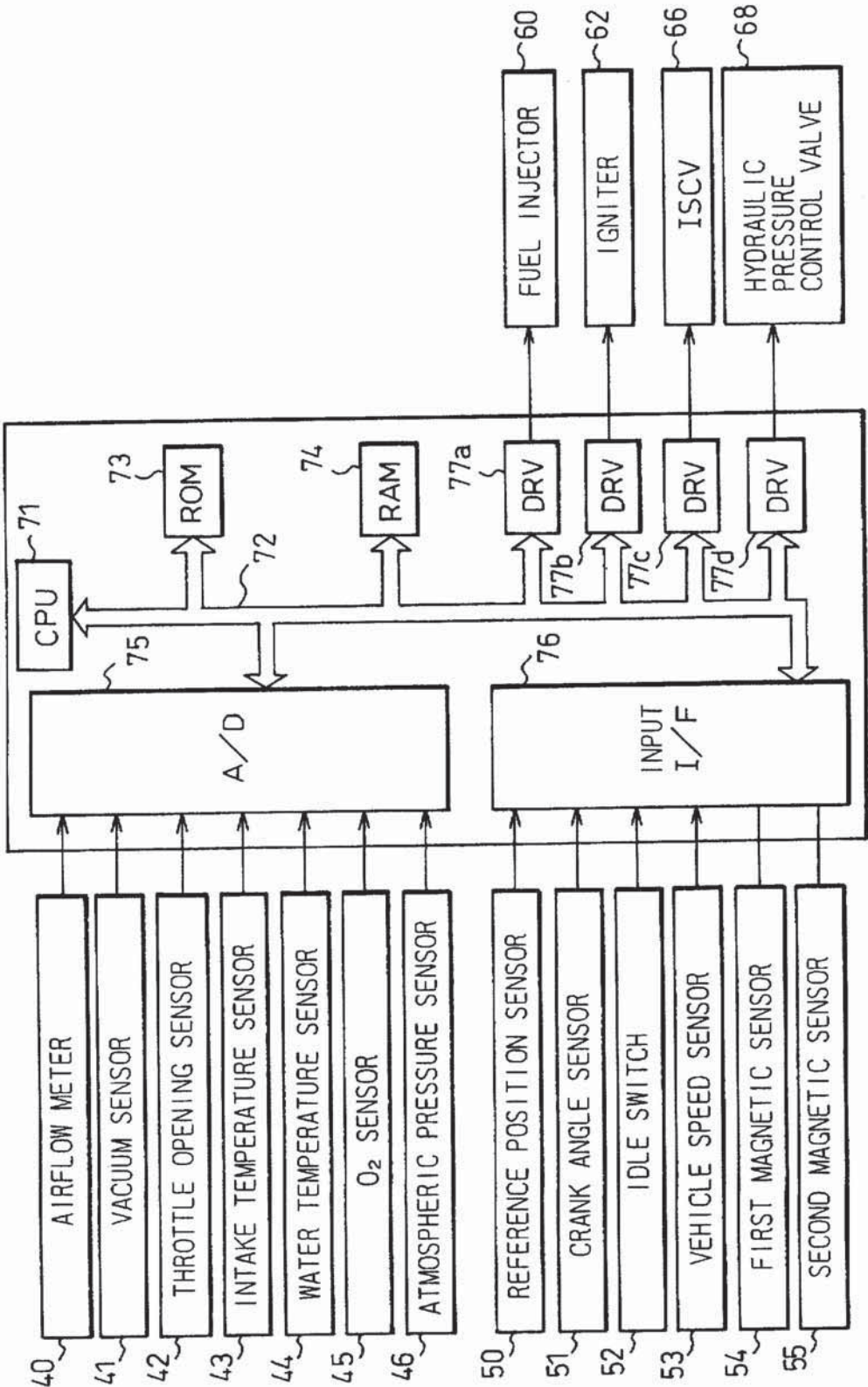


Fig. 3

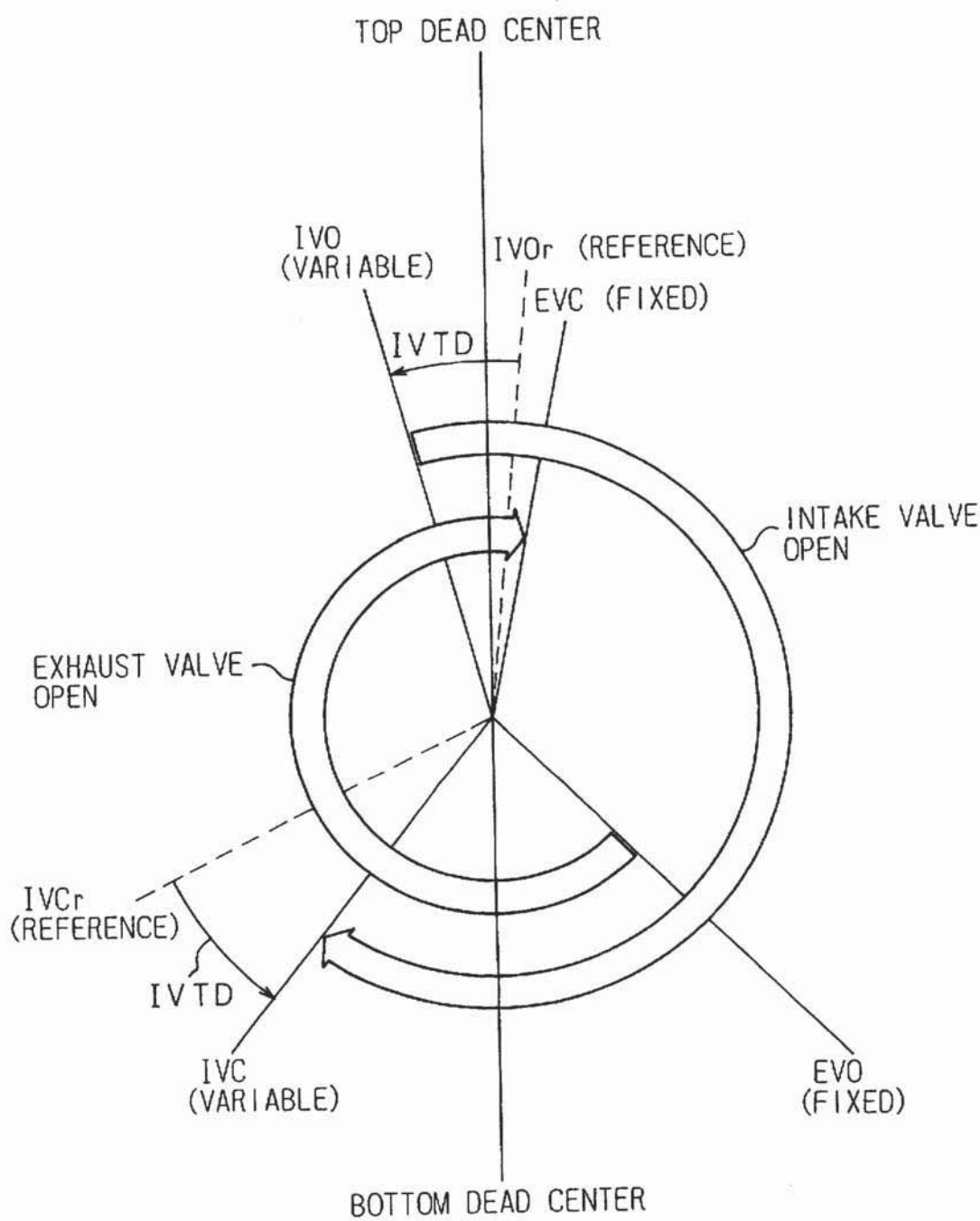


Fig. 4

		ENGINE REVOLUTION SPEED NE [rpm]						
ENGINE LOAD GN [g/rev.]		800	1200	1600	2000	~	6000	6400
	0.2	0	0	0	0	~	0	0
	0.4	0	5	10	10	~	5	5
	0.6	5	20	30	30	~	10	10
	0.8	20	40	50	50	~	20	20
	1.0	35	55	60	60	~	20	20
	1.2	40	60	60	60	~	10	10
	1.4	30	55	60	60	~	5	5
	1.6	30	50	50	50	~	5	5
	1.8	30	50	40	40	~	5	0
	2.0	30	50	40	40	~	5	0

Fig. 5

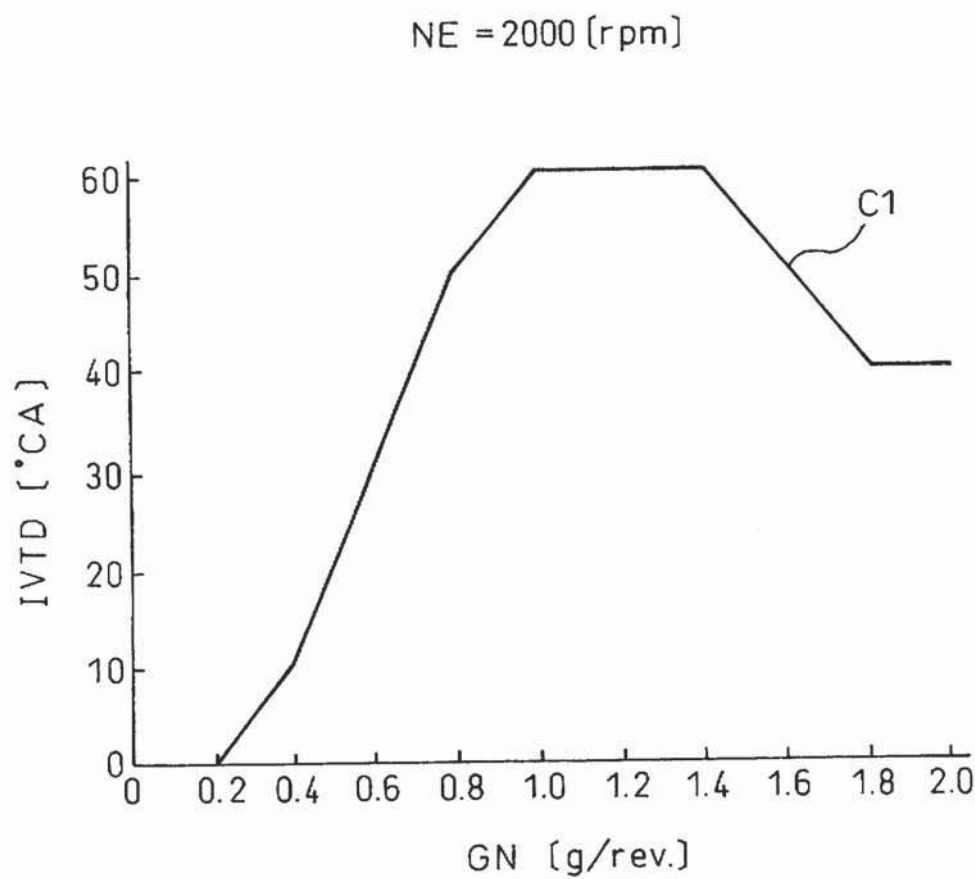


Fig. 6

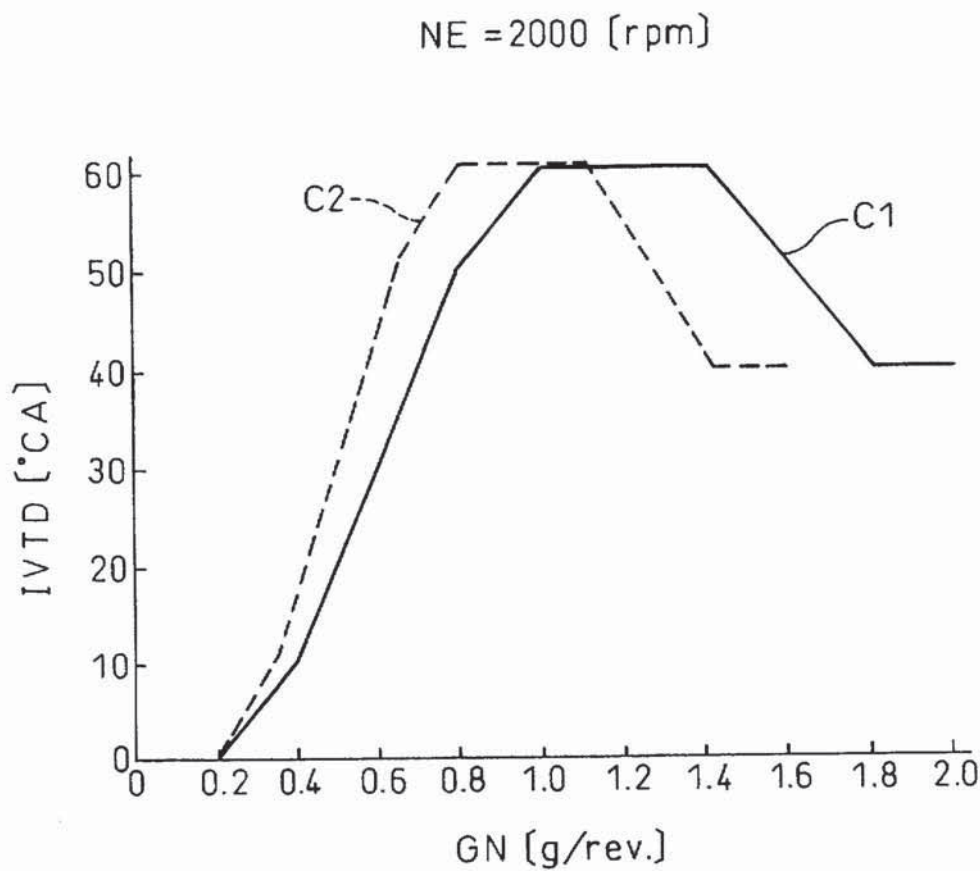


Fig. 7

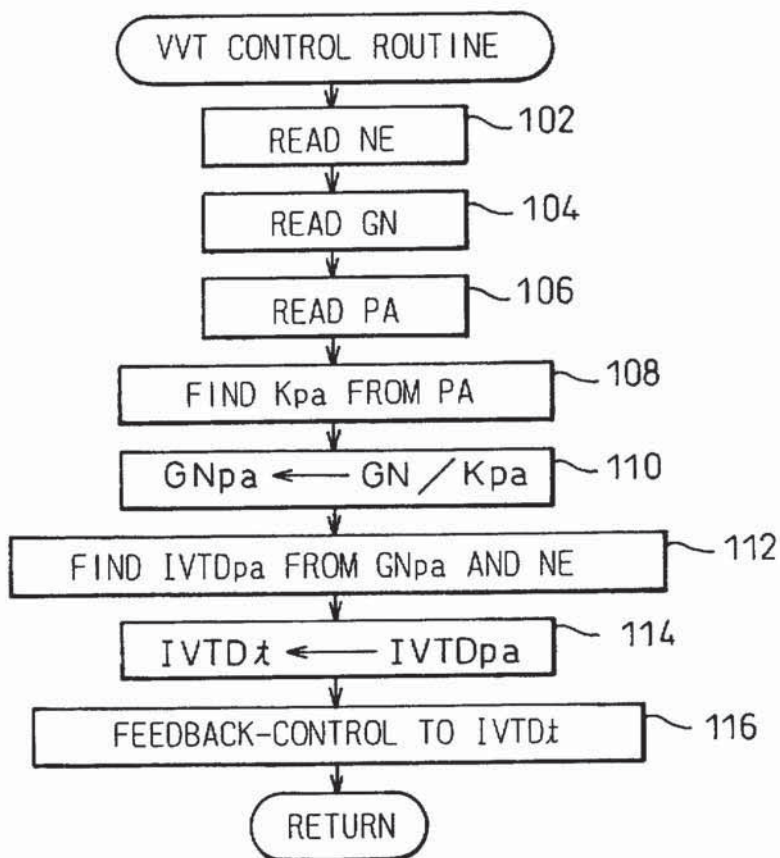


Fig. 8

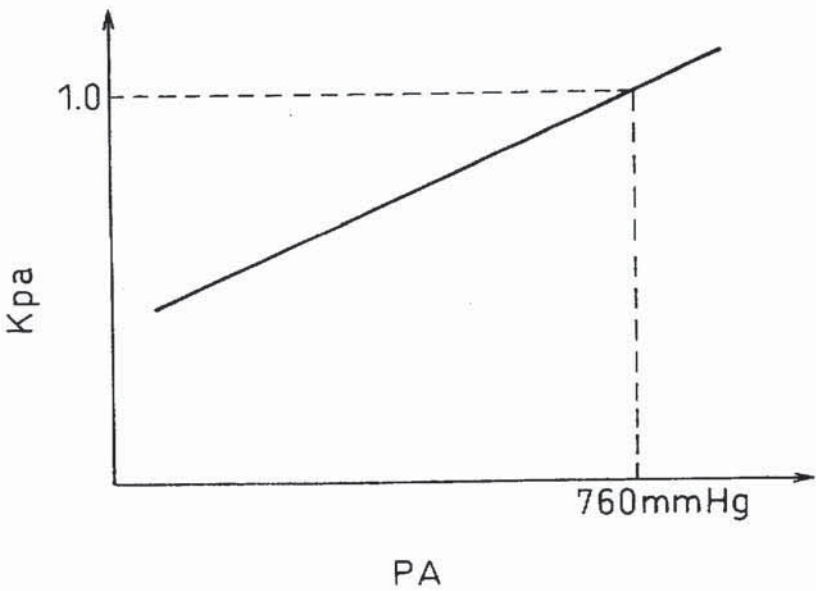


Fig. 9

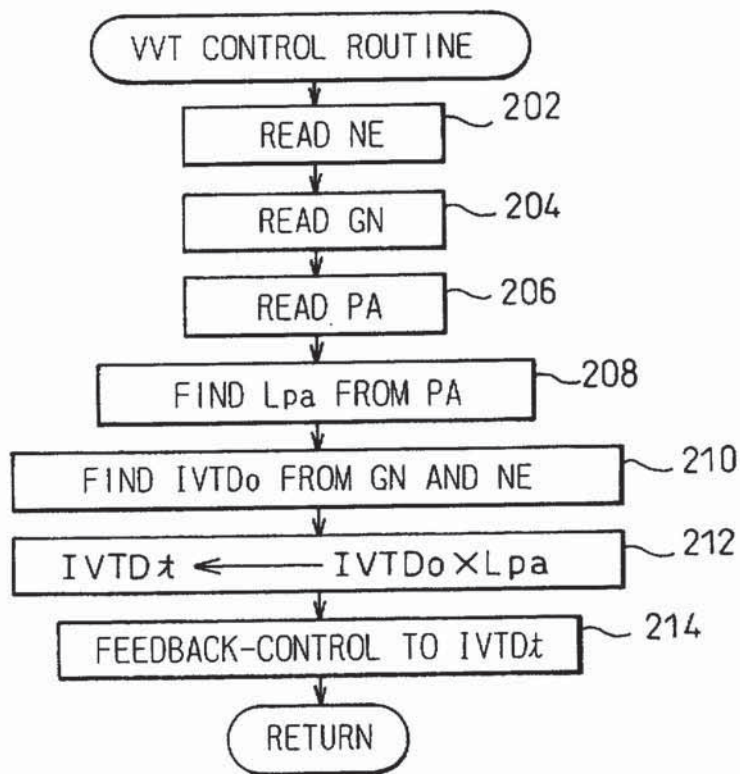


Fig. 10

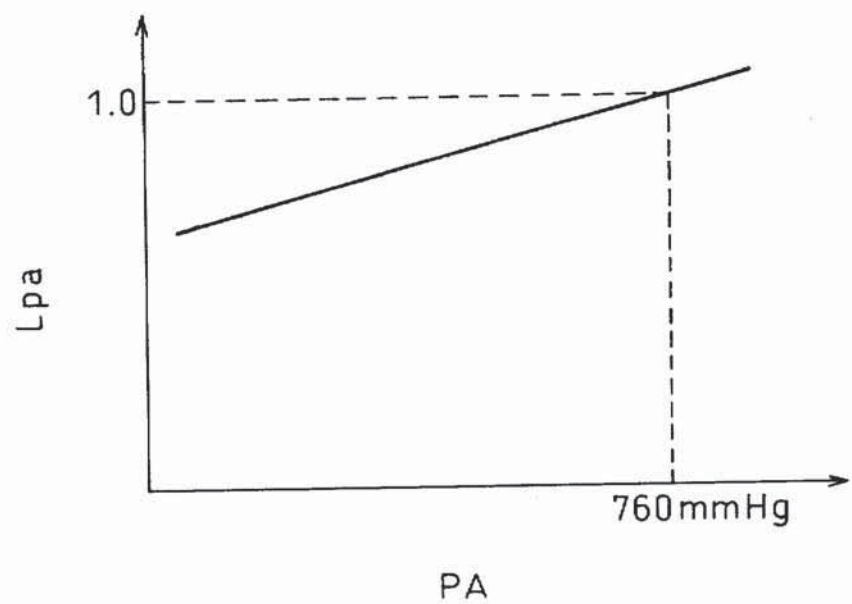


Fig.11

NE = 2000 [rpm]

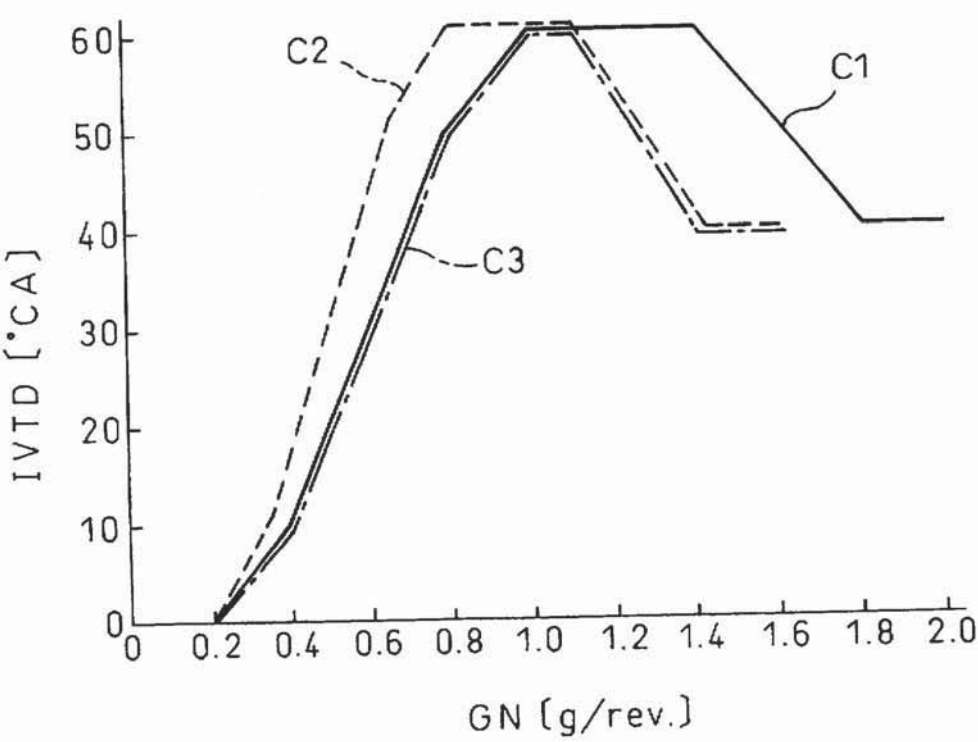


Fig. 12

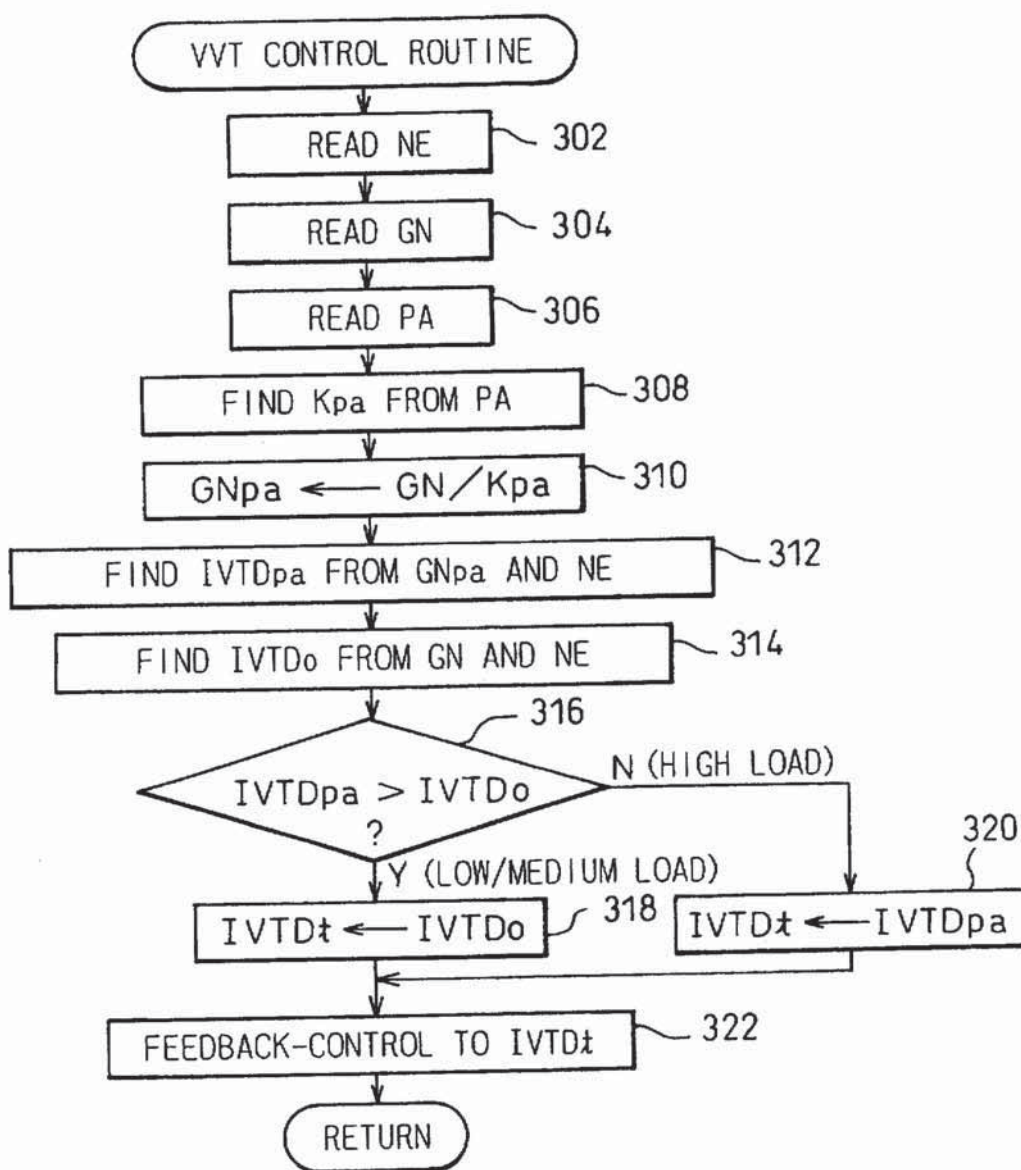


Fig.13

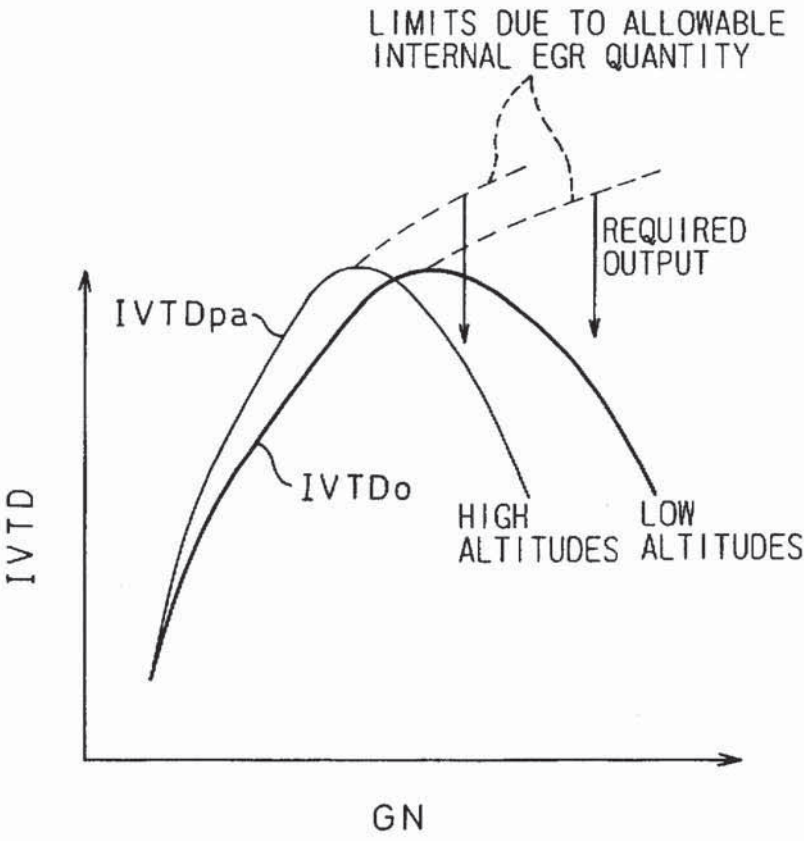


Fig.14

NE = 2000 [rpm]

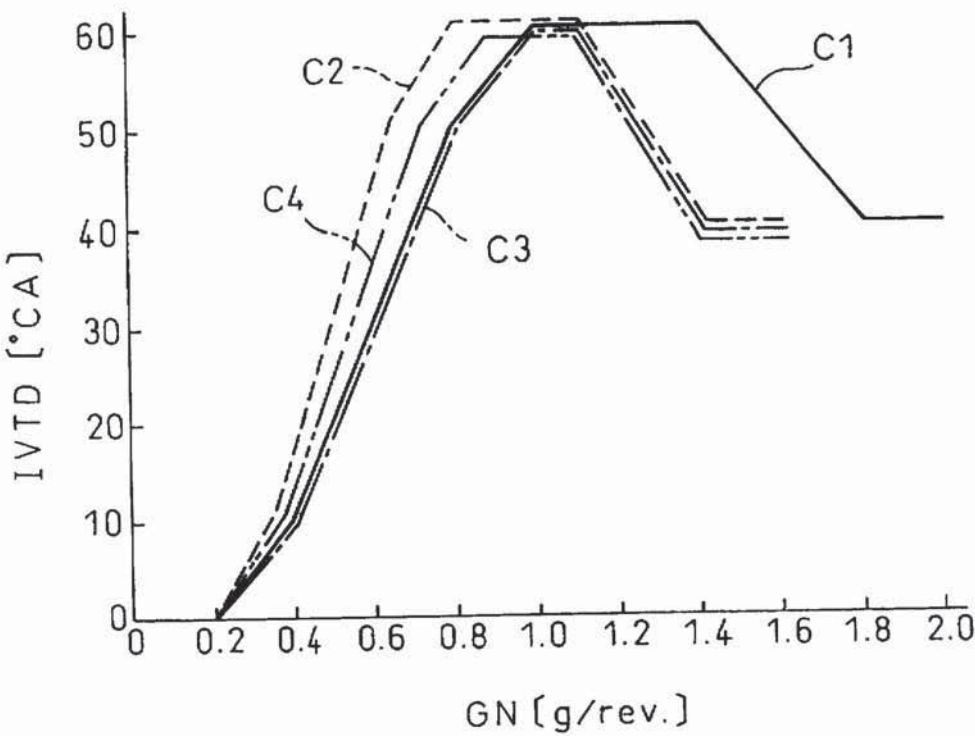
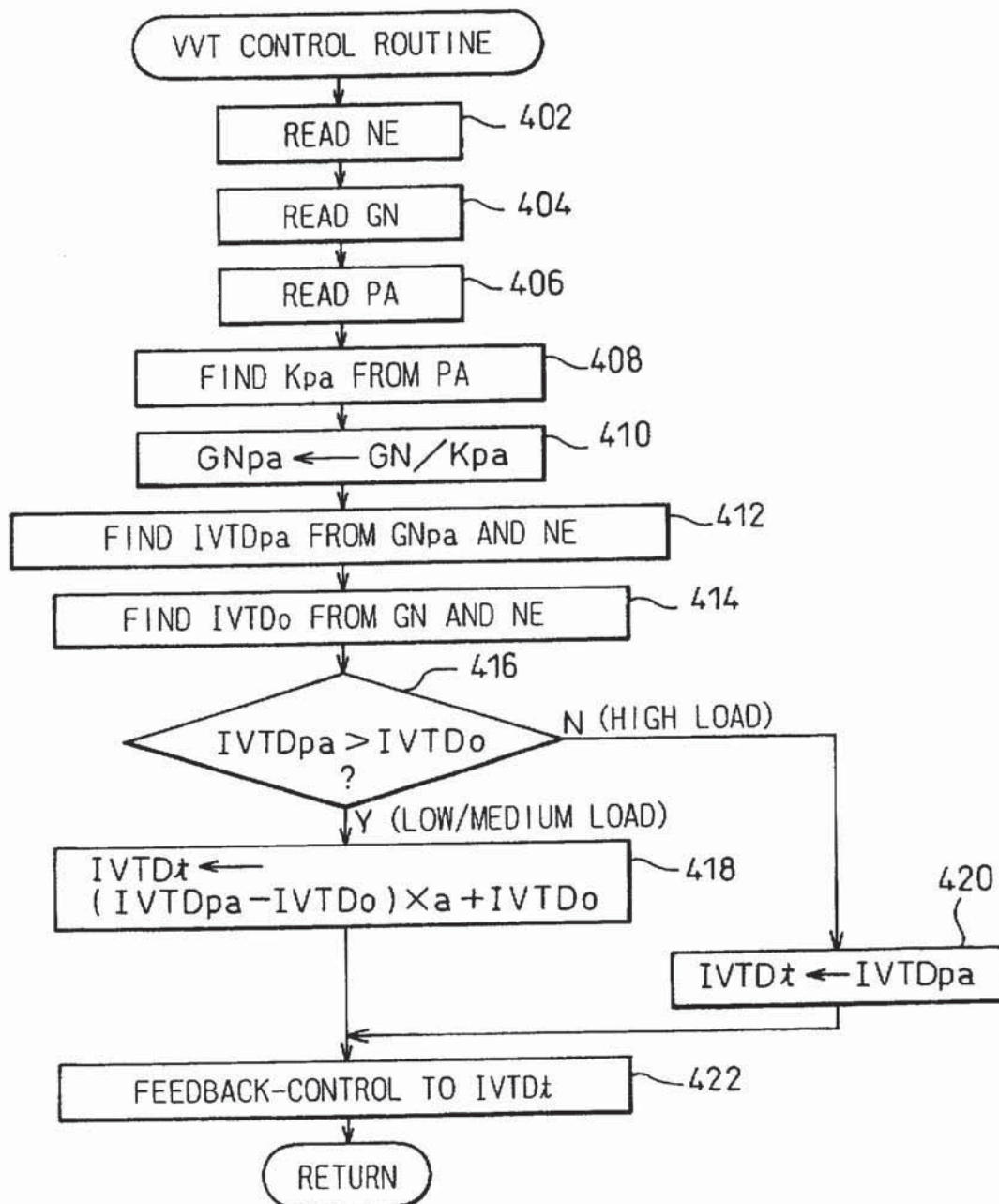


Fig. 15



1

METHOD OF AND APPARATUS FOR CONTINUOUSLY AND VARIABLY CONTROLLING VALVE TIMING OF INTERNAL COMBUSTION ENGINE

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a method of and an apparatus for controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of an intake or exhaust valve of the engine according to a target timing.

2. Description of the Related Art

There are many variable mechanisms that optimize the open/close timing of a valve of an automobile engine according to driving conditions. One of them is an ON/OFF mechanism, which changes the timing of an intake valve of the engine from normal timing to early timing when an engine load exceeds a switching threshold. Japanese Unexamined Utility Model Publication No. 3-99844 decreases the switching threshold or increases a detected engine load in response to a decrease in atmospheric pressure, to optimize the valve timing.

To provide high-performance engines, continuously variable valve timing mechanisms for optimizing valve timing at all times have been developed in place of the ON/OFF mechanisms, as disclosed in Japanese Unexamined Patent Publication No. 4-175430 (corresponding to U.S. Pat. No. 5,271,360). This mechanism continuously changes the open/close timing of a valve, and therefore, never employs the idea of changing the switching threshold in response to a change in atmospheric pressure. There is a requirement, therefore, to provide a method of properly controlling the continuously variable valve timing mechanism.

SUMMARY OF THE INVENTION

An object of the present invention is to provide a method of and an apparatus for optimizing valve timing according to atmospheric pressure for a continuously variable valve timing mechanism of an internal combustion engine.

The present invention intends to improve the output of an engine at high altitudes under high load, as well as reducing fuel consumption and improving exhaust purifying performance at high altitudes under low/medium load. In order to accomplish the objects, the present invention provides the following technical arrangements:

A first aspect of the present invention provides a method of controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of at least one of the intake and the exhaust valves of the engine, including the steps of (a) detecting an engine load, (b) detecting an atmospheric pressure, (c) correcting the detected engine load such that the lower the detected atmospheric pressure the more the engine load is increased, and (d) setting a target valve timing for the valve timing mechanism according to the corrected engine load.

The first aspect also provides an apparatus for controlling valve timing of an internal combustion engine, having a valve timing mechanism for continuously and variably controlling the open/close timing of at least one of the intake and the exhaust valves of the engine, a load detector for detecting an engine load, a pressure detector for detecting an atmospheric pressure, a load corrector for correcting the

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detected engine load such that the lower the detected atmospheric pressure the more the engine load is increased, and a timing setter for setting target a valve timing for the valve timing mechanism according to the corrected engine load.

A second aspect of the present invention provides a method of controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control the open/close timing of at least one of intake and exhaust valves of the engine, including the steps of (a) detecting an engine load, (b) detecting an atmospheric pressure, (c) setting, according to the detected engine load, a displacement from a reference timing as a target valve timing for the valve timing mechanism, and (d) correcting the set displacement such that the lower the detected atmospheric pressure the more the displacement is reduced.

The second aspect also provides an apparatus for controlling valve timing of an internal combustion engine, having a valve timing mechanism for continuously and variably controlling the open/close timing of at least one of intake and exhaust valves of the engine, a load detector for detecting an engine load, a pressure detector for detecting an atmospheric pressure, a displacement setter for setting, according to the detected engine load, a displacement from a reference timing as target valve timing for the valve timing mechanism, and a displacement corrector for correcting the set displacement such that the lower the detected atmospheric pressure the more the set displacement is reduced.

When driving at high altitudes, an engine receives a smaller quantity of intake air due to the low concentration of air. The first and second aspects of the present invention are capable of optimizing valve timing to maintain the output of the engine at high altitudes.

A third aspect of the present invention provides a method including the step of, in addition to the steps of the first aspect, (e) prohibiting the correction in step (c) if the detected engine load is in a low/medium range.

The third aspect also provides an apparatus having, in addition to the components of the first aspect, a prohibition unit for prohibiting the correction carried out by the load corrector if the detected engine load is in a low/medium range.

The first aspect corrects an engine load in any range according to atmospheric pressure to maintain the output of the engine when driving at high altitudes. If such a correction is carried out under a low/medium engine load, the quantity of internally recirculated exhaust gas of an internal exhaust gas recirculation (EGR) system will increase excessively to cause incomplete combustion. The internal EGR system is employed to improve exhaust gas purifying performance and reduce pumping loss and fuel consumption. Accordingly, the third aspect prohibits the engine load correction under a low/medium engine load, thereby preventing an excessive amount of exhaust gas being recirculated.

A fourth aspect of the present invention provides a method including the step of, in addition to the steps of the first aspect, (e) correcting the target valve timing set in step (d) to an intermediate value between the target valve timing and valve timing determined according to the detected engine load, if the detected engine load is in a low/medium range.

The fourth aspect also provides an apparatus having, in addition to the components of the first aspect, a timing corrector for correcting the target valve timing set by the timing setter to an intermediate value between the target

valve timing and valve timing determined according to the engine load detected by the load detector, if the detected engine load is in a low/medium range.

The third aspect simply prohibits the engine load correction according to atmospheric pressure if the engine load is in a low/medium range. When driving at high altitudes, the negative pressure of an intake duct of the engine decreases to reduce the quantity of internally recirculated exhaust gas. Accordingly, the fourth aspect corrects the engine load according to atmospheric pressure, to optimize the quantity of recirculated internal exhaust gas at high altitudes under the low/medium engine load.

BRIEF DESCRIPTION OF THE DRAWINGS

Further features and advantages of the present invention will be apparent from the following description with reference to the accompanying drawings, in which:

FIG. 1 is a general view showing an electronically controlled internal combustion engine having an apparatus for continuously and variably controlling valve timing according to an embodiment of the present invention;

FIG. 2 is a block diagram showing an electronic engine control unit according to the embodiment;

FIG. 3 is a timing chart showing crank angles representing the open/close timing of intake and exhaust valves;

FIG. 4 is a map used to determine a displacement to be imposed on the open/close timing of an intake valve according to the revolution speed and load of an engine (the load being represented by the mass of intake air per revolution);

FIG. 5 is a graph showing the relationship between an engine load and a target displacement to be imposed on the open/close timing of an intake valve at an engine speed (NE) of 2000 rpm;

FIG. 6 is a graph explaining a correction of an engine load according to atmospheric pressure according to a first embodiment of the present invention;

FIG. 7 is a flowchart showing a routine of variably controlling the open/close timing of a valve according to the first embodiment;

FIG. 8 is a map used to determine an atmospheric pressure correction coefficient Kpa according to an engine load GN;

FIG. 9 is a flowchart showing a routine of variably controlling the open/close timing of a valve according to a second embodiment of the present invention;

FIG. 10 is a map used to determine an atmospheric pressure correction coefficient Lpa according to a displacement to be imposed on the open/close timing of an intake valve;

FIG. 11 is a graph explaining a correction of an engine load according to atmospheric pressure according to a third embodiment of the present invention;

FIG. 12 is a flowchart showing a routine of variably controlling the open/close timing of a valve according to the third embodiment;

FIG. 13 is a graph showing the relationship between an engine load and a displacement to be imposed on the open/close timing of an intake valve at low altitudes (IVTD0) and high altitudes (IVTDpa);

FIG. 14 is a graph explaining a correction of an engine load according to atmospheric pressure according to a fourth embodiment of the present invention; and

FIG. 15 is a flowchart showing a routine of variably controlling the open/close timing of a valve according to the fourth embodiment.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

The embodiments of the present invention will be explained with reference to the drawings.

FIG. 1 is a general view showing an electronically controlled internal combustion engine having an apparatus for continuously and variably controlling the open/close timing of a valve according to an embodiment of the present invention. An air cleaner 2 filters air necessary for combustion, a throttle body 4 passes the air, and a surge tank (intake manifold) 6 distributes the air to an intake duct 7 of each cylinder. A throttle valve 5 arranged in the throttle body 4 adjusts the quantity of intake air. An airflow meter 40 measures the mass flow rate of intake air. A temperature sensor 43 detects the temperature of intake air. A vacuum sensor 41 detects the pressure of the intake duct 7. An atmospheric pressure sensor 46 is arranged so that it is not influenced by wind pressure during driving.

A throttle sensor 42 detects the opening of the throttle valve 5. When the throttle valve 5 is completely closed, an idle switch 52 is turned on to generate a throttle closed signal. An idling adjusting path 8 bypasses the throttle valve 5 and has an idling speed control valve (ISCV) 66 for adjusting an airflow rate during an idling period.

Fuel in a fuel tank 10 is pumped up by a fuel pump 11, is passed through a fuel pipe 12, and is injected by a fuel injector 60.

The intake duct 7 mixes fuel with air, and the air-fuel mixture is drawn into a combustion chamber 21 of a cylinder 20 through an intake valve 24. The air-fuel mixture is compressed by a piston 23, ignited, exploded, and combusted to produce power. Namely, an igniter 62 generates an ignition signal to control a primary current of an ignition coil 63. A secondary current of the ignition coil 63 is supplied to a spark plug 65 through an ignition distributor 64, to thereby ignite the air-fuel mixture in the combustion chamber 21.

The ignition distributor 64 has a reference position sensor 50 and a crank angle sensor 51. The reference position sensor 50 generates a reference position pulse for every 720 degrees of crank angle. The crank angle sensor 51 generates a position pulse for every 30 degrees of crank angle. A vehicle speed sensor 53 generates a pulse representing an actual running speed. A cooling water path 22 guides cooling water to cool the engine 20. A water temperature sensor 44 detects the temperature of the cooling water.

An exhaust valve 26 discharges an exhaust gas of the combusted air-fuel mixture into an exhaust manifold 30 connected to an exhaust pipe 34. The exhaust pipe 34 has an O₂ sensor 45 to detect the concentration of oxygen in the exhaust gas. A catalytic converter 38 is arranged in the exhaust pipe 34 downstream from the sensor 45. The catalytic converter 38 contains a three-way catalyst for promoting the oxidization of unburned components such as HC and CO as well as the reduction of nitrogen oxides (NO_x). The purified exhaust gas from the catalytic converter 38 is discharged outside.

A mechanism for opening and closing the intake valve 24 and exhaust valve 26 will be explained. The piston 23 is connected to a crankshaft 81 through a connecting rod 80. An end of the crankshaft 81 has a timing pulley 84. The intake valve 24 is driven by a cam 87 attached to a cam shaft 85. The exhaust valve 26 is driven by a cam 88 attached to a cam shaft 86. An end of the cam shaft 85 has a timing pulley 89. An end of the cam shaft 86 has a timing pulley 90. The timing pulleys 89 and 90 are connected to the timing

pulley 84 through a timing belt 91. As a result, the crankshaft 81 drives the cam shafts 85 and 86 to open and close the intake valve 24 and exhaust valve 26 at predetermined crank angles. The crank shaft 81 has a buried magnet 82, so that a first magnetic sensor 54 arranged close to the crankshaft 81 may generate a reference pulse. The cam shaft 85 of the intake valve 24 has a buried magnet 93, so that a second magnetic sensor 55 arranged close to the cam shaft 85 may generate a reference pulse.

A continuously variable mechanism 92 having a known structure to determine the open/close timing of the intake valve 24 is arranged between the cam shaft 85 and the timing pulley 89. The mechanism 92 turns the cam shaft 85 and timing pulley 89 relative to each other. More precisely, the mechanism 92 uses the cam shaft 85 and timing pulley 89 as external gears and connects them to each other through an intermediate helical gear. The helical gear is axially moved by hydraulic pressure, to drive the cam shaft 85 and timing pulley 89 relative to each other. The hydraulic pressure is controlled by a control valve 68.

An electronic engine control unit (ECU) 70 is a micro-computer to control fuel injection, ignition timing, idling speed, and in addition, the open/close timing of a valve according to the present invention. FIG. 2 is a block diagram showing the hardware of the control unit 70. A read-only memory (ROM) 73 stores programs and maps. A central processing unit (CPU) 71 receives signals from the sensors and switches through an A/D converter 75 and an interface 76, processes the signals, and provides actuator control signals through drive control circuits 77a to 77d. A random-access memory (RAM) 74 temporarily stores data during the operation of the CPU 71. These elements are connected to one another through a system bus 72 containing address, data, and control buses.

An engine control process carried out by the control unit 70 will be explained.

Fuel injection control refers to the mass of intake air per engine revolution, calculates the fuel injection quantity, i.e., injection period of the fuel injector 60 to achieve a target air-fuel ratio, and instructs the drive control circuit 77a to inject fuel from the fuel injector 60 at a given crank angle. The mass of intake air per engine revolution is calculated according to the mass flow rate of intake air measured by the airflow meter 40 and an engine revolution speed detected by the crank angle sensor 51. Alternatively, it is estimated according to the pressure of the intake duct 7 detected by the vacuum sensor 41 and the engine revolution speed. The fuel injection quantity is subjected to corrections such as a basic correction according to signals from the throttle sensor 42, intake temperature sensor 43, water temperature sensor 44, etc., an air-fuel ratio feedback correction according to a signal from the O₂ sensor 45, and an air-fuel ratio learning correction to equalize a central feedback correction value with a theoretical air-fuel ratio.

Ignition timing control collectively checks engine conditions according to an engine revolution speed detected by the crank angle sensor 51 and signals from other sensors, calculates optimum ignition timing, and provides an ignition signal to the igniter 62 through the drive control circuit 77b.

Idling speed control detects an idling state according to a throttle closed signal from the idle switch 52 and a vehicle speed signal from the vehicle speed sensor 53, calculates a target revolution speed according to the temperature of cooling water measured by the water temperature sensor 44, compares the target speed with an actual revolution speed, determines a control quantity to attain the target revolution

speed according to the difference between the target and actual revolution speeds, and controls the idling speed control valve 66 through the drive control circuit 77c, to thereby adjust the quantity of air and optimize the idling speed.

Valve timing control sets the target open/close timing of the intake valve 24 according to operating conditions and controls the continuously variable mechanism 92. More precisely, the hydraulic pressure control valve 68 is feedback-controlled according to signals from the first and second magnetic sensors 54 and 55 so that the cam shaft 85 of the intake valve 24 maintains a required rotational phase with respect to the crankshaft 81. The valve timing control according to the present invention will be explained in detail.

FIG. 3 is a timing chart showing crank angles corresponding to the open/close timing of the intake and exhaust valves 24 and 26. The exhaust valve 26 is opened at fixed valve open timing EVO, for example, 50 degrees before a bottom exhaust dead center and is closed at fixed valve close timing EVC, for example, 3 degrees after a top exhaust dead center. On the other hand, the intake valve 24 involves a fixed open period and variable valve opening timing IVO and valve close timing IVC. The most delayed timing IVOr and IVCr of the intake valve 24 serve as reference positions. The open/close timing of the intake valve 24 is optionally set at positions ahead of the reference positions. A valve timing displacement IVTD serves as a control target quantity. The reference valve open timing IVOr is, for example, 3 degrees after the top exhaust dead center, and the reference valve close timing IVCr is, for example, 65 degrees after a bottom intake dead center. If the displacement IVTD is equal to a crank angle of 30 degrees, the valve open timing IVO is 27 degrees before the top exhaust dead center, and the valve close timing IVC is 35 degrees after the bottom intake dead center.

FIG. 4 is a map used to determine a displacement IVTD for optimizing the open/close timing of the intake valve 24. The displacement IVTD is determined according to an engine revolution speed NE and an engine load GN, i.e., the mass of intake air per engine revolution.

FIG. 5 is a graph showing the relationship between an engine load GN and a displacement IVTD with NE=2000 rpm. In a low/medium range (GN<1.2 g/rev), a valve overlapping period in which the intake and exhaust valves are both open is increased to increase the quantity of internally recirculated exhaust gas, to thereby improve exhaust gas purifying performance and decrease a pumping loss to reduce fuel consumption. To achieve this, the displacement IVTD is gradually increased. In a high range (GN>1.2 g/rev), the displacement IVTD is adjusted to improve the output of the engine. When GN=2.0 g/rev corresponding to a full throttle condition, the displacement IVTD is set to a crank angle of 40 degrees.

Values of FIGS. 4 and 5 are for low altitudes. At high altitudes, air concentration is low to reduce the quantity of intake air. When driving at low altitudes with NE=2000 rpm and if the throttle valve 5 is fully opened, the quantity of intake air is 2.0 g/rev as shown in FIG. 5. Namely, if GN=2.0 g/rev, it is determined that the throttle valve 5 is fully opened and that high output is required. To meet the requirement, the displacement IVTD is set to 40 degrees in crank angle. At high altitudes, the quantity of intake air is, for example, 1.6 g/rev even if the throttle valve 5 is fully opened. If the displacement IVTD is calculated normally under this situation, it will be 50 degrees in crank angle as shown in

FIG. 5. This displacement will never provide high output. It is preferable to set IVTD=40 degrees in crank angle at high altitudes if the throttle valve 5 is fully opened. High-altitude corrections of atmospheric pressure and air concentration will be explained according to four embodiments of the present invention.

The first embodiment according to the first aspect of the present invention will be explained. The first embodiment increases an engine load as a detected atmospheric pressure becomes lower, to maintain the output of the engine at high altitudes under a high load. A curve C1 of FIG. 6, which is equal to the curve C1 of FIG. 5, is proper when driving at low altitudes. Atmospheric pressure is low at high altitudes, and therefore, the present invention employs a curve C2 of FIG. 6 to correct an engine load GN.

FIG. 7 is a flowchart showing a variable valve timing (VVT) control routine according to the first embodiment. This routine is carried out regularly. Step 102 reads an engine revolution speed NE calculated according to the output of the crank angle sensor 51. Step 104 reads an engine load (the mass of intake air per engine revolution) calculated according to the mass flow rate of intake air detected by the airflow meter 40 and the engine revolution speed provided by the crank angle sensor 51. Step 106 reads an atmospheric pressure PA detected by the atmospheric pressure sensor 46.

Step 108 refers to a map of FIG. 8 according to the atmospheric pressure PA and finds an atmospheric pressure (air concentration) correction coefficient Kpa. The coefficient Kpa is 1.0 at low altitudes with PA=760 mmHg and decreases as the altitude increases, i.e., as the atmospheric pressure PA decreases. The map of FIG. 8 is stored in the ROM 73 in advance. Step 110 calculates an engine load GNpa after the atmospheric pressure correction as follows:

$$GNpa \leftarrow GN/Kpa$$

Step 112 refers to the map of FIG. 4 according to the engine load GNpa and engine revolution speed NE, to find a displacement IVTDpa to be imposed on the open/close timing of the intake valve 24. The map of FIG. 4 is also stored in the ROM 73 in advance.

Step 114 sets the displacement IVTDpa as a target displacement IVTDt. To bring an actual displacement to the target displacement IVTDt, signals from the first and second magnetic sensors 54 and 55 are monitored to feedback-control the hydraulic pressure control valve 68.

The second embodiment according to the second aspect of the present invention will be explained. The second embodiment sets a target displacement to be imposed on the open/close timing of the intake valve 24 according to a detected engine load and decreases the target displacement as an atmospheric pressure becomes lower, to maintain the output of the engine at high altitudes under high load, similar to the first embodiment. FIG. 9 is a flowchart showing a variable valve timing (WT) control routine according to the second embodiment. Steps 202 to 206 are the same as steps 102 to 106 of the first embodiment of FIG. 7.

Step 208 refers to a map of FIG. 10 according to the atmospheric pressure PA and finds a correction coefficient Lpa for correcting a displacement to be imposed on the open/close timing of the intake valve 24. The correction coefficient Lpa is 1.0 at low altitudes with PA=760 mmHg and decreases as the altitude increases, i.e., as the atmospheric pressure PA decreases. The map of FIG. 10 is stored in the ROM 73 in advance. Step 210 refers to the map of FIG. 4 according to the engine load GN and engine revolution speed NE and finds a displacement IVTD0 to be imposed on the open/close timing of the intake valve 24.

Step 212 corrects the displacement IVTD0 and provides a target displacement IVTDt as follows:

$$IVTDt \leftarrow IVTD0 \times Lpa$$

To bring the actual displacement of the intake valve 24 to the target displacement IVTDt, the hydraulic control valve 68 is feedback-controlled.

The third embodiment according to the third aspect of the present invention will be explained. The first embodiment corrects an engine load in any range according to atmospheric pressure. If this correction is made when the engine load is in a low/medium range, a valve overlapping period increases to excessively increase the quantity of internally recirculated exhaust gas, to deteriorate combustion. Accordingly, the third embodiment prohibits the engine load correction based on atmospheric pressure if the engine load is in the low/medium range. As indicated with a curve C3 of FIG. 11, the third embodiment largely corrects an engine load when the engine load is in a high range with atmospheric pressure being low. However, it carries out no correction if the engine load is in the low/medium range. Curves C1 and C2 of FIG. 11 correspond to the curves C1 and C2 of FIG. 6.

FIG. 12 is a flowchart showing a variable valve timing (VVT) control routine according to the third embodiment. Steps 302 to 312 are the same as steps 102 to 112 of the first embodiment of FIG. 7. Step 314 refers to the map of FIG. 4 according to the engine load GN and engine revolution speed NE and finds a displacement IVTD0 to be imposed on the open/close timing of the intake valve 24.

Step 316 compares the displacement IVTDpa of step 312 with the displacement IVTD0 of step 314 and determines whether the engine load is in a low/medium range or in a high range. The reason why this comparison can determine a load range will be explained. As a displacement to be imposed on the open/close timing of the intake valve 24 increases, the quantity of internally recirculated exhaust gas increases, and an allowable quantity of internally recirculated exhaust gas increases as the engine load increases. Accordingly, a maximum valve timing displacement is in the high load range in principle. However, if the maximum valve timing displacement is kept until the throttle valve 5 is fully opened, combustion will deteriorate to lower the output of the engine. Accordingly, the valve timing displacement is adjusted to realize a required engine output in the high load range. As a result, the displacement map has a maximum in the middle load range. This is the reason why the displacement IVTDpa corrected for high altitudes is compared with the displacement IVTD0 without correction, as shown in FIG. 13. Then, it is determined as follows:

if $IVTDpa > IVTD0$ then engine load is in low/medium range

if $IVTDpa \leq IVTD0$ then engine load is in high range

If it is determined that it is in the low/medium range, step 318 sets the displacement IVTD0 as a target displacement IVTDt. If it is determined that it is in the high range, step 320 sets the displacement IVTDpa as the target displacement IVTDt. Step 322 feedback-controls the hydraulic pressure control valve 68 according to the target displacement IVTDt.

The fourth embodiment according to the fourth aspect of the present invention will be explained. The third embodiment only prohibits the engine load correction according to atmospheric pressure if the engine load is in the low/medium range. The quantity of internally recirculated exhaust gas decreases at high altitudes due to a decrease in the negative pressure of the intake duct 7, and the fourth embodiment

considers this in correcting the engine load to optimize the quantity of internally recirculated exhaust gas in the low/medium range. As indicated with a curve C4 of FIG. 14, the engine load is corrected between the curve C2 of the first embodiment and the curve C3 of the third embodiment when the engine load is in the low/medium range. Curves C1, C2, and C3 of FIG. 14 correspond to the curves C1, C2, and C3 of FIG. 11.

FIG. 15 is a flowchart showing a variable valve timing (VVT) control routine according to the fourth embodiment. What is different from the third embodiment of FIG. 12 is only step 418 corresponding to step 318 of the third embodiment. Namely, step 418 sets a target displacement to be imposed on the open/close timing of the intake valve 24 when the engine load is in the low/medium range as follows:

$$IVTD_t = (IVTD_{pa} - IVTD_0) \times a + IVTD_0 \quad (a=0.5, \text{ for example})$$

The fourth embodiment optimizes the quantity of internally recirculated exhaust gas at high altitudes with the engine load being in the high or low/medium range, similar to driving at low altitudes.

Each of the above embodiments uses the two-dimensional map of engine load and engine revolution speed, to continuously control the open/close timing of the intake valve both at low and high highlands, thereby smoothing the torque characteristics with respect to the engine load and the throttle opening.

Although the present invention has been described by way of embodiments, it is apparent to those skilled in the art that other embodiments and modifications of the present invention are possible without departing from the spirit or essential features thereof. For example, the electronic engine control unit 70 used by each embodiment to variably control valve timing may be a dedicated electronic control unit. Although the embodiments variably control the open/close timing of the intake valve 24, the present invention is naturally applicable to control the exhaust valve 26, too. An engine load may be represented with the pressure of the intake duct 7.

As explained above, the present invention provides a method of and apparatus for setting optimum valve timing according to atmospheric pressure for a continuous variable valve timing mechanism of an internal combustion engine. The first and second aspects of the present invention are capable of setting optimum valve timing to maintain the output of the engine at high altitudes. The first aspect corrects an engine load in any range according to atmospheric pressure to maintain the output of the engine. If such a correction is carried out under a low/medium engine load, the quantity of internally recirculated exhaust gas of the internal EGR system will increase excessively to cause incomplete combustion. The internal EGR system is employed to improve exhaust gas purifying performance and reduce a pumping loss and fuel consumption. Accordingly, the third aspect of the present invention prohibits the engine load correction if it is in a low/medium range, thereby preventing an excessive exhaust gas to be recirculated and, at the same time, the third aspect is also capable of maintaining the output of the engine at high altitudes under a high load. The fourth aspect of the present invention maintains the output of the engine at high altitudes under a high load and corrects an engine load according to atmospheric pressure at high altitudes under a low/medium load in consideration of a decrease in the quantity of internally recirculated exhaust gas, thereby optimizing the quantity of internally recirculated exhaust gas at high altitudes.

What is claimed is:

1. A method of controlling a continuously variable valve timing mechanism of an internal combustion engine, to continuously and variably control an opening and closing timing of at least one of intake and exhaust valves of the engine, comprising the steps of:

- (a) preparing a map for determining a target valve timing according to an engine load, in which map a valve overlapping period in which both an intake and an exhaust valve are open is set smaller in a high load range than in a medium load range;
- (b) detecting an engine load;
- (c) detecting an atmospheric pressure;
- (d) correcting the detected engine load such that the lower the detected atmospheric pressure the more the engine load is increased; and
- (e) referring to the map to set the target valve timing for the valve timing mechanism according to the corrected engine load.

2. The method as claimed in claim 1, further comprising the step of:

- (f) prohibiting the correction in step (d) if the detected engine load is in a low to medium range.

3. The method as claimed in claim 1, further comprising the step of:

- (f) correcting the target valve timing set in step (e) to an intermediate value between the target valve timing and a valve timing determined according to the detected engine load, if the detected engine load is in a low to medium range.

4. An apparatus for controlling valve timing of an internal combustion engine, comprising:

a valve timing mechanism for continuously and variably controlling an opening and closing timing of at least one of intake and exhaust valves of the engine;

a control map for determining a target valve timing according to an engine load, in which map a valve overlapping period in which both an intake and an exhaust valve are open is set smaller in a high load range than in a medium load range;

load detection means for detecting an engine load;

pressure detection means for detecting an atmospheric pressure;

load correction means for correcting the detected engine load such that the lower the detected atmospheric pressure the more the engine load is increased; and

timing set means for setting the target valve timing for the valve timing mechanism by referring to the map according to the corrected engine load.

5. The apparatus as claimed in claim 4, further comprising:

prohibition means for prohibiting the correction carried out by the load correction means if the detected engine load is in a low to medium range.

6. The apparatus as claimed in claim 4, further comprising:

timing correction means for correcting the target valve timing set by the timing set means to an intermediate value between the target valve timing and a valve timing determined according to the engine load detected by the load detection means, if the detected engine load is in a low to medium range.

* * * * *

UNITED STATES PATENT AND TRADEMARK OFFICE
CERTIFICATE OF CORRECTION

PATENT NO. : 5,713,317
DATED : February 3, 1998
INVENTOR(S) : Mamoru YOSHIOKA

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

Column 2, line 3, change "setting target a" to --setting
a target--.

Column 9, line 25, change "highlands" to --altitudes--.

Column 10, line 27, claim 3, change "(f)" to --(g)--.

Column 10, line 57, change "lead" to --load--.

Column 10, line 63, change "lead" to --load--.

Column 10, line 64, change "lead" to --load--.

Column 10, line 65, change "lead" to --load--.

Signed and Sealed this

Twenty-fourth Day of November, 1998

Attest:



BRUCE LEHMAN

Attesting Officer

Commissioner of Patents and Trademarks

United States Patent [19]

Ohkubo et al.

[11] Patent Number: 4,995,351

[45] Date of Patent: Feb. 26, 1991

[54] VALVE TIMING CONTROL APPARATUS FOR AN INTERNAL COMBUSTION ENGINE

[75] Inventors: Satoru Ohkubo, Amagasaki; Akira Demizu, Himeji, both of Japan

[73] Assignee: Mitsubishi Denki Kabushiki Kaisha, Tokyo, Japan

[21] Appl. No.: 561,022

[22] Filed: Aug. 1, 1990

[30] Foreign Application Priority Data

Nov. 21, 1989 [JP] Japan 1-304259

[51] Int. Cl.⁵ F01L 9/04; F01L 1/34

[52] U.S. Cl. 123/90.11; 123/90.15

[58] Field of Search 123/90.11, 90.15, 90.16, 123/90.17

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80712 3/1989 Japan .

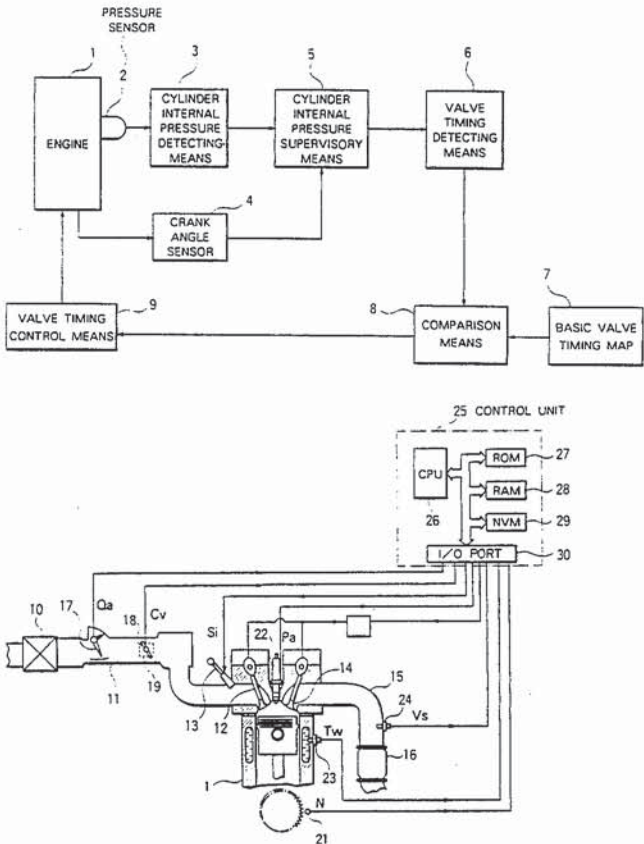
Primary Examiner—David A. Okonsky
Assistant Examiner—Weilun Lo

Attorney, Agent, or Firm—Sughrue, Mion, Zinn, Macpeak and Seas

[57] ABSTRACT

A valve timing control apparatus for an internal combustion engine capable of controlling the opening and closing timing of the intake and exhaust valves with a high degree of preciseness, as well as improving the air-intake and exhaust-discharge efficiency of the engine for increased engine output power. To this end, the valve timing control apparatus includes a pressure sensor for sensing the internal pressure in an engine cylinder, a crank angle sensor for sensing the crank angle of the engine, a valve timing detector for detecting the opening and closing timings of the intake and exhaust valves from the sensed cylinder internal pressure and the sensed engine crank angle, a disabling element for disabling the operation of the valve timing detector during combustion periods of the engine, a valve timing storage element for storing basic valve opening and closing timings for the intake and exhaust valves which are predetermined based on the operating conditions of the engine, a comparison element for comparing the detected valve opening and closing timings as detected by the valve timing detector and the predetermined valve opening and closing timings stored in the valve timing storage element so as to calculate deviations therebetween, and a valve timing control element for controlling the opening and closing timings of the intake and exhaust valves in accordance with the deviations thus calculated in such a manner that the actual opening and closing of the intake and exhaust valves take place at the predetermined basic valve opening and closing timings.

2 Claims, 5 Drawing Sheets



VW EX1006
U.S. Patent No. 6,557,540

FIG. 1

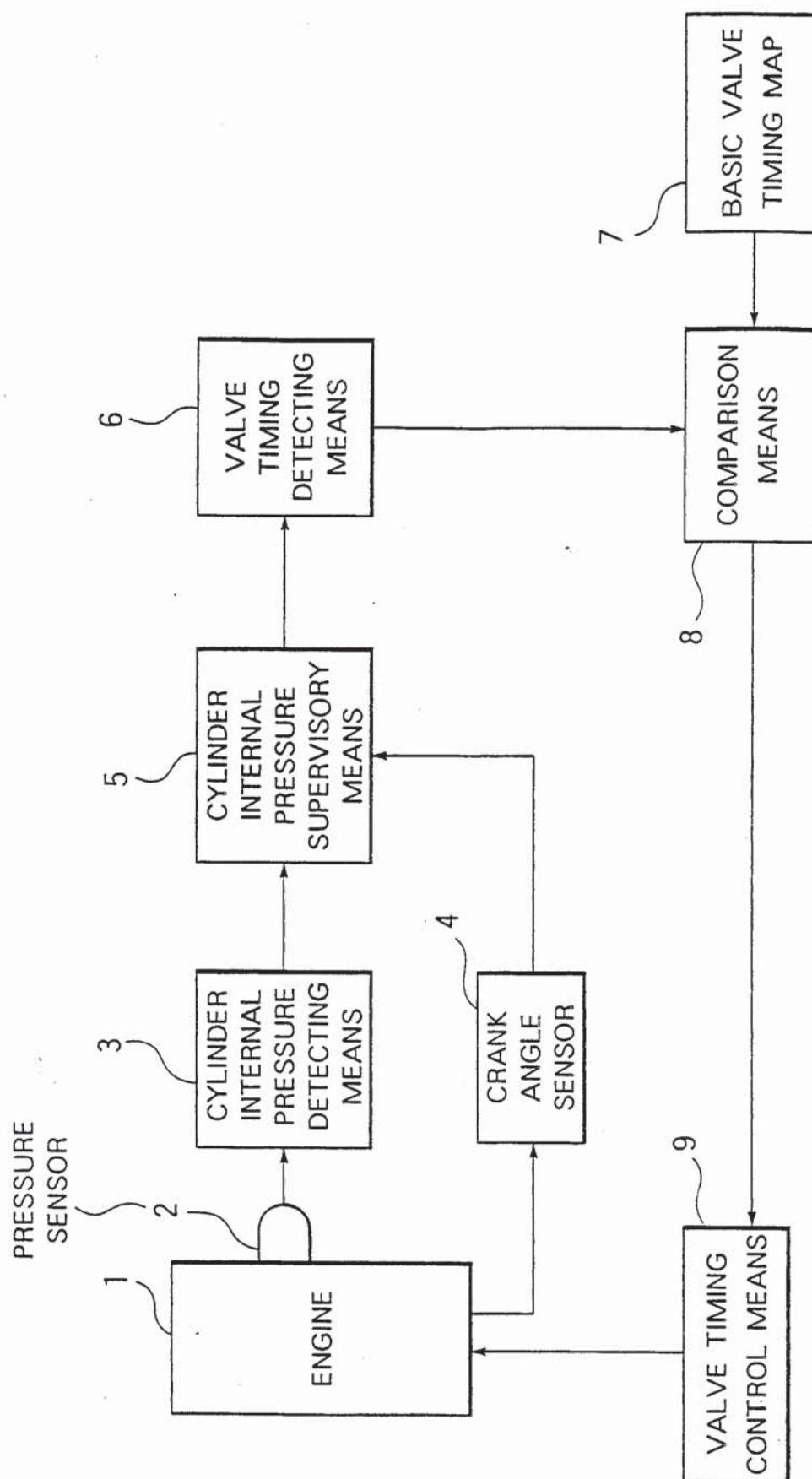


FIG. 2

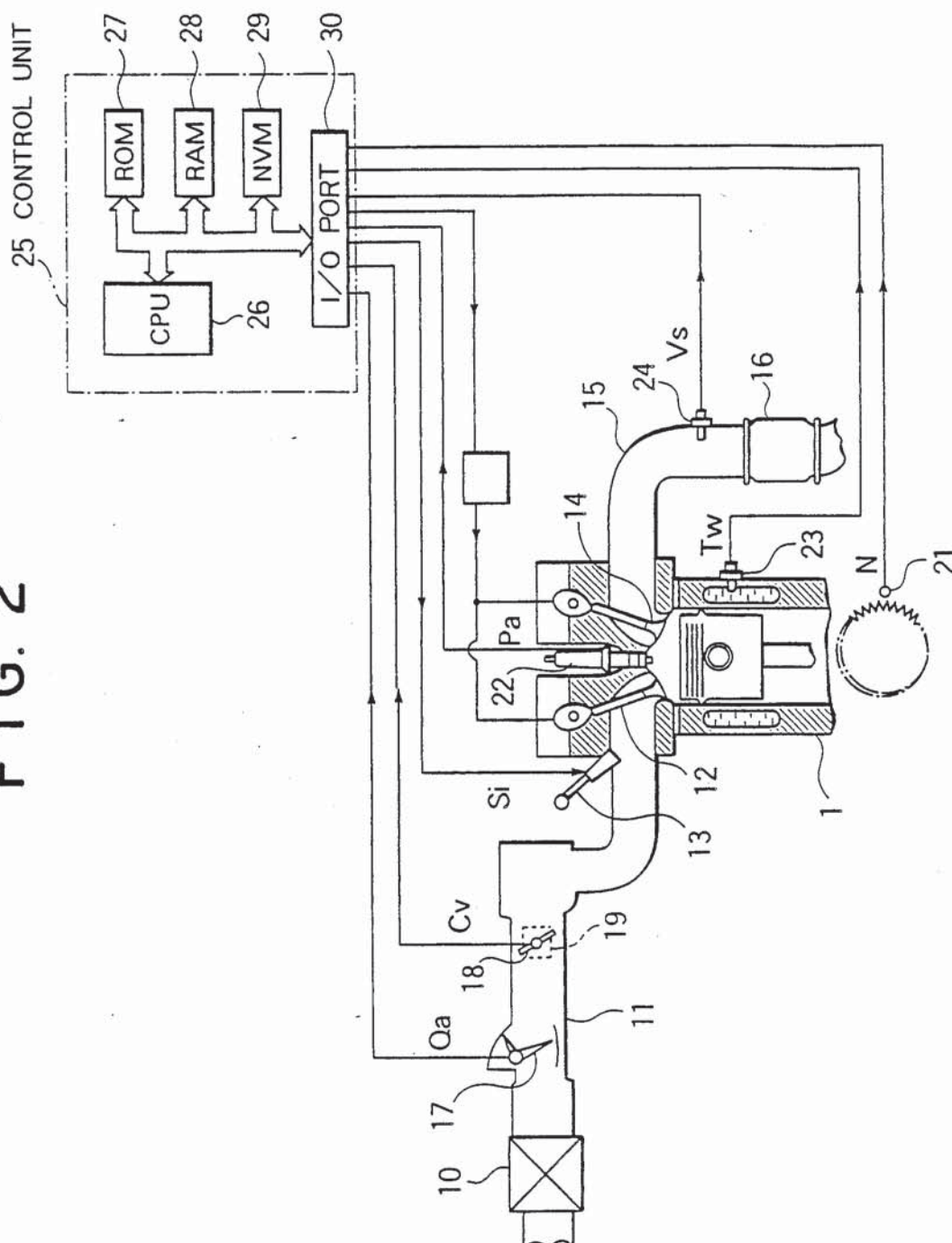


FIG. 3

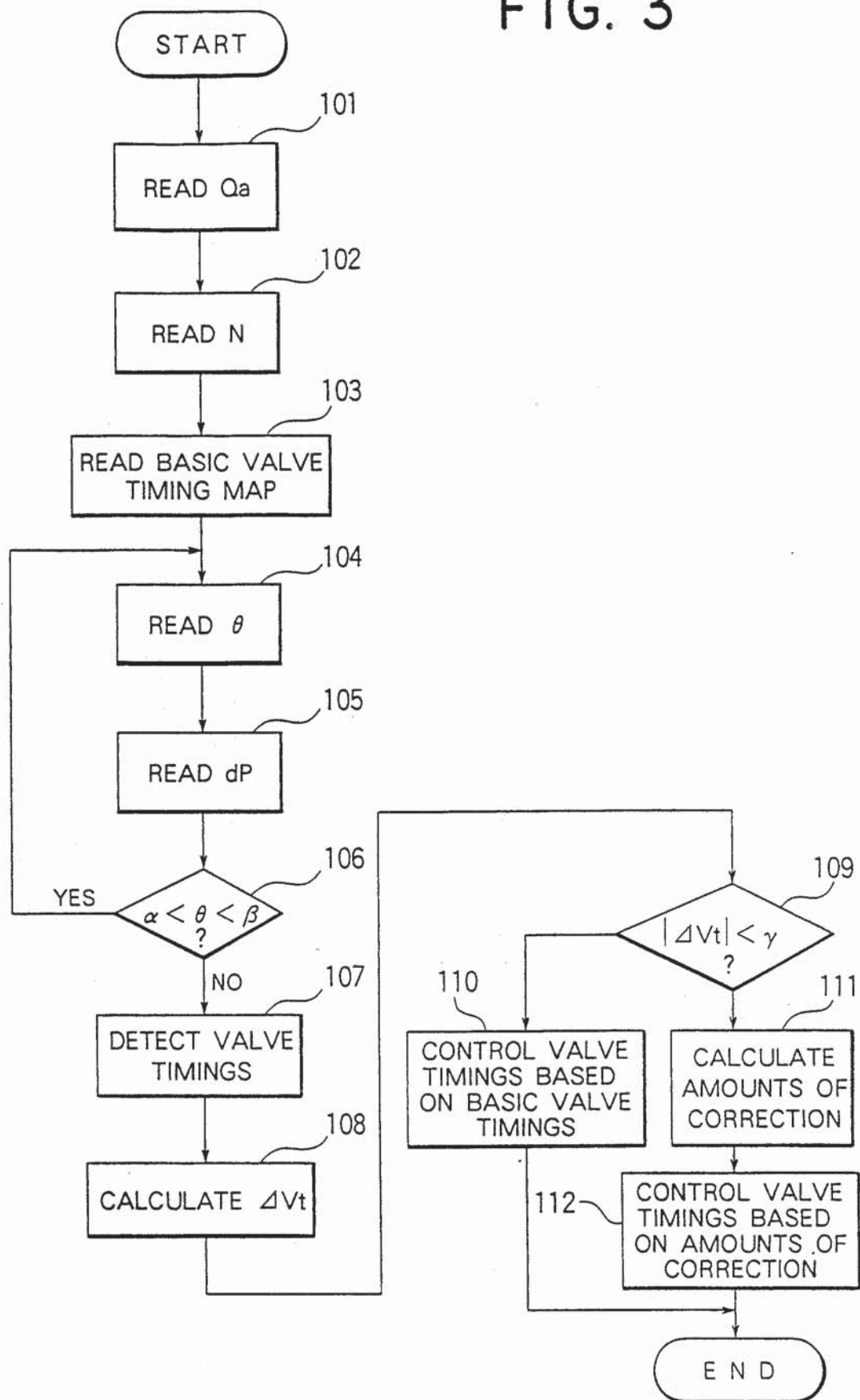


FIG. 4

IVo : INTAKE VALVE OPENING SIGNAL
IVc : INTAKE VALVE CLOSING SIGNAL
EVo : EXHAUST VALVE OPENING SIGNAL
EVc : EXHAUST VALVE CLOSING SIGNAL

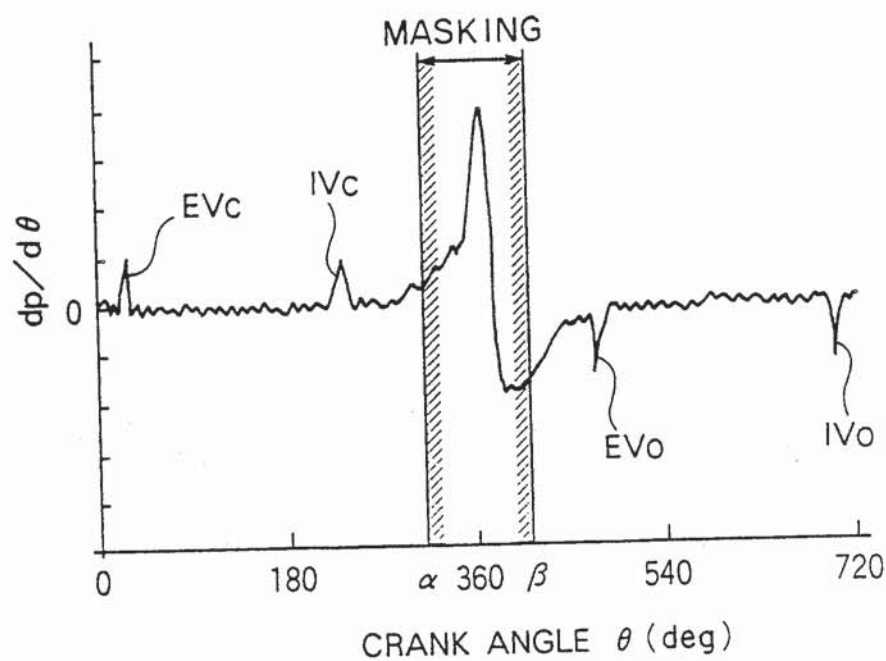


FIG. 5A

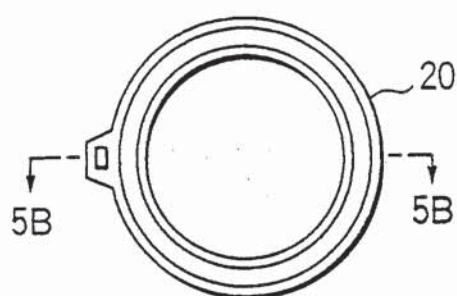


FIG. 5B

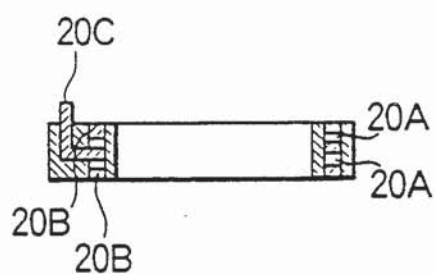
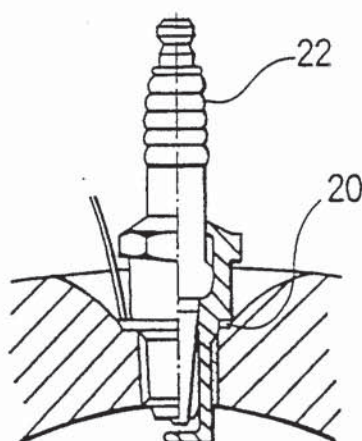


FIG. 5C



VALVE TIMING CONTROL APPARATUS FOR AN INTERNAL COMBUSTION ENGINE

BACKGROUND OF THE INVENTION

The present invention relates to a valve timing control system for an internal combustion engine which controls the opening and closing of intake and exhaust valves using a pressure sensor which senses the internal pressure in an engine cylinder.

Control apparatuses for controlling ignition timing or the quantity of fuel to be injected for an internal combustion engine in response to the internal pressure in an engine cylinder have been known. A typical example of such an ignition timing control apparatus is disclosed in Japanese Patent Publication No. 49-29209 in which peak crank positions of a piston in an engine cylinder at which the internal pressure in the engine cylinder takes a peak value are detected, and ignition timing of the cylinder is controlled in a feedback manner so as to make the peak pressure take place at prescribed piston position. Another typical example of such a fuel injection control apparatus is disclosed in Japanese Laid-Open No. 59-221433 in which the quantity of fuel to be injected into an engine cylinder is controlled based on the amount of intake air to be sucked therein which is calculated from the pressure differential between the cylinder internal pressure at bottom dead center of the piston during intake stroke and the cylinder internal pressure at a predetermined crank angle of the piston during compression stroke.

Further, it is well known that changing the opening and closed timings for the intake and exhaust valves of an engine depending upon the engine operating ranges results in improvements in the operating performance of the engine. Example of this are shown in Japanese Patent Laid-Open Nos. 64-80711 and 65-80712.

With the above-described conventional engine control apparatuses, however, valve timing control is performed in an open-loop manner, and hence accuracy in detecting the valve opening and closing timings is inadequate, so it is impossible to control the valve opening or closing timings with a high degree of preciseness.

SUMMARY OF THE INVENTION

Accordingly, the present invention is intended to obviate the above-described problems of the conventional engine control apparatuses.

It is an object of the present invention to provide a valve timing control apparatus for an internal combustion engine which is able to control the opening and closing timing of their intake and exhaust valves with a high degree of preciseness.

Another object of the present invention is to provide a valve timing control apparatus for an internal combustion engine which is able to improve the air-intake and exhaust-discharge efficiency of the engine, thereby increasing engine output power.

In order to achieve the above objects, according to the present invention, there is provided a valve timing control apparatus for an internal combustion engine equipped with an intake valve and exhaust valve, the valve timing control apparatus comprising:

a pressure sensor for sensing the internal pressure in an engine cylinder;

a crank angle sensor for sensing the crank angle of the engine;

valve timing detecting means for detecting the opening and closed timings of the intake and exhaust valves from the sensed cylinder internal pressure and the sensed engine crank angle;

disabling means for disabling the operation of the valve timing detecting means during combustion periods of the engine;

valve timing storage means for storing basic valve opening and closing timings for the intake and exhaust valves which are predetermined based on the operating conditions of the engine;

comparison means for comparing the detected valve opening and closing timings as detected by the valve timing detecting means and the predetermined valve opening and closing timings stored in the valve timing storage means so as to calculate deviations therebetween; and

valve timing control means for controlling the opening and closing timings of the intake and exhaust valves in accordance with the deviations thus calculated in such a manner that the actual opening and closing of the intake and exhaust valves take place at the predetermined basic valve opening and closing timings.

Preferably, the disabling means operates to suspend the operation of the valve timing detecting means for a period of time from a prescribed crank angle before top dead center to another prescribed crank angle after top dead center on each combustion stroke of the engine.

The above and other objects, features and advantages of the present invention will become more readily apparent from the following detailed description of a preferred embodiment of the invention taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a block diagram illustrating the conceptual construction of a valve timing control apparatus for an internal combustion engine in accordance with the present invention;

FIG. 2 is a view showing a concrete construction of a valve timing control apparatus embodying the conceptual construction of FIG. 1;

FIG. 3 is a flow chart showing the operating process of the valve timing control apparatus of FIG. 2;

FIG. 4 is a graphic representation showing the relationship between the change rate of the cylinder internal pressure and the crank angle of the engine;

FIG. 5(A) is a plan view of a pressure sensor of the present invention;

FIG. 5(B) is a cross section taken along the line 5B—5B in FIG. 5(A); and

FIG. 5(C) is a partially cross-sectional view showing the mounting state of the pressure sensor of FIGS. 5(A) and 5(B).

DESCRIPTION OF THE PREFERRED EMBODIMENT

The present invention will now be described in detail with reference to a preferred embodiment thereof as illustrated in the accompanying drawings.

Referring to the drawings and first to FIG. 1, there is shown the general conception of a valve timing control apparatus for an internal combustion engine in accordance with the present invention. The apparatus illustrated includes an engine cylinder 1, a pressure sensor 2 for sensing the internal pressure in the engine cylinder 1, a pressure detecting means 3 for detecting the internal pressure in the engine cylinder 2 from the output signal

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of the pressure sensor 2, a crank angle sensor 4 for generating a pulse signal, e.g., generating 360 pulses per revolution of the engine crank shaft, a cylinder pressure supervisory means or disabling means 5 for supervising the internal pressure in the engine cylinder 1, a valve timing detecting means 6 for detecting the opening and closing timings of the intake and exhaust valves of the engine, a basic valve timing map 7, a comparison means 8 for comparing the valve opening and closing timings detected by the valve timing detecting means 6 with the basic valve timing map 7, and a valve timing control means 9 for controlling the opening and closing timings of the intake and exhaust valves based on the output of the comparison means 8.

In operation, the cylinder pressure supervisory means 5 receives the output signal of the cylinder pressure detecting means 3 and the output signal of the crank angle sensor 4 and calculates, based on these signals, a change rate $dP/d\theta$ of the internal pressure P in the engine cylinder 1 relative to the crank angle of the piston in the engine cylinder 1, as illustrated in FIG. 4 in which EV_c represents the closing point of the exhaust valve, IV_c the closing point of the intake valve, EO the opening point of the exhaust valve, and IO the opening point of the intake valve. The valve timing detecting means 6 detects the opening and closing timings of the intake and exhaust valves from the output signal of the cylinder internal pressure supervisory means 5. In this regard, the cylinder internal pressure supervisory means 5 also acts as a disabling means for disabling the operation of the valve timing detecting means during combustion periods of the engine. The comparison means 8 compares the opening and closing timings of the intake and exhaust valves as detected by the valve timing detecting means 6 with the predetermined valve opening and closing timings as read from the basic valve timing map 7, and calculates deviations therebetween and sends them to the valve timing control means 9. The valve timing control means 9 controls the operation of the engine cylinder 1 in a feedback manner so that the actual opening and closing of the intake and exhaust valves take place at the prescribed timings.

FIG. 2 shows a more concrete embodiment of the present invention. In this embodiment, intake air is supplied to an engine cylinder 1 through an air cleaner 10 and an intake pipe 11, and fuel is injected by a fuel injector 13 into the intake pipe 11 based on an injection signal Si from a control unit 25 which will be described in detail later. The fuel thus injected forms an air/fuel mixture in the intake pipe 11 which is supplied to the engine cylinder 1 upon opening of the intake valve 12 and ignited there by a spark plug 22 installed on the cylinder head. The mixture in the engine cylinder 1 is thus combusted and then discharged therefrom as an exhaust gas to an exhaust pipe 15 upon opening of the exhaust valve 14. The exhaust gas then flows into a catalytic converter 16 wherein harmful components such as CO, HC, NOx, etc. in the exhaust gas are purified by a three-way catalyst contained therein. The exhaust gas thus purified is discharged from the catalytic converter 16 to the ambient atmosphere. The quantity of intake air Q_a sucked into the engine cylinder 1 is detected by an air flow meter 17 and controlled by a throttle valve 18 disposed in the intake pipe 11. The degree of opening C_v of the throttle valve 18 is sensed by a throttle sensor 19. The opening and closing timings

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of the intake and exhaust valve 12, 14 are sensed by a pressure sensor 20 and a crank angle sensor 21.

The pressure sensor 20 is in the form of a washer-type pressure sensor, as clearly shown in FIGS. 5(A) and 5(B), which is mounted around the spark plug 22 in the head of the engine cylinder 1 in the manner as shown in FIG. 5(C). As clearly seen from FIG. 5(B), the washer-type pressure sensor 20 has a pair of annular piezoelectric elements 20A, a pair of annular negative electrodes 20B disposed on the outer sides of the piezoelectric elements 20A, and a positive electrode interposed between the piezoelectric elements 20A and having an externally extended terminal lead. The piezoelectric elements 20A sense, through the spark plug 22, a change dP in the internal pressure P in the engine cylinder 1 which takes place upon the opening and closing of the intake and exhaust valves 12, 14, and generates an analog output signal Pa having a voltage corresponding to the sensed pressure change dP .

The crank angle sensor 21 senses the crank angle θ of the piston in the engine cylinder 1, and the crank angle at which the pressure change dP in the cylinder internal pressure P takes place is detected by the control unit 25 so as to control the opening and closing timings of the intake and exhaust valves 12, 14 in a precise manner. The crank angle sensor 21 also senses the number of revolutions per minute of the engine N . A temperature sensor 23 senses the temperature T_w of an engine coolant such as cooling water flowing through a water jacket formed in the engine cylinder 1. An oxygen sensor 24 senses the concentration of oxygen contained in the exhaust gas discharged from the engine cylinder 1, and to this end, it has a characteristic in which the output thereof abruptly changes when the air/fuel ratio of the mixture changes across the stoichiometric value.

The air flow meter 17, the throttle sensor 19 and the crank angle sensor 21 together constitute an engine operating condition detecting means, and the output signals of these sensors are inputted to the control unit 25 together with the output signals of the pressure sensor 20, the temperature sensor 23 and the oxygen sensor 24. The control unit 25 shares with the pressure sensor 20 the valve timing detecting function, and has by itself an air/fuel ratio setting function and an air/fuel ratio correcting function.

The control unit 25 includes a central processing unit (CPU) 26, a read only memory (ROM) 27, a random access memory (RAM) 28, a non-volatile memory (NVM) 29 and an input/output (I/O) port 30. The CPU 26 takes in external data as necessary from the I/O port in accordance with a program stored in the ROM 27, transmits data between the RAM 28 and the NVM 29, and determines whether the opening and closing timings of the intake and exhaust valves 12, 14 are optimal. Based on the result of such determination of the valve timings, the CPU 26 performs calculations for properly controlling the opening and closing timings of the intake and exhaust valves 12, 14, and outputs the resultant data of calculations to the I/O port 30 as required. The valve timing control means 31 receives the data from the I/O port 30 and properly controls, based thereupon, the opening and closing of the intake and exhaust valves 12, 14. In this regard, predetermined optimal valve opening and closing timings are stored in the NVM 29 in an appropriate form such as a map.

Next, the operation of the valve timing control apparatus of FIG. 2 will be described in detail while referring to the flow chart of FIG. 3. A program for per-

forming the processing steps as illustrated in the flow chart of FIG. 3 is stored in the ROM 27 and executed in synchrony with the rotation of the engine. First, in Step 101, the quantity of intake air Q_a is read in from the output signal of the air flow meter 17, and in Step 102, the number of revolutions per minute of the engine N is calculated based on the time interval between the successive basic signals (i.e., the time between successive signals generated by the crank angle sensor 21 every 360 degrees of the crank angle), or the number of pulses per a predetermined time of a position signal (which is issued by the crank angle sensor 21 every 1 degree of the crank angle). The time interval or the number of pulses is measured from the output signal of the crank angle sensor 21. Then in Step 103, the values for the basic valve timings such as IVo (intake valve opening timing), IVc (intake valve closing timing), EVo (exhaust valve opening timing) and EVc (exhaust valve closing timing) are read in from a two-dimensional table map which is stored in the NVM 29, the map having, as parameters, the quantity of intake air Q_a and the number of revolutions per minute of the engine N . In Step 104, the crank angle θ is read in from the output signal of the crank angle sensor 21, and in Step 105, the quantity of change dP in the cylinder internal pressure P is read in from the output signal of the pressure sensor 20. In Step 106, in order to prevent incorrect detection of the valve opening and closing timings due to knocking and the like, a certain interval such as, for example, the interval from 40 degrees before top dead center (the crank angle of α degrees in FIG. 4) to 40 degrees after top dead center (the crank angle of β degrees in FIG. 4) during the compression stroke, in which combustion takes place, is masked so that detection of the valve timings is suspended or halted. That is to say, in Step 106, it is determined whether $\alpha < \theta < \beta$, and if the answer is YES, then the program returns to Step 104. On the other hand, if the answer is NO, the program goes to Step 107 where the valve opening and closing timings are detected. In this case, detection of the valve timings is done by detecting the crank angle at which an abrupt change in the value of $dP/d\theta$ takes place. In Step 108, a deviation ΔV_t between each of the thus detected values and the corresponding one of the predetermined basic values for valve opening and closing timings is calculated, and then in Step 109, it is determined whether each ΔV_t falls within an allowable range γ . If the answer is YES, the valve opening and closing timings are determined to be correct and the program goes to Step 110 where the valve opening and closing timings are controlled based exactly on the basic valve timings determined from the basic valve timing map. On the other hand, if the answer is NO, the program goes to Step 111 where appropriate amounts of correction for properly correcting the valve timings are calculated. Then in Step 112, the valve opening and closing timings are correctively controlled based on the thus calculated amounts of correction in a feedback manner.

Although in the above-described embodiment, the engine having a single cylinder is described, the present invention is applicable to a multi-cylinder engine having two or more cylinders. Also, the quantity of intake air Q_a is detected by the air flow meter 17, it can be calculated from the pressure differential in the cylinder internal pressure at two arbitrary points during compression stroke. In addition, based on the cylinder internal pressure, other information can be calculated such as an average effective pressure in the engine cylinder, a heat generation rate, a peak-pressure generation position at

which a peak in the cylinder internal pressure takes place, so that the information thus calculated is utilized for controlling other engine control devices such as a fuel controller, an ignition controller, an exhaust gas recirculation (EGR) controller and the like.

As described in the foregoing, according to the present invention, the opening and closing timings of the intake and exhaust valves are detected based on the internal pressure in the engine cylinder and the crank angle of the engine, both of which are closely related to the valve opening and closing timings, so that the accuracy in the detection of the valve opening and closing timings is greatly improved. Further, the valve opening and closing timings are controlled in a feedback manner in accordance with the deviations between the detected values of the valve opening and closing timings and the predetermined basic values. This leads to further improvements in the accuracy of controlling the valve opening and closing timings. As a result, the air-intake efficiency and the exhaust-discharge efficiency of the engine can be improved, thus increasing the output power of the engine. In addition, during the combustion periods in which engine knocking is liable to occur, detection of the valve opening and closing timings is suspended so as to prevent reduction in accuracy of the valve opening and closing timing control due to incorrect or erroneous detection of the valve timings.

What is claimed is:

1. A valve timing control apparatus for an internal combustion engine equipped with an intake valve and exhaust valve, said valve timing control apparatus comprising:

a pressure sensor for sensing the internal pressure in an engine cylinder;

a crank angle sensor for sensing the crank angle of the engine;

valve timing detecting means for detecting the opening and closing timings of the intake and exhaust valves from the sensed cylinder internal pressure and the sensed engine crank angle;

disabling means for disabling the operation of said valve timing detecting means during combustion periods of the engine;

valve timing storage means for storing basic valve opening and closing timings for the intake and exhaust valves which are predetermined based on the operating conditions of the engine;

comparison means for comparing the detected valve opening and closing timings as detected by said valve timing detecting means and the predetermined valve opening and closing timings stored in said valve timing storage means so as to calculate deviations therebetween; and

valve timing control means for controlling the opening and closing timings of the intake and exhaust valves in accordance with the deviations thus calculated in such a manner that the actual opening and closing of the intake and exhaust valves take place at the predetermined basic valve opening and closing timings.

2. A valve timing control apparatus for an internal combustion engine as claimed in claim 1, wherein said disabling means operates to suspend the operation of said valve timing detecting means for a period of time from a prescribed crank angle before top dead center to another prescribed crank angle after top dead center on each combustion stroke of the engine.

* * * * *



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Reed et al.

(10) **Patent No.:** **US 6,308,671 B1**
(45) **Date of Patent:** **Oct. 30, 2001**

(54) **METHOD OF INCREASING TORQUE AND/OR REDUCING EMISSIONS BY VARYING THE TIMING OF INTAKE AND/OR EXHAUST VALVES**

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6,182,636 * 2/2001 Russell et al. 123/399

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(51) **Int. Cl.**⁷ **F01L 1/34; F02D 9/10**

(52) **U.S. Cl.** **123/90.15; 123/399; 123/478**

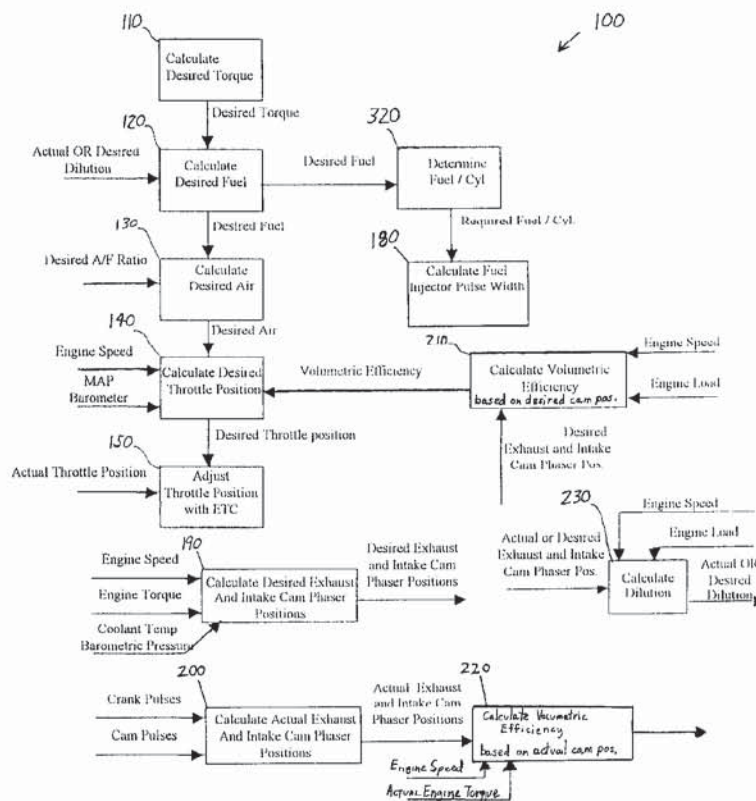
(58) **Field of Search** 123/90.15, 90.16, 123/90.17, 90.18, 90.31, 399, 478; 701/103, 104

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10 Claims, 8 Drawing Sheets



VW EX1007
U.S. Patent No. 6,557,540

FIGURE 1

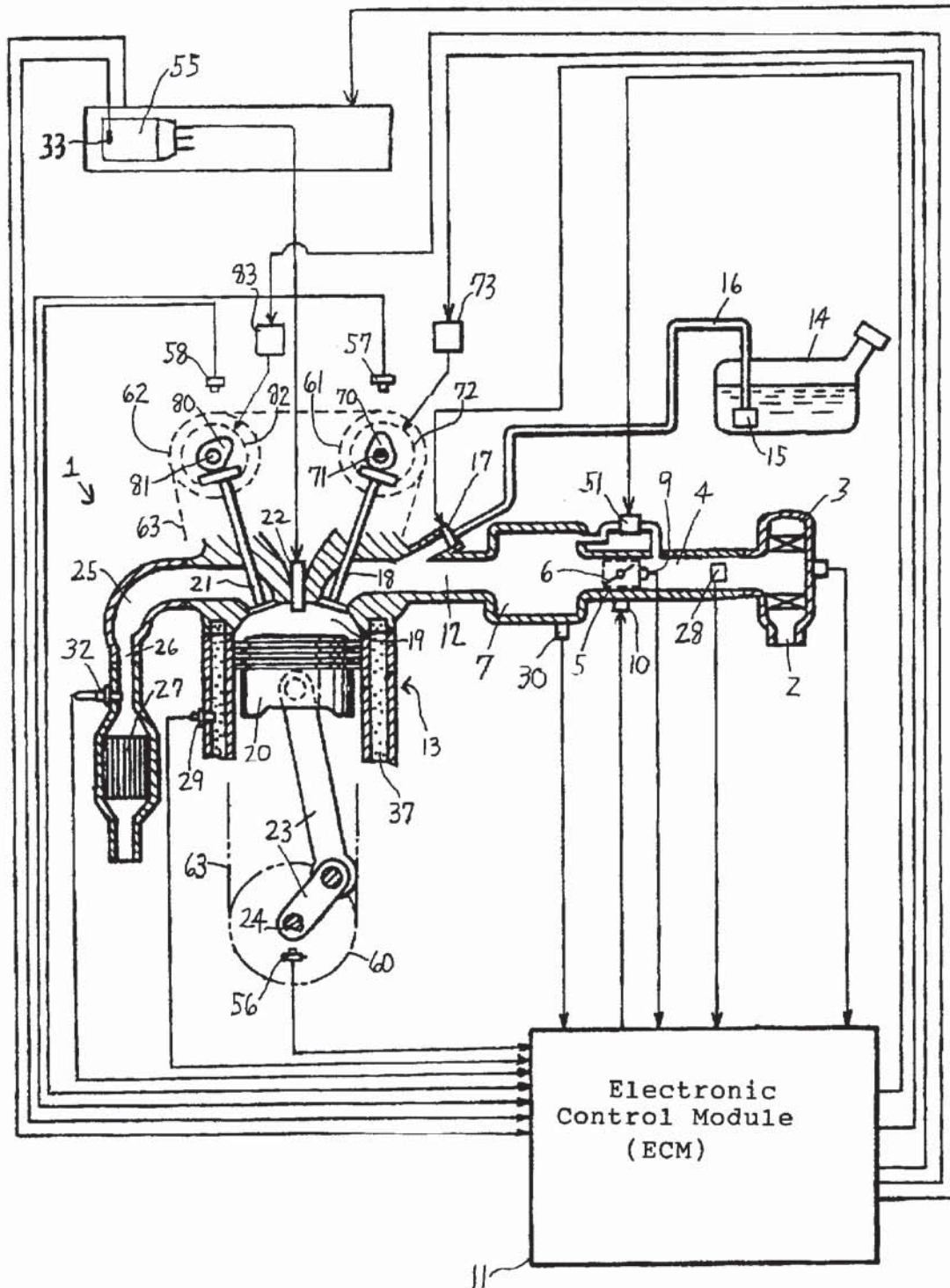


FIGURE 2

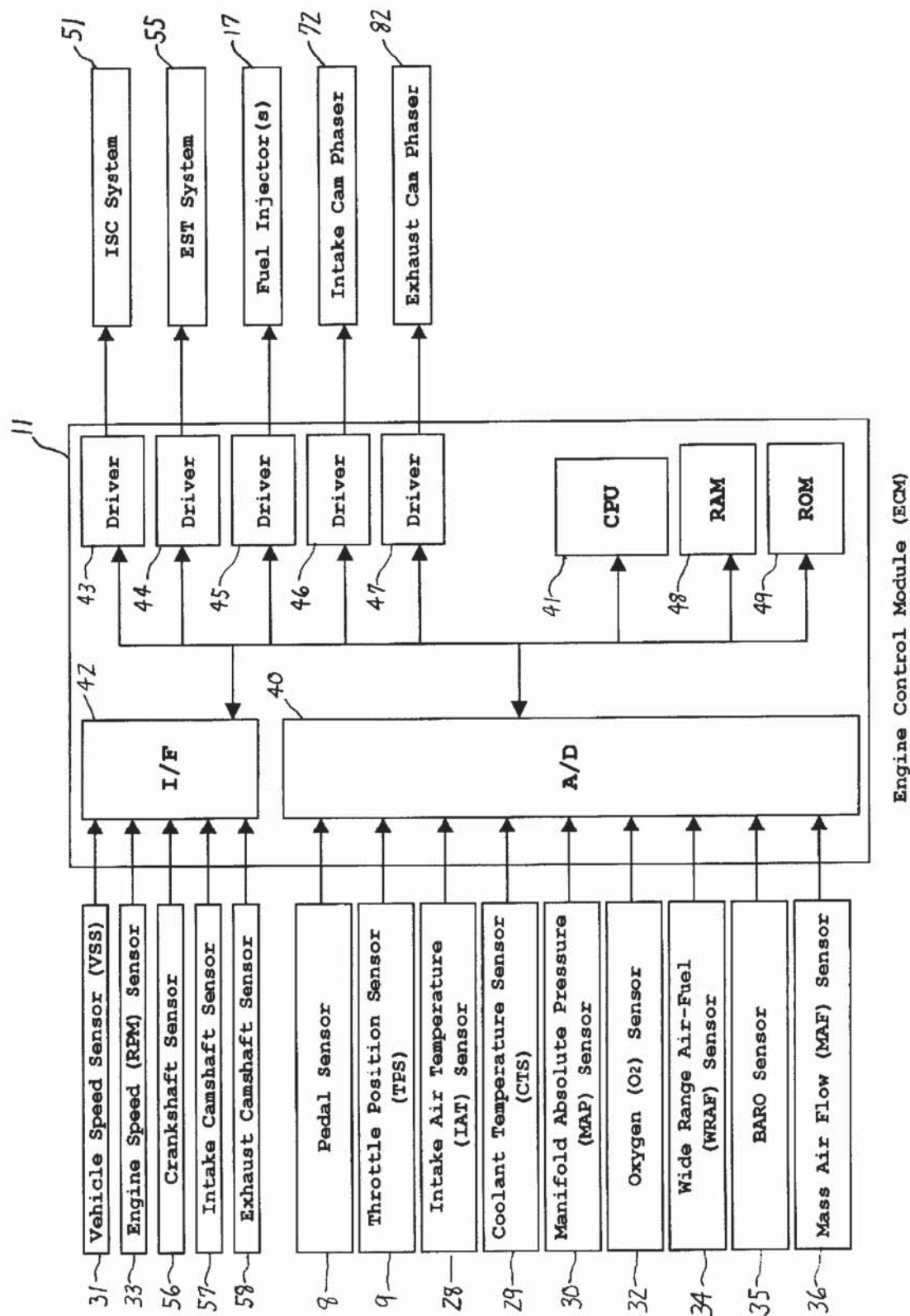


FIGURE 3

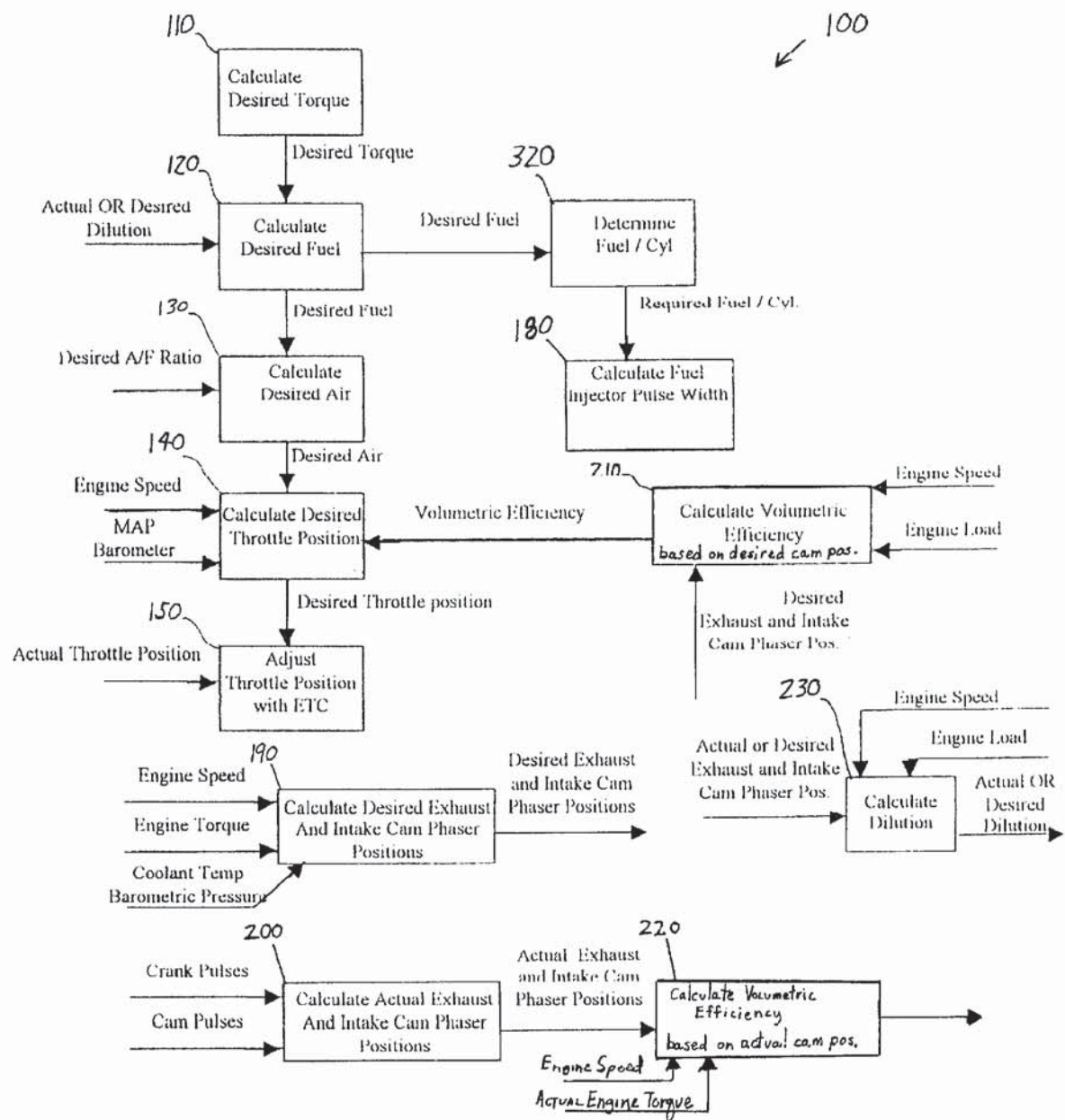


FIGURE 4

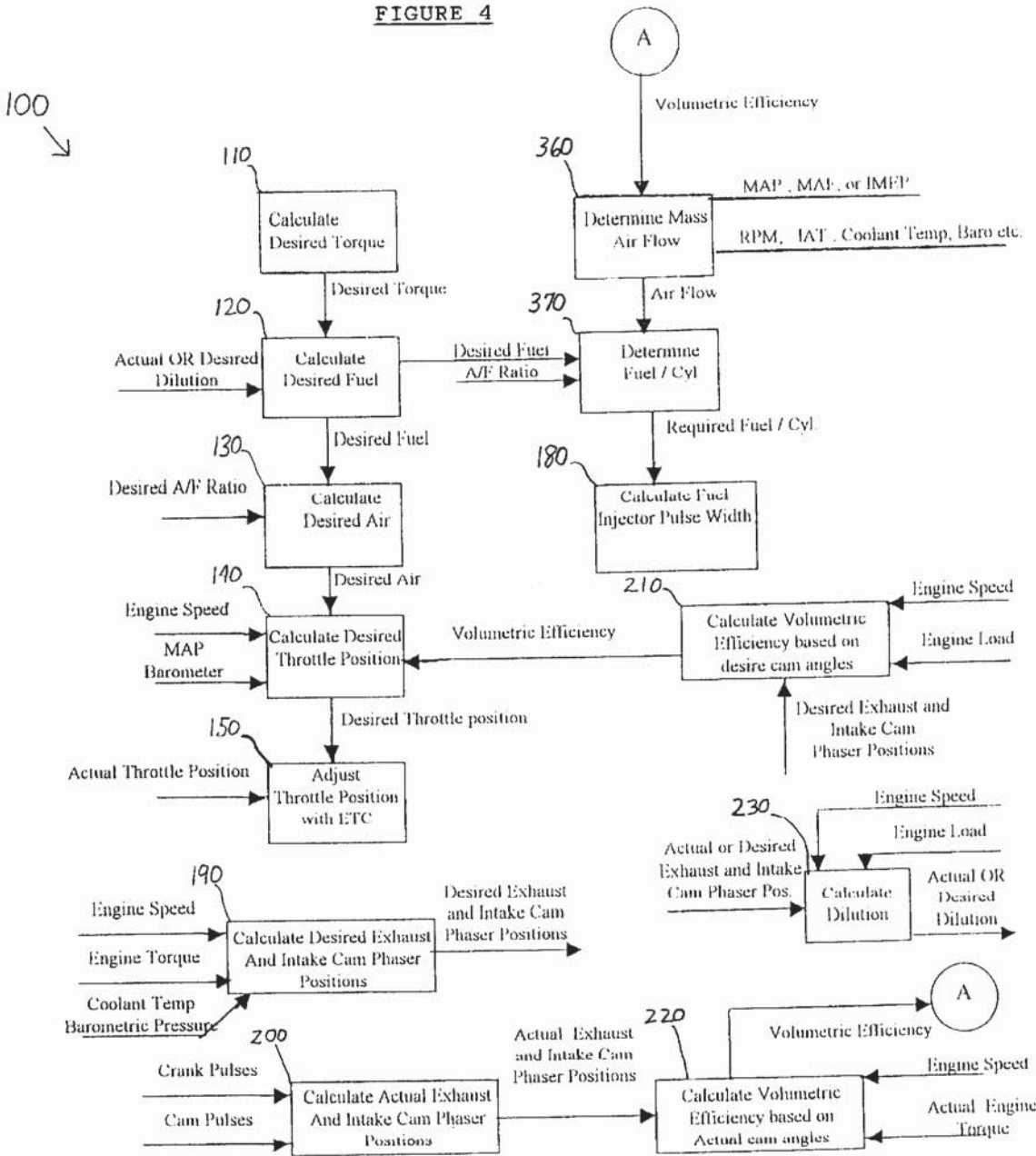


FIGURE 5

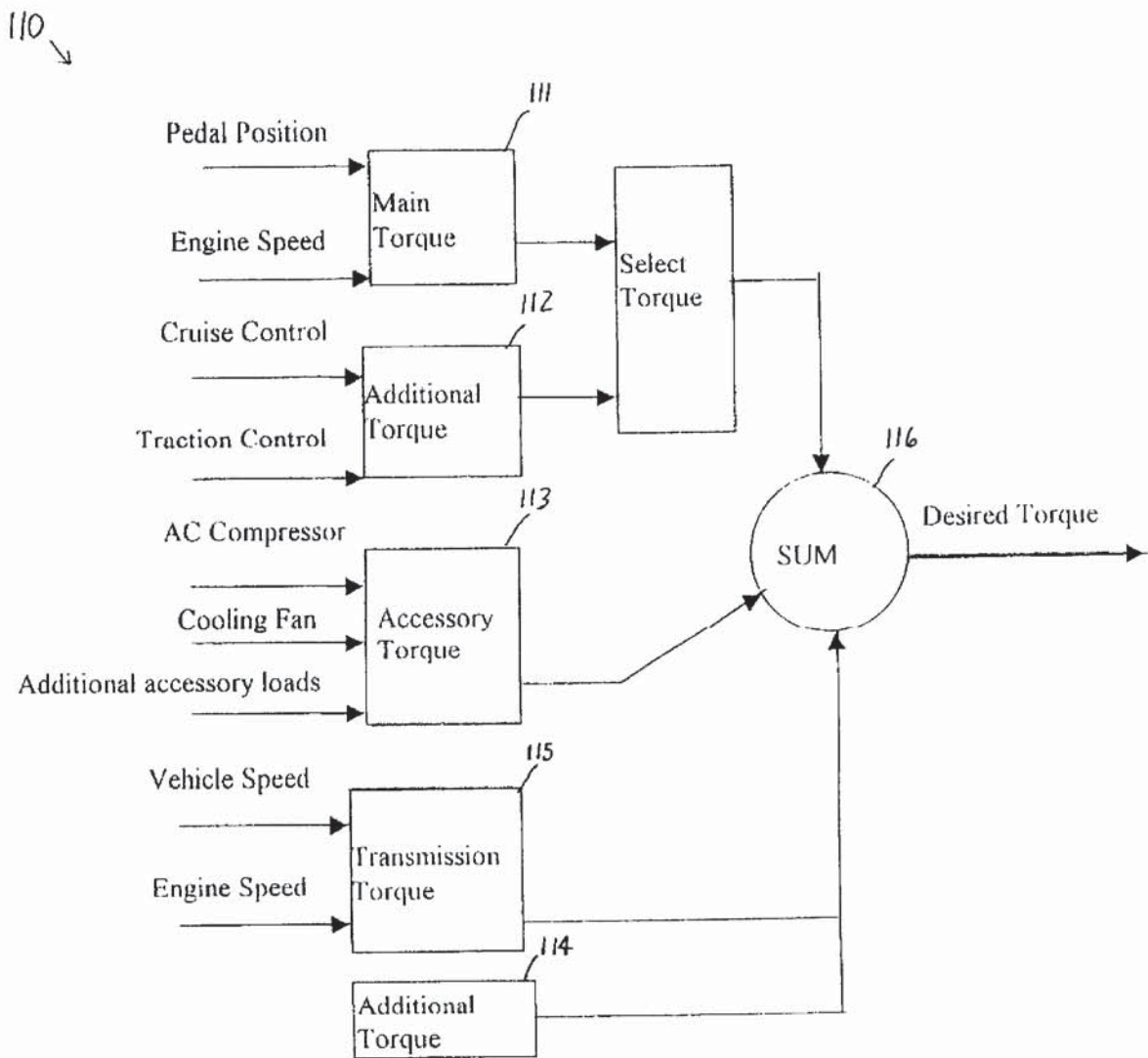


FIGURE 6

MAP/MEP %TORQUE	800 RPM	1200 RPM	1600 RPM	2000RPM	2400 RPM	2800 RPM	3200 RPM	3600 RPM
20/0/0	0	0	0	10	10	20	27	20
30/200/12.5	0	0	0	10	15	20	33	20
40/400/25	0	0	0	10	20	25	35	20
50/600/37.5	8	12	20	20	30	30	40	20
60/800/50	10	18	32	36	40	44	46	20
70/1000/62.5	12	20	35	40	40	40	40	20
80/1200/75	30	30	40	50	40	42	45	20
90/1400/87.5	40	40	40	50	40	40	43	20
100/1600/100	50	40	40	50	40	37	41	20

	4000 RPM	4400 RPM	4800 RPM	5200 RPM	5600 RPM	6000 RPM	6400 RPM
20/0/0	15	10	7	5	0	0	0
30/200/12.5	15	10	7	5	0	0	0
40/400/25	15	10	7	5	0	0	0
50/600/37.5	15	10	7	5	0	0	0
60/800/50	15	10	7	5	0	0	0
70/1000/62.5	15	10	7	5	0	0	0
80/1200/75	15	0	0	0	0	0	0
90/1400/87.5	15	0	0	0	0	0	0
100/1600/100	15	0	0	0	0	0	0

FIGURE 7

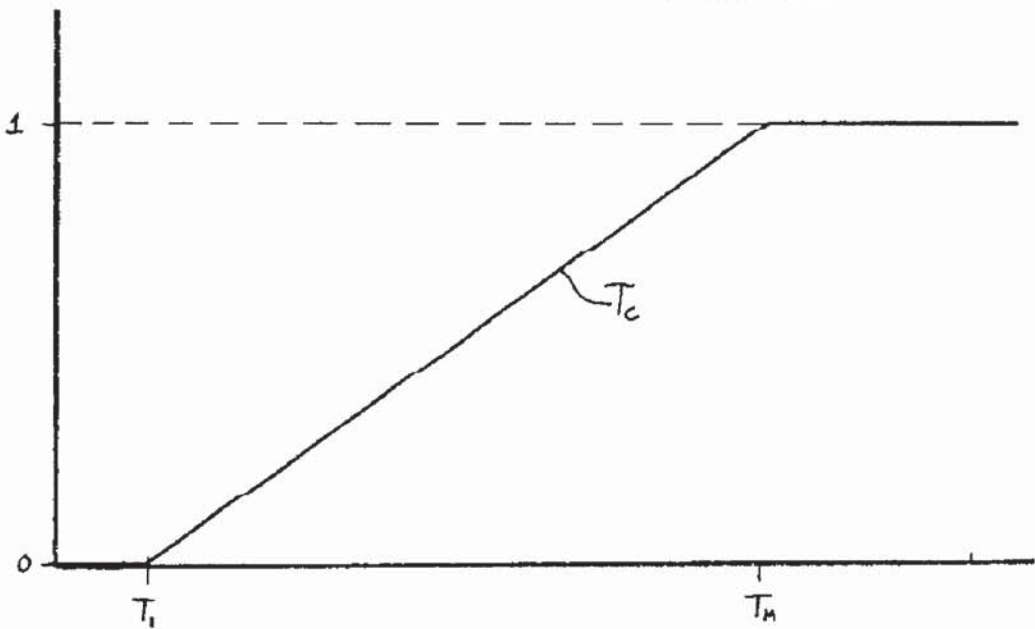


FIGURE 8

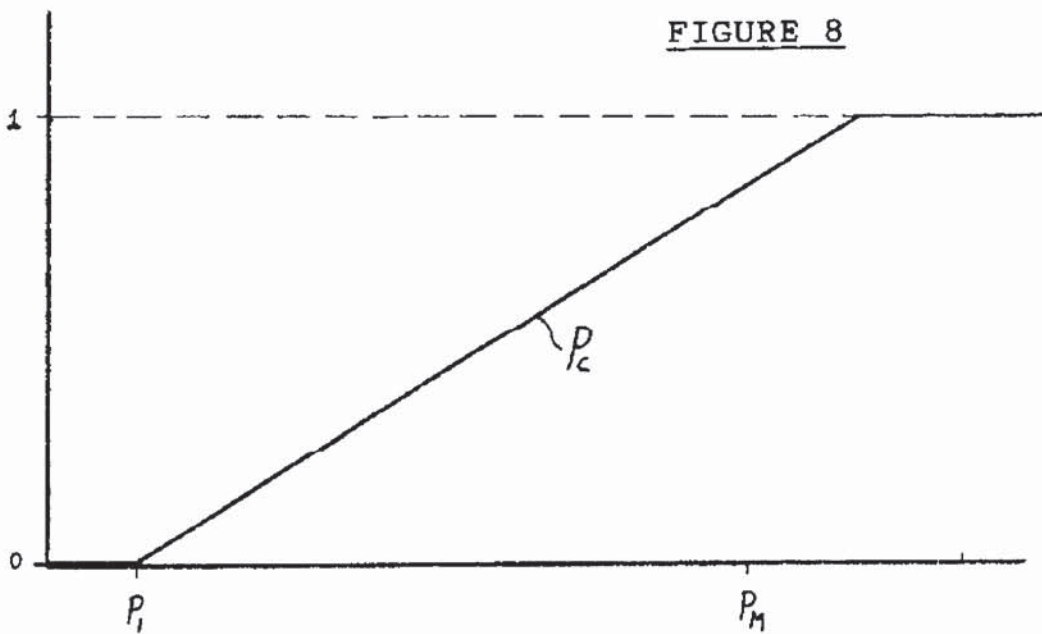


FIGURE 9

MAP/MEP/ %TORQUE	800 RPM	1200 RPM	1600 RPM	2000 RPM	2400 RPM	2800 RPM	3200 RPM	3600 RPM
20/0/0	0	0	0	0	10	10	15	20
30/200/12.5	0	0	0	0	10	10	15	25
40/400/25	0	0	0	0	10	10	15	25
50/600/37.5	8	12	16	16	20	15	20	30
60/800/50	10	18	16	16	25	20	44	50
70/1000/62.5	12	16	16	16	40	30	40	50
80/1200/75	0	0	0	0	0	10	10	10
90/1400/87.5	0	0	0	0	0	0	0	0
100/1600/100	0	0	0	0	0	0	0	0

MAP/MEP/ %TORQUE	4000 RPM	4400 RPM	4800 RPM	5200 RPM	5600 RPM	6000 RPM	6400 RPM
20/0/0	34	32	0	0	0	0	0
30/200/12.5	38	37	0	0	0	0	0
40/400/25	25	20	0	0	0	0	0
50/600/37.5	25	20	0	0	0	0	0
60/800/50	25	20	0	0	0	0	0
70/1000/62.5	25	20	0	0	0	0	0
80/1200/75	10	20	0	0	0	0	0
90/1400/87.5	0	0	0	0	0	0	0
100/1600/100	0	0	0	0	0	0	0

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METHOD OF INCREASING TORQUE AND/OR REDUCING EMISSIONS BY VARYING THE TIMING OF INTAKE AND/OR EXHAUST VALVES

FIELD OF THE INVENTION

The present invention relates, in general, to an internal combustion engine of the type equipped with an electronic throttle control system and a variable valve timing mechanism(s), and of the type controlled by a torque based management system. More particularly, the invention pertains to a method of increasing engine torque and/or of reducing engine emissions by varying the phasing of either the intake valves or the exhaust valves or both.

BACKGROUND OF THE INVENTION

The following background information is provided to assist the reader to understand one of the many environments in which the invention will typically be used. Upon reading this document, the reader will appreciate that the invention may also be applied or adapted to environments other than that described below.

FIG. 1 illustrates one cylinder of an electronically controlled multi-cylinder engine that is equipped with a mechanism capable of varying the timing of the opening and closing of the intake and exhaust valves. While the engine 1 is operating, air at atmospheric pressure is drawn into an inlet 2 through a filter 3 and into an intake duct 4. The incoming air then flows into a throttle body 5 in which is disposed a throttle valve 6. The throttle valve 6 typically takes the form of a rotatable plate.

Controlled by an electronic throttle control (ETC) system, the throttle plate 6 has its position adjusted regularly to allow an amount of air appropriate to present conditions to pass through the throttle body 5 and thereafter into an intake manifold 7. The throttle control system typically features a pedal sensor 8, a throttle position sensor (TPS) 9, a motor 10 and an electronic control module (ECM) 11. The pedal sensor 8 enables the ECM 11 to monitor the position of the accelerator pedal, and thus to determine whether the driver wants the vehicle to maintain, increase or decrease torque. The TPS sensor 9 enables the ECM 11 to monitor the angular position that the throttle plate 6 occupies in the throttle body 5. Pursuant to prior art algorithms, the ECM 11 uses the input from these sensors, as well as other sensors, such as those shown in FIG. 2, to control the engine 1 so that it delivers the desired torque according to the conditions under which the vehicle is operating. In doing so, the ECM 11 controls via motor 10 the position of the throttle plate 6, and thus the quantity of air that is drawn into the intake manifold 7.

From the intake manifold 7 the incoming air then passes to an intake duct 12 that leads to the cylinder 13. Meanwhile, fuel from a fuel tank 14 is pumped via a pump 15 through a pipe 16 to a fuel injector 17. According to known practice, the ECM 11 uses data from several sensors to calculate the injector pulse width, i.e., the electrical signal that the ECM 11 uses to activate the fuel injector 17 for a time appropriate to current conditions. Activated via a drive circuit, the fuel injector 17 injects the precise amount of fuel into the intake duct 12. There, the fuel mixes with the inlet air coming from the intake manifold 7.

As noted in greater detail infra, a cam timing mechanism drives the intake valve 18 to the open position in timed relationship with the intake cycle of cylinder 13. During the intake cycle, a low pressure condition develops within the

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combustion chamber 19 due to the downward movement of a piston 20 within the cylinder 13. The low pressure draws the fuel-air mixture from the intake duct 12 past the intake valve 18 and into the combustion chamber 19. For the subsequent compression cycle, the action of the cam timing mechanism(s), as noted infra, closes the intake and exhaust valves 18 and 21 at the top of the cylinder 13. During the compression cycle itself, as is well known, the upward movement of the piston 20 compresses the air-fuel mixture in the combustion chamber 19 of the cylinder 13.

During the combustion cycle, the fuel-air mixture is ignited and exploded to produce power. Operating according to the spark sequence controlled by an electronic spark timing system, the ECM 11 causes the air-fuel mixture to be ignited in the combustion chamber 19. More specifically, the combustion cycle is initiated, at the appropriate time, by a spark driven across the spaced electrodes of a spark plug 22. The explosive force of the combustion drives the piston 20 downward within cylinder 13. The downward thrust of the piston 20 is imparted via connecting rods 23 as a torque upon a crankshaft 24 of the engine 1. Combined with the torque it receives from the other pistons in the engine 1, the crankshaft 24 drives the wheels and the accessory loads, etc., of the motor vehicle, as is generally understood in the art.

For the exhaust cycle, a cam timing mechanism drives the exhaust valve 21, at the appropriate time, to the open position. During the exhaust cycle, the upward movement of the piston 20 forces the exhaust gases produced by combustion past the exhaust valve 21 and into an exhaust manifold 25. An exhaust pipe 26 then channels the exhaust gases to a catalytic converter 27. A catalyst within the converter 27 aids the oxidization of unburned constituents, such as carbon monoxide (CO) and hydrocarbons (HC), and the reduction of nitrogen oxides (NO_x). From the converter 27, the purified exhaust gases are conveyed typically through a muffler and then through a tail pipe to atmosphere.

The ECM 11 monitors and controls the operation of the engine 1 through many data sensors, switches and control devices, some of which are shown in FIGS. 1 and 2. In addition to the pedal and TPS sensors 8 and 9, the data sensors include an intake air temperature (IAT) sensor 28, a coolant temperature sensor (CTS) 29, a manifold absolute pressure (MAP) sensor 30, a vehicle speed sensor (VSS) 31, an oxygen (O₂) sensor 32, and an engine speed (RPM) sensor 33. On some vehicles, additional data sensors are used. These include a wide range air-fuel (WRAF) sensor 34, a barometric pressure (BARO) sensor 35, and a mass air flow (MAF) sensor 36. The devices and subsystems that the ECM 11 controls, include the electronic throttle control system, the electronic spark timing system, the fuel injection system and the cam timing mechanisms.

The data sensors generate electrical signals, typically in analog form, indicative of the parameters they are intended to measure. The IAT sensor 28 typically measures the temperature of the air in the inlet 2 of the engine 1. The CTS sensor 29 senses the temperature of the coolant that flows in channels 37 around the cylinders to keep the engine cool. The MAP sensor 30 measures the absolute air pressure in the intake manifold 7. The VSS, sensor 31 generates a pulse representing the actual speed of the vehicle. The O₂ sensor 32 is typically mounted to the exhaust system downstream of the converter 27 so that its head lies exposed to the stream of exhaust gases produced by the engine 1. It senses the free oxygen concentration in the exhaust gases, and conveys a corresponding signal to the ECM 11. Typically exposed to the exhaust gases upstream of the converter 27, the WRAF sensor 34 measures the air-fuel ratio. It is used on some

vehicles to measure directly the ratio of air to fuel for purposes of controlling the delivery of fuel to the engine 1. The ECM 11 uses the signals from the O₂ and WRAF sensors 32 and 34 to control more precisely the fuel-air mixture to achieve stoichiometry. This correction process is known as closed loop operation.

On vehicles equipped with BARO and MAF sensors, the BARO sensor 35 measures the pressure of the ambient air and provides data to the ECM 11 as to pressure changes due to altitude and weather. The MAF sensor 36 measures the rate at which the air mass flows into the intake manifold 7. For vehicles not equipped with a BARO sensor 35, the ECM 11 is programmed to estimate the barometric pressure using data from various other sensors according to well-known practice. For vehicles not equipped with a MAF sensor 36, the ECM 11 estimates the air mass flow rate using data from the various other sensors, as is also known in the art.

The analog signals generated by the data sensors are conveyed to the ECM 11 where an A/D converter 40 converts them into digital signals. This conversion is necessary because the central processing unit (CPU) 41 of the ECM 11 can only manipulate digital information. Along with the input received by the interface (I/F) 42, the digital sensor data is conveyed to input registers in the ECM 11. Using the data it reads from the registers, the CPU 41 not only performs the mathematical computations and logic functions necessary to calculate inter alia the spark timing, the cam timing and the proper fuel-air mixture, but also provides control signals through drive circuits 43-47. The CPU 41 performs all of its functions according to the programming code stored in its associated memory devices. The memory devices include random access memory (RAM) 48 and read only memory (ROM) 49 inclusive of programmable ROM (PROM). The CPU 41 uses RAM 48 to temporarily store information such as the data received from the data sensors, the diagnostic codes and the results of its calculations. The ROM 49 is where the calibration data and fuel delivery algorithms are typically stored along with various lookup tables and control algorithms that collectively constitute the programming code. The elements in the ECM 11 are connected to one another through a system bus 50 containing address, data and control buses.

Used primarily to maintain the engine 1 at idle, the idle speed control (ISC) system includes the ECM 11 and an idle air control (IAC) valve 51. The IAC valve 51 is situated in a flow path parallel to that through the throttle body 5. Upon closure of the throttle plate 6 and feedback from the VSS sensor 31 indicating the vehicle has stopped, the ISC system compares the actual engine speed with a target engine speed it derives according to known practice. Based on the difference between the target and actual values, the ISC system controls the IAC valve 51 via drive circuit 43 so as to adjust the rate at which air flows into the engine 1 and thereby attain the target idle speed.

The electronic spark timing (EST) system includes the ECM 11, the RPM sensor 33 and a distributor module 55. The RPM sensor 33 generates a pulse for every 30 degrees that the crankshaft 24 rotates, thereby providing a measure of the speed, or revolutions per minute (rpm), at which the engine 1 is operating. Through the data sensors, the ECM 11 monitors the speed and other operating conditions of the engine 1, and, from those parameters, calculates the proper spark timing. According to the spark timing sequence, the ECM 11 then directs the distributor module 55 via drive circuit 44 to distribute to each of the spark plugs 22, at the appropriate time, the energy required to achieve combustion.

The fuel injection system includes the ECM 11 and the fuel injector 17. Operating according to known principles,

the ECM 11 uses data from several sensors to calculate the target air-fuel ratio. The mass of intake air per engine revolution is calculated from the mass flow rate of intake air measured by the MAF sensor 36 and the engine speed detected by the RPM sensor 33. Alternatively, it may also be estimated using data from other sensors, such as the MAP sensor 30 and the RPM sensor 33. Using the mass of intake air per engine revolution, the ECM 11 then determines the injector pulse width warranted by the current operating conditions. The ECM 11 continually adjusts the injector pulse width to correct for changes in various parameters, such as in the readings taken from the TPS, IAT, O₂ and WRAF sensors, so as to maintain as closely as possible the target air-fuel ratio. At a given angle in the operational cycle of the crankshaft 24, the ECM 11 then directs drive circuit 45 to inject fuel from the fuel injector 17 for the time dictated by the injector pulse width.

Most four cycle engines are designed so that the intake and exhaust valves operate (i.e., open and close) in a fixed angular relationship to the angular position of the crankshaft. Many engines use only a single camshaft to control the opening and closing of the intake and exhaust valves. The newer, more advanced engines often use a dual cam arrangement, i.e., one camshaft to control the open/close timing of the intake valves and another camshaft to govern the open/close timing of the exhaust valves. In either case, each valve is biased by a spring to the closed position. Affixed to the camshaft(s) are as many cams as there are valves, with the cams for the intake valves being oriented at one angle and the cams for the exhaust valves being oriented at another angle. Because a camshaft rotates at half the speed of the crankshaft, each intake cam causes its corresponding intake valve to be open (against the bias of the spring) and closed at fixed intervals during the operational cycle of the crankshaft. Similarly, each exhaust cam causes its corresponding exhaust valve to be open and closed at fixed intervals.

The term "standard cam timing" refers to the opening and closing of the intake and/or exhaust valves at such fixed intervals. In engines that employ standard cam timing, a compromise must be reached between how smooth will the engine run at idle, how much torque will it be able to deliver at medium to high speeds, the toxicity of its emissions, aid how much fuel will the engine consume. It involves a decision as to when and how long the intake and exhaust valves should be open at the same time (i.e., valve overlap). The amount and phasing of valve overlap is a trade-off between stable idling and the amount of power that will be available at medium to high speeds. It also is a trade-off between engine performance, emissions and fuel economy.

The automotive industry is now investigating the use of variable cam timing (VCT) schemes to improve the overall performance of an engine without the strict compromises required by standard cam timing techniques. VCT allows the timing of the camshafts, and thus the opening and closing of the valves, to be optimized over a wider range of operating conditions. It offers the possibility of improved performance at medium to full loads coupled with reduced emissions and improvements in fuel economy.

Referring to FIG. 1, the opening and closing of intake valve 18 is controlled by a cam 70 attached to an intake camshaft 71. A cam 80 attached to a camshaft 81 likewise controls the opening and closing of exhaust valve 21. As the pistons reciprocate within their respective cylinders, the torque they impart to the crankshaft 24 via the connecting rods 23 also drives a timing pulley 60. Each camshaft at its end also has a pulley, with camshaft 71 having timing pulley

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61 and camshaft 81 having timing pulley 62. A timing belt 63 connects the timing pullers 60, 61 and 62. Consequently, as the crankshaft 24 rotates, it also drives the camshafts 71 and 81, with the cams 70 and 80 thereon opening and closing the intake and exhaust valves 18 and 21 at predetermined angles in the operational cycle of the crankshaft 24. A crankshaft sensor 56 generates a set number of pulses (e.g., 58 pulses) for each rotation of the crankshaft 24. Similarly, there are two camshaft sensors 57 and 58. Each camshaft sensor 57 and 58 generates a set number of pulses (e.g., 4 pulses) for each rotation of its respective camshaft 71 and 81.

The dual cam engine shown in FIG. 1 has two continuously variable cam timing mechanisms 72 and 82, one for the intake valves and the other for the exhaust valves. Controlled by the ECM 11, each VCT mechanism enables its respective camshaft to be phase-shifted relative to the crankshaft 24 as a function of the conditions under which the engine 1 is operating. Also referred to as cam phasers, VCT mechanisms take a variety of forms such as the vane type or helical gear type cam phasers. The latter is discussed below for illustrative purposes.

Situated between the camshaft 71 and the timing pulley 61, the intake VCT mechanism 72 turns the camshaft 71 and timing pulley 61 relative to each other. More specifically, the intake cam phaser 72 uses the camshaft 71 and timing pulley 61 as external gears and interconnects them via an intermediate helical gear. Through drive circuit 46, the ECM 11 controls a valve 73 that affects the hydraulic pressure acting upon the helical gear. It also uses feedback from sensors 56 and 57 to monitor the angular relationship between the crankshaft 24 and the intake camshaft 71. By changing the hydraulic pressure via valve 73, the ECM 11 can move the helical gear axially, and thus alter the angular relationship between the intake camshaft 71 and the timing pulley 61 as well as the crankshaft 24. In doing so, the ECM 11 can adjust the open/close timing of the intake valve 18.

The exhaust VCT mechanism 82 is situated between the camshaft 81 and the timing pulley 62. Like the intake cam phaser 72, the exhaust cam phaser 82 uses the camshaft 81 and timing pulley 62 as external gears and interconnects them via an intermediate helical gear. Through drive circuit 47, the ECM 11 controls a valve 83 that affects the hydraulic pressure acting upon this helical gear. It also uses feedback from sensors 56 and 58 to monitor the angular relationship between the crankshaft 24 and the exhaust camshaft 81. By changing the hydraulic pressure via valve 83, the ECM 11 can move this helical gear axially, and thus alter the angular relationship between the exhaust camshaft 81 and the timing pulley 62 as well as the crankshaft 24. In doing so, the ECM 11 can adjust the open/close timing of the exhaust valve 21.

Using VCT mechanisms, the open/close timing of the intake and exhaust valves 18 and 21 can be optimized to improve the overall performance of the engine 1. In dual overhead cam (DOHC) engines, there are four possible types of VCT: (1) phasing only the intake cam (Intake Only); (2) phasing only the exhaust cam (Exhaust Only); (3) phasing the intake and exhaust cams equally (Dual Equal); and (4) phasing the intake and exhaust cams independently (Dual Independent). The Dual Equal strategy is also applicable to single overhead cam (SOHC) engines.

It is well known that use of a VCT mechanism on only the intake camshaft 71 improves engine operation. This involves varying the open/close timing of the intake valve 18, as compared to standard cam timing, when the engine 1 is operating at part load. For example, by advancing the

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opening of the intake valve 18, the valve overlap is extended into the exhaust stroke. This means that the intake valve 18 starts to open near the end of the exhaust stroke. Viewing FIG. 1, this allows the piston 20, in its upward exhaust stroke, to push a small amount of the exhaust gases back into the intake duct 12. On the subsequent (downward) intake stroke, this exhaust gas is then re-ingested into the cylinder 13 for combustion with the fuel-air mixture. By advancing the closing of the intake valve 18, the intake valve 18 closes earlier in the compression stroke. This means that less of the fuel-air mixture is pushed back into the intake duct 12, thereby enabling more power to be produced during combustion.

The benefits of intake cam phasing are well known. First, it reduces NO_x emissions. This is due to what is referred to as increased residual dilution. The re-ingested exhaust gases (i.e., the diluent) lowers the temperature at which combustion occurs, thereby reducing the amount of NO_x emissions. The extent of the NO_x reduction depends on the load and speed of the engine. Second, it reduces HC emissions. The last portion of the exhaust gases ejected from cylinder 13 during the exhaust stroke is rich in unburned HC. It is this portion of the exhaust gases that is re-ingested during the intake stroke and subsequently burned. Advanced intake cam timing also increases the torque output by the engine at medium to high speeds, improves fuel economy and enables the engine to be operated more stably at idle.

It is also well known that use of a VCT mechanism on only the exhaust camshaft 81 has a significant effect on emissions. This involves varying the close/open timing of the exhaust valve 21, as compared to standard cam timing, when the engine 1 is operating at part load. For example, by delaying the closing of the exhaust valve 21, the valve overlap is extended into the intake stroke. This means that the exhaust valve 21 stays open at the start of the intake stroke. Viewing FIG. 1, this allows the piston 20, in its downward intake stroke, to draw a small amount of the exhaust gases from the exhaust manifold 25 not only back into the cylinder 13 but also into the intake duct 12 due to vacuum. Along with the fuel-air mixture, this exhaust gas is then burned in the combustion chamber 19 during the combustion cycle.

The benefits of exhaust cam phasing are well known. First, it also reduces NO_x emissions due to increased residual dilution. Second, HC emissions are reduced because the HC-rich portion of the exhaust gases is drawn back into the cylinder 13. Delayed exhaust cam timing also improves fuel economy and enables the engine to be operated more stably at idle. In addition, exhaust cam phasing can be used as a substitute for an external exhaust gas recirculation (EGR) system, as it performs the same function. The cost of equipping a vehicle with an exhaust cam phaser can be less than that for a conventional EGR system.

U.S. Pat. No. 5,713,317 to Yoshioka describes a method of controlling a VCT mechanism through which to vary the open/close timing of a valve. It purports to optimize the valve timing so as to improve the output of the engine at high altitudes while it is operating under high loads. It also purports to reduce the fuel consumption and emissions of the engine at high altitudes as it operates under low to medium loads. The method essentially controls the amount of residual dilution (i.e., re-ingested exhaust gases). In doing so, the Yoshioka reference teaches advance of the intake valve only, in a way that attempts to compensate for the effects of altitude.

U.S. Pat. No. 5,755,202 to Stefanopoulou et al. teaches the use of a Dual Equal VCT strategy on a vehicle equipped

with ETC and a torque based engine control system. According to the method, the range of torque that can be demanded of the engine is divided into five regions, namely, negligible, small, moderate, high and maximum. The engine control system chooses the particular cam timing schedule to use according to the region into which the actual torque demand falls. For example, in the negligible torque region, standard cam timing is used to maintain the engine at, a stable idle. In the small torque region, the timing scheme falls between standard and fully retarded cam phasing, with the exact timing dependent on the magnitude of the torque demand. In the moderate torque region, fully retarded cam timing is used, and the throttle position is adjusted to meet the torque demand. It is in this region that the benefits of VCT (e.g., reduced feed gas (NO_x and HC) emissions and improved fuel economy) are most fully realized. In the high torque region, the timing scheme falls between fully retarded and standard cam phasing, with the exact timing dependent on the magnitude of the torque demand. Here, the throttle position is held constant. It is in this region that the cam scheduling is relaxed to best meet the torque demand. In the maximum torque region, the schedule reverts to standard cam timing. This enables the engine control system to satisfy the torque demanded by the driver.

The disadvantage of the Stefanopoulou et al. system lies in the way it selects the cam timing mode according to the region into which the actual torque demand falls. Such transitions between modes can lead not only to discontinuities during throttle transient maneuvers but also to increased calibration time. It is therefore desirable to devise a strategy that employs variable cam timing in a way that assures smoother transitions in torque while achieving even lower feed gas emissions and increased torque.

OBJECTIVES OF THE INVENTION

It is, therefore, an objective of the invention to provide a method of varying the phasing of either the intake valves or the exhaust valves or both to yield smooth transitions in torque while achieving low feed gas emissions and increased torque.

Another objective is to provide a method of using variable cam timing on an engine equipped with electronic throttle control to vary the phasing of the intake and/or exhaust valves to achieve lower feed gas emissions and improved fuel economy while delivering as closely as possible the desired torque.

A further objective is to provide a method of adjusting the position of the throttle valve, the amount of fuel injected by the fuel injector(s), the timing of the spark and the phasing of the intake and/or exhaust cams to deliver the desired torque according to the conditions under which the vehicle is operating.

Still another objective is to enable an internal combustion engine of a motor vehicle to adjust automatically the torque output by the engine in response to variations in accessory loads.

Yet another objective is to enable an internal combustion engine to meet regulations governing feed gas emissions without the use of an external exhaust gas recirculation (EGR) system.

An additional objective is to enable an engine to operate more efficiently over a wider range of torque levels.

Another objective is to enable an engine to operate at lower combustion temperatures, as compared to most prior art engines.

A further objective is to embody the valve timing algorithms for an engine in a ROM, PROM or other type of

memory chip and thereby make it easy to update the valve timing schedule merely by removing the old chip from the ECM and replacing it with a new chip in which the new algorithms are stored.

In addition to the objectives and advantages listed above, various other objectives and advantages of the invention will become more readily apparent to persons skilled in the relevant art from a reading of the other sections of this document. The other objectives and advantages will become particularly apparent when the detailed description is considered along with the drawings and claims presented herein.

SUMMARY OF THE INVENTION

The foregoing objectives and advantages are attained by the various embodiments of the invention summarized below.

In a preferred embodiment in which an air-lead strategy is employed, the invention provides a method of improving the operating characteristics of an internal combustion engine equipped with an electronically controlled throttle, a fuel injector and a variable valve timing mechanism. The variable valve timing mechanism is used to control the timing of the intake valves and/or the exhaust valves. The method includes the repeated performance of at least the following steps. The desired torque requested of the engine is ascertained. The actual air mass flow into the intake manifold is determined based on at least one of the following variables: the absolute pressure in the intake manifold, the indicated mean effective pressure in the intake manifold, the speed of the engine, the intake air temperature, the coolant temperature, the barometric pressure, the input from a mass air flow sensor and the volumetric efficiency of the engine. The desired fuel flow is calculated based on at least the actual air mass flow according to a desired air-fuel ratio. The per-cylinder fuel amount is determined based on the desired fuel flow and the number of cylinders in the engine. The injector pulse width with which to activate the fuel injector is calculated based on the per-cylinder fuel amount. The desired air flow is calculated based on the desired fuel flow according to the desired air-fuel ratio. The actual position of the throttle is monitored. The desired position of the throttle is determined as a function of at least two of the following variables: the desired air flow, the absolute pressure in the intake manifold, the engine speed and the volumetric efficiency of the engine. The throttle is adjusted to the desired position with regard to the actual position of the throttle. The desired position for the variable valve timing mechanism is determined as a function of at least the speed of the engine and the load of the engine. The variable valve timing mechanism is directed to move to the desired position. This causes the timing at which the intake and/or exhaust valves open and close to operate according to a schedule that improves the operating characteristics of the engine. The actual position of the variable valve timing mechanism is monitored. The volumetric efficiency is determined as a function of the speed and load of the engine and one of the desired and actual positions of the variable valve timing mechanism. The dilution occurring within the cylinders is calculated based on the engine speed, the load of the engine and one of the desired and actual positions of the variable valve timing mechanism.

In an alternative embodiment in which a fuel-lead strategy is employed, the method includes the repeated performance of at least the following steps. The desired torque requested of the engine is ascertained. The desired fuel flow is calcu-

lated based not only on an amount necessary for the engine to provide the desired torque but also on the dilution occurring within the cylinders. The per-cylinder fuel amount is determined based on the desired fuel flow and the number of cylinders in the engine. The injector pulse width with which to activate the fuel injector is calculated based on the per-cylinder fuel amount. The desired air flow is calculated based on the desired fuel flow according to a desired air-fuel ratio. The actual position of the throttle is monitored. The desired position of the throttle is determined as a function of at least two of the following variables: the desired air flow, the absolute pressure in the intake manifold, the engine speed and the volumetric efficiency. The throttle is adjusted to the desired position with regard to the actual position of the throttle. The desired position for the variable valve timing mechanism is determined as a function of at least the engine speed and the engine load. The variable valve timing mechanism is directed to move to the desired position. This causes the timing at which the intake and/or exhaust valves open and close to operate according to a schedule that improves the operating characteristics of the engine. The actual position of the variable valve timing mechanism is monitored. The volumetric efficiency is determined as a function of the speed and load of the engine and one of the desired and actual positions of the variable valve timing mechanism. The dilution occurring within the cylinders is calculated based on the engine speed, the engine load and one of the desired and actual positions of the variable valve timing mechanism.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a diagram of one cylinder of an electronically controlled multi-cylinder engine showing various associated components such as the throttle body, the fuel injector, the spark timing system, the intake and exhaust valley, the variable cam timing (VCT) apparatus and the electronic control module (ECM).

FIG. 2 is a diagram of the internal architecture of the ECM shown in FIG. 1, showing the various sensors from which it receives data and the various external devices under its control.

FIG. 3 is a flowchart of a method, according to the invention, through which the operating characteristics of an engine are improved using a fuel-lead strategy.

FIG. 4 is a flowchart of the method, according to the invention, through which the operating characteristics of an engine are improved using an air-lead strategy.

FIG. 5 depicts a process of ascertaining the amount of torque demanded by the driver and by the conditions under which the vehicle is operating.

FIG. 6 is a table of values of intake cam phaser position in which each value is a function of engine speed and engine load.

FIG. 7 is a graph of a temperature-correction curve T_c whose values are plotted against engine temperature on the horizontal axis and the correction factor on the vertical axis.

FIG. 8 is a graph of a pressure-correction curve P_c whose values are plotted against ambient pressure on the horizontal axis and the correction factor on the vertical axis.

FIG. 9 is a table of values of exhaust cam phaser position in which each value is a function of engine speed and engine load.

DETAILED DESCRIPTION OF THE INVENTION

FIGS. 3 and 4 illustrate in flowchart form two embodiments of the invention, namely, a method that can be used

to improve the operating characteristics of an internal combustion engine. More specifically, the method enables the phasing of the intake and/or exhaust valves to be varied so as to achieve lower feed gas emissions, improved fuel economy and greater stability at idle, while improving the ability of the engine to deliver the desired torque. The method, generally designated **100**, is designed for a torque based control system of the type used to control an engine equipped with electronic throttle control (ETC) and a cam phaser on either the intake camshaft or the exhaust camshaft or both.

FIG. 3 depicts one embodiment of the invention, namely, using the method to implement a fuel-lead strategy. FIG. 4 illustrates the presently preferred embodiment, i.e., using the method to implement an air-lead strategy. In either embodiment, the method **100** comprises a series of generalized steps. Using known techniques, a person of ordinary skill in the programming arts has the expertise to reduce the method to a corresponding set of instructions and to incorporate those instructions within the programming code normally executed by the control system for the engine. It should be apparent that the method could be carried out by a subroutine that is called and executed at frequent, predetermined time intervals. All steps of the invention need not necessarily be carried out in the order shown or described.

The invention is described below as if intended for the environment set out in the background section of this document. Nevertheless, it should be apparent from a reading of this document that the invention may be implemented in a variety of engine control systems, even ones unrelated to the automotive industry. The invention is presented in this context not to limit the scope of the claims set forth below but merely to simplify the description, and thus the understanding, of the invention.

In either embodiment, step **110** of the method **100** involves ascertaining the desired torque, i.e., the amount of torque demanded by the driver and by the conditions under which the vehicle is operating. In a torque based engine control system, the ECM **11** can derive the desired torque generally according to the process depicted in FIG. 5. As indicated by block **111**, the ECM **11** determines the amount of torque sought by the driver using the pedal, TPS and RPM sensors **8**, **9** and **33** as well as other inputs. In its torque calculations, the ECM **11** also adds the torque demands of the onboard cruise control and traction control systems, if any, as shown in block **112**. The ECM **11** also includes the torque lost to various other loads and accessories (e.g., the AC compressor, cooling fan, alternator/generator, etc), as suggested by blocks **113** and **114**. Block **115** reveals that the ECM **11** also uses the VSS and RPM sensors **31** and **33** to ascertain the torque lost to the transmission, the amount of the loss being dependent on the gear in which the transmission is then operating. In calculating the desired torque in block **116**, the ECM **11** will thus preferably be programmed to compensate not only for the torque inevitably lost to friction but also for the torque lost to additional loads. Optionally, the desired torque could be calculated without compensating for one or more of the aforementioned torque losses.

As the description below indicates, some of the other steps of the method **100** depend on whether the engine is being controlled according to the fuel-lead strategy or the air-lead strategy.

FIG. 3 illustrates the embodiment of the method **100** in which the fuel-lead strategy is realized. It is preferred that the fuel-lead strategy be implemented so that the engine **1** is

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operated in a lean mode. Because of the excess oxygen during lean operation, the production of carbon monoxide (CO) will be greatly reduced. It is preferred that the engine 1 be operated sufficiently lean with the combustion temperature reduced so as to lower the amount of nitrogen oxides that are produced.

Step 120 involves calculating the desired fuel flow, i.e., the amount of fuel that is deemed necessary to provide the desired torque. Using the desired torque and data from various sensors through which it receives feedback concerning the operating conditions of the engine 1, the ECM 11 determines the desired fuel flow. In doing so, however, the ECM 11 also takes into account the residual dilution. As explained further below, this parameter may be either the actual amount of dilution currently occurring within the cylinders by virtue of the intake and/or exhaust valves 18 and 21 or the desired amount of dilution. It should be apparent that the desired fuel flow is a quantity that is dependent on, inter alia, the displacement of and the number of cylinders within the engine 1. The use of such constants and other variables in the programming arts is, of course, well known in the automotive industry.

Step 130 involves calculating the desired air flow. For the fuel-lead strategy, the desired amount of air is determined using the desired fuel flow previously calculated in step 120. In particular, this calculation can be expressed by the equation:

$$\text{DesiredAirFlow} = \text{DesiredAir-FuelRatio} / \text{DesiredFuelFlow}$$

where the desired air-fuel ratio is a known parameter.

Step 140 represents calculation of the desired throttle position, i.e., the position of throttle plate 6 that corresponds to desired air flow calculated in step 130. Using the desired air flow and current operating conditions such as those obtained from the MAP sensor 30, RPM sensor 33 and, if available, BARO sensor 35, the ECM 11 determines the desired throttle position. The ECM 11 also takes into account the volumetric efficiency of the engine, as explained further below. In determining the desired throttle position, the ECM 11 will preferably use a lookup table. The use of such lookup tables is well known in the programming arts. This lookup table will contain an array of throttle position values as a function of at least two variables, such as the desired air flow, the manifold absolute pressure, the engine speed and, optionally, barometric pressure. This relationship is characterized by the equation:

$$\text{DesiredThrottlePosition} = f(\text{DesiredAirFlow}, \text{MAP}, \text{RPM}, \text{etc.}).$$

Operating according to the algorithms into which the method is incorporated, the ECM 11 will regularly select from this table the value of desired throttle position that corresponds to the desired air flow and the current operating conditions.

As it moves the throttle plate 6 to the desired throttle position via the ETC system, the ECM 11 regularly receives feedback from the TPS sensor 9 as to the actual throttle position. Step 150 shows that the ECM 11 uses this feedback to adjust the position of the throttle plate 6 according to the method 100.

Step 320 involves determining the amount of fuel that is required on a per cylinder basis by using the desired fuel flow that was calculated in step 120. This is a simple calculation based on the number of cylinders in the engine 1. Using the required fuel per cylinder, the ECM 11 then determines the corresponding injector pulse width, the signal that is used to activate each injector(s) 17 according to the timing sequence. This is represented by step 180.

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In a DOHC engine in which each -camshaft is equipped with a continuously variable cam timing mechanism, step 130 pertains to the calculation of the desired positions for both the intake and exhaust cam phasers 72 and 82. Consider first the intake valves. Using the current operating conditions such as those indicative of engine load and those obtained from the CTS sensor 29, RPM sensor 33 and, if available, BARO sensor 35, the ECM 11 determines the desired position for the intake cam phaser 72. This relationship is characterized by the equation:

$$\text{DesiredIntakeCamPosition} = f(\text{EngineSpeed}, \text{EngineLoad}) * f(\text{CoolantTemp}) * f(\text{BarometricPressEngineLoad}).$$

The current engine load can be calculated according to known practice from the TPS sensor 9, the MAP sensor 30, the indicated mean effective pressure (IMEP) in the intake manifold 7, engine torque, and, if available, MAF sensor 36. IMEP is a measure of engine load, one that can be ascertained according to well known practice. (It should be noted that engine load is a parameter that is engine/calibration specific. Based on the delays inherent to engine control systems, instabilities could occur in some engine configurations if the actual, rather than desired, engine load is used. This is due to the effect that cam phasing has on engine load.)

In determining the desired intake cam phaser position, the ECM 11 will preferably use a lookup table. This lookup table will contain an array of values of intake cam phaser position as a function of at least two variables, such as engine speed and engine load. FIG. 6 illustrates an example of such a lookup table. Engine load values are listed vertically at left, and engine speed values are listed horizontally at the top of the table. The engine load values are expressed in terms of manifold absolute pressure, IMEP, and engine torque as a percentage of the torque that would be obtainable at wide open throttle (WOT). In the body of the table, the values of intake cam phaser position are expressed in degrees of advance. According to its algorithms, the ECM 11 will regularly select from this table the value of intake cam phaser position that most closely corresponds to the current engine speed and load conditions. For improved accuracy, interpolation between table values will preferably be employed.

The desired value of intake cam phaser position will preferably be corrected to accommodate the effects of engine temperature and atmospheric pressure. FIG. 7 illustrates an example of a temperature-correction curve T_c . It is plotted against engine temperature on the horizontal axis and against the correction factor on the vertical axis. It increases monotonically, preferably linearly, until temperature T_M is reached, at which point the curve T_c has its maximum value, unity. When the temperature of the engine 1 is less than or equal to temperature T_1 , intake cam phasing is not desired. When the engine temperature lies above T_1 , temperature-correction is required. The desired intake cam phaser position is then made equal to the value obtained from the table lookup multiplied by the temperature-correction value taken from curve T_c . It is presently preferred that the minimum temperature (T_1) be approximately 20° C. and that the maximum temperature (T_M) be about 110° C.

FIG. 8 illustrates an example of a pressure-correction curve P_c . It is plotted against ambient pressure on the horizontal axis and against the correction factor on the vertical axis. It increases monotonically, preferably linearly, until pressure P_M is reached, at which point the curve P_c has its maximum value, unity. When atmospheric pressure is

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less than or equal to pressure P_1 , intake cam phasing is not desired. When the pressure lies above P_1 , pressure-correction is required. The desired intake cam phaser position is then made equal to the value obtained from the table lookup multiplied not only by the temperature-correction value taken from curve T_c but also by the pressure-correction value taken from curve P_c . It is preferred that the minimum pressure (P_1) be approximately 80 KPa and that the maximum pressure (P_M) be about 105 KPa. The purpose of this correction is to eliminate or reduce cam phasing when the engine is operated at high altitudes or while atmospheric pressure is reduced due to weather.

Once it calculates the desired value of intake cam position, the ECM 11 directs the intake cam phaser 72 through drive circuit 46 to alter the angular relationship between the intake camshaft 71 and the crankshaft 24 accordingly. In doing so, the ECM 11 adjusts the open/close timing of the intake valves 18, and thereby enables the engine 1 to obtain the desired operating characteristics in regards to the intake camshaft 71.

Consider now the exhaust valves. Using the current operating conditions such as those indicative of engine load and those obtained from the CTS sensor 29, RPM sensor 33 and, if available, BARO sensor 35, the ECM 11 determines the desired position for the exhaust cam phaser 82. This relationship can be expressed as:

$$\text{DesiredExhaustCamPosition} = f(\text{EngineSpeed}, \text{EngineLoad}) * f(\text{CoolantTemp}) * f(\text{BarometricPress}, \text{EngineLoad}).$$

In determining the desired exhaust cam phaser position, the ECM 11 will preferably use a lookup table. This lookup table will contain an array of values of exhaust cam phaser position as a function of at least two variables, such as engine speed and engine load. FIG. 9 illustrates such a lookup table, one similar to that used in calculation of the desired intake cam phaser position. In the body of this table, however, the values of exhaust cam phaser position are expressed in degrees of retard. According to its algorithms, the ECM 11 will regularly select from this table the value of exhaust cam phaser position that most closely corresponds to the current engine speed and load conditions. For improved accuracy, interpolation between table values will preferably be employed.

The desired value of exhaust cam phaser position will preferably be corrected to accommodate the effects of engine temperature and atmospheric pressure. Using FIGS. 7 and 8, the temperature and pressure-correction factors can be ascertained for the exhaust cam phaser 82 in the manner previously described for the intake cam phaser 72. The desired exhaust cam phaser position would then be equal to the value obtained from the table lookup multiplied by the temperature-correction value taken from curve T_c and by the pressure-correction value taken from curve P_c .

Once it calculates the desired value of exhaust cam position, the ECM 11 directs the exhaust cam phaser 82 through drive circuit 47 to alter the angular relationship between the exhaust camshaft 81 and the crankshaft 24 accordingly. In doing so, the ECM 11 adjusts the open/close timing of the exhaust valves 21, and thereby enables the engine 1 to obtain the desired operating characteristics in regards to the exhaust camshaft 81.

Step 200 involves determining the actual positions of the intake and exhaust camshafts 71 and 81. As noted earlier, the ECM 11 uses feedback from sensors 56 and 57 to monitor the angular relationship between the crankshaft 24 and the intake camshaft 71. Similarly, it uses feedback from sensors 56 and 58 to monitor the angular relationship between the

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exhaust camshaft 81 and crankshaft 24. It is thus through the crankshaft and camshaft sensors 56, 57 and 58 that the ECM 11 ascertains the actual positions of the intake and exhaust camshafts 71 and 81.

It is well known that a change in the positions of the intake and/or exhaust cam phasers 72 and 82 will change the volumetric efficiency of the engine 1. Consequently, step 210 involves calculating the new volumetric efficiency based on the desired positions of the camshafts. More specifically, the ECM 11 can calculate the new volumetric efficiency based on the engine speed, the engine load and the desired positions of the intake and exhaust camshafts 71 and 81. This calculation is characterized by the equation:

$$\text{VolumetricEfficiency} = f(\text{ExhaustCamPosition}, \text{IntakeCamPosition}, \text{EngineSpeed}, \text{EngineLoad} \text{ etc.}).$$

Step 220 indicates that the POM 11 can calculate the volumetric efficiency based on the actual, rather than desired, positions of the camshafts. In this calculation, the engine load parameter is preferably derived from the actual engine torque. For the fuel-lead strategy, the desired throttle position as calculated in step 140 can be determined using the volumetric efficiency resulting from either step 210 or 220.

In either case, the volumetric efficiency will preferably be determined through use of a lookup table. This table will contain an array of values of volumetric efficiency as a function of several variables, such as the cam position(s), engine speed and engine load. According to its algorithms, the ECM 11 will regularly select from the table the value of volumetric efficiency that most closely corresponds to the current operating conditions. Interpolation may, of course, be used to improve accuracy.

Step 230 involves calculating the dilution, i.e., the extent to which the exhaust gases should be or are being re-ingested into the cylinder(s) 13. Using data obtained from the relevant sensors, the ECM 11 can determine the desired amount of dilution from the engine speed, engine load, and the desired positions of the intake and exhaust camshafts 71 and 81. The desired camshaft positions are made available by step 190. Similarly, the ECM 11 can determine the actual amount of dilution from the engine speed, engine load, and the actual positions of the intake and exhaust camshafts. The actual camshaft positions are made available by step 200. As shown in step 120, the ECM 11 compensates for the effects of the dilution in ascertaining the desired fuel flow. The ECM 11 will also compensate for the effects of the dilution in calculating the proper spark timing.

FIG. 4 depicts the preferred embodiment of the invention, namely, using the method 100 to implement the air-lead strategy. This strategy enables the engine 1 to operate in a substantially stoichiometric manner, and permits the catalytic converter 27 to operate at maximum efficiency. Operating according to this strategy, the ECM 11 tries to keep the air-fuel ratio at stoichiometry and, in doing so, overrides the fuel command. Although the throttle plate 6 takes a finite time to move to the desired position, the intake manifold 7 takes a relatively long time to draw the air past the throttle plate 6 and thus to fill. Consequently, in the air-lead strategy, step 360 involves calculating the actual amount of air that is being delivered to the cylinders (i.e., the flow of air mass into the intake manifold 7). The ECM 11 can calculate this air mass flow from the data it receives from sensors such as the MAP, RPM, IAT and CTS sensors 30, 33, 28 and 29, and, if available, the BARO and MAF sensors 35 and 36. The ECM 11 also takes into account the volumetric efficiency of the engine, as noted below. In calculating the actual air flow,

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the ECM 11 may also use the indicated mean effective pressure (IMEP) in the combustion chamber 19.

Based on the desired air-fuel ratio, various transient fuel effects and the actual air flow ascertained in step 360, the desired fuel flow can be calculated in step 120. In the air-lead strategy, however, the desired fuel flow will thus be the amount of fuel necessary to achieve stoichiometry. This can be calculated according to the equation:

$$\text{DesiredFuelFlow} = \text{DesiredAir} \cdot \text{FuelRatio} / \text{ActualAirFlow}.$$

Step 370 involves determining the amount of fuel that is required on a per cylinder basis. As with the fuel-lead strategy, this is a simple calculation using the number of cylinders in the engine 1. Using the required fuel per cylinder, the ECM 11 then determines the corresponding injector pulse width for the air-lead strategy. This is represented by step 180.

The ECM 11 regularly updates the desired throttle position according to the changing operating conditions, and moves the throttle plate 6 accordingly via the ETC system. It also monitors the actual throttle position using the feedback from the TPS sensor 9, as indicated by step 150. This enables the ECM 11 to determine the actual air flow at any given instant, according to step 360. As indicated by steps 120, 370 and 180, the ECM 11 thus regularly adjusts the injector pulse width in accordance with the actual (instantaneous) air flow.

In using the method 100 to implement the air-lead strategy, steps 190, 200, 210, 220 and 230 are performed in essentially the same way as described above in connection with the fuel-lead strategy. This includes correcting the desired values of the intake and exhaust cam phaser positions to accommodate the effects of engine temperature and atmospheric pressure. Once it calculates the desired value of intake cam position, the ECM 11 directs the intake cam phaser 72 through drive circuit 46 to alter the angular relationship between the intake camshaft 71 and the crankshaft 24 accordingly. In doing so, the ECM 11 adjusts the open/close timing of the intake valves 18, and thereby enables the engine 1 to obtain the desired operating characteristics in regards to the intake camshaft 71. Likewise, after calculating the desired value of exhaust cam position, the ECM 11 directs the exhaust cam phaser 82 through drive circuit 47 to alter the angular relationship between the exhaust camshaft 81 and crankshaft 24. By doing so, the ECM 11 adjusts the open/close timing of the exhaust valves 21, and thereby enables the engine 1 to obtain the desired operating characteristics in regards to the exhaust camshaft 81.

It is again noted that the volumetric efficiency of the engine 1 changes whenever the positions of the intake and/or exhaust cam phasers 72 and 82 change. Step 210 calculates the volumetric efficiency based on the desired positions of the intake and exhaust camshafts 71 and 81. Step 220 calculates the volumetric efficiency based on the actual positions of the camshafts. For the air-lead strategy, the desired throttle position as calculated in step 140 is preferably determined using the volumetric efficiency resulting from step 210. The air mass flow calculated in step 360, however, is preferably determined using the volumetric efficiency resulting from step 220.

In either of its embodiments, the method 100 improves the operating characteristics of the engine 1. Generally stated, whenever a change is requested in the amount of torque, the method 100 responds by adjusting the amount of fuel flow, the spark timing, the position of the throttle plate 6, and/or the positions of the intake and/or exhaust cam phasers 72

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and 82 to deliver the desired torque. The method 100 adjusts those operating parameters in a way that improves the ability of the engine to deliver the desired torque, produce less feed gas emissions, idle more stably, and consume less fuel, as compared to prior art schemes.

Although the method 100 is described and depicted herein as applying to both the intake and exhaust camshafts 71 and 81, it should be apparent that it can be applied to either alone. In an Intake Only system, the ECM 11 will respond to a change in torque by adjusting the fuel flow, the air flow, the external EGR system and the position of the intake cam phaser 72 to settings that allow the engine to deliver as closely as possible the desired torque while still providing low feed gas emissions. In an Exhaust Only system, the ECM 11 will respond to a change in torque by adjusting the fuel flow, the air flow and the position of the exhaust cam phaser 82 to settings that enable the engine to deliver improved fuel economy and lower emissions while still maintaining driveability.

The invention also contemplates that the intake and exhaust valves could be electronically driven, as opposed to being camshaft driven. Consequently, the term variable valve timing encompasses both mechanically and electronically controlled valves, and may apply to the phasing of either the intake valves or the exhaust valves or both. In addition, the position of a variable valve timing mechanism may be construed to mean the position of a camshaft and its associated cams.

The presently preferred and various alternative embodiments for carrying out the invention have been set forth in detail according to the Patent Act. Persons of ordinary skill in the art to which this invention pertains may nevertheless recognize alternative ways of practicing the invention without departing from the spirit and scope of the following claims. Persons of such skill will also recognize that the foregoing description is illustrative of the invention and not intended to limit any of the ensuing claims to any particular narrow interpretation.

What is claimed is:

1. A method of improving the operating characteristics of an internal combustion engine equipped with an electronically controlled throttle, a fuel injector and a variable valve timing mechanism, said variable valve timing mechanism being used to control the timing of at least one of intake valves and exhaust valves of said engine, said method comprising the repeated performance of at least the following steps:

- (a) ascertaining a desired torque requested of said engine;
- (b) calculating a desired fuel flow based on an amount necessary for said engine to provide said desired torque and on dilution occurring within cylinders of said engine;
- (c) determining a per-cylinder fuel amount based on said desired fuel flow and the number of said cylinders in said engine;
- (d) calculating an injector pulse width with which to activate said fuel injector based on said per-cylinder fuel amount;
- (e) calculating a desired air flow based on said desired fuel flow according to a desired air-fuel ratio;
- (f) monitoring an actual position of said throttle;
- (g) determining a desired position of said throttle as a function of at least two of said desired air flow, an absolute pressure in an intake manifold of said engine, a speed of said engine and a volumetric efficiency of said engine;

(h) adjusting said throttle to said desired position with regard to said actual position of said throttle;

(i) determining a desired position for said variable valve timing mechanism as a function of at least said speed of said engine and a load of said engine;

(j) directing said variable valve timing mechanism to move to said desired position thus causing said timing at which said at least one of said intake valves and said exhaust valves open and close to operate according to a schedule that improves said operating characteristics of said engine;

(k) monitoring an actual position of said variable valve timing mechanism;

(l) determining said volumetric efficiency of said engine as a function of said speed of said engine, said load of said engine and one of said desired and actual positions of said variable valve timing mechanism; and

(m) calculating said dilution occurring within said cylinders of said engine based on said speed of said engine, said load of said engine and one of said desired and actual positions of said variable valve timing mechanism.

2. The method, as claimed in claim 1, wherein said desired position of said variable valve timing mechanism is corrected to accommodate the effects of at least one of a temperature of said engine and a barometric pressure.

3. The method, as claimed in claim 1, wherein said desired torque includes the torque demanded by an operator of said

engine and the torque consumed by at least one of a transmission, a cruise control system, a traction control system and at least one of various other loads on said engine.

4. The method, as claimed in claim 1, wherein said desired position of said throttle is also a function of barometric pressure.

5. The method, as claimed in claim 1, wherein said variable valve timing mechanism includes at least one of an intake cam phaser and an exhaust cam phaser.

6. The method, as claimed in claim 5, wherein said desired position of said variable valve timing mechanism is corrected to accommodate the effects of at least one of a temperature of said engine and a barometric pressure.

7. The method, as claimed in claim 1, wherein said variable valve timing mechanism is an intake cam phaser.

8. The method, as claimed in claim 1, wherein said variable valve timing mechanism is an exhaust cam phaser.

9. The method, as claimed in claim 1, wherein said load of said engine is determined from at least one of said position of said throttle, said absolute pressure in said intake manifold, an indicated mean effective pressure in said intake manifold, actual torque of said engine and flow of air mass into said intake manifold.

10. The method, as claimed in claim 1, wherein said desired air flow is sufficient to enable said engine to operate according to a lean air-fuel mixture whereby production of carbon monoxide and nitrogen oxide(s) is minimized.

* * * * *

Torque Management of Engines with Variable Cam Timing

Mrdjan Jankovic, Florian Frischmuth, Anna Stefanopoulou, and Jeffrey A. Cook

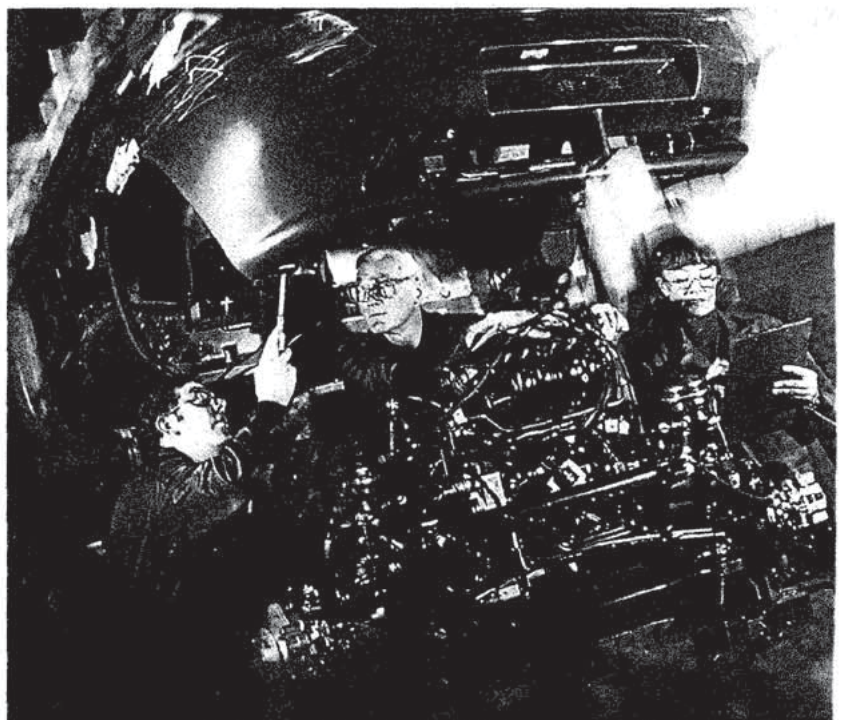
Automotive powertrain control systems are subject to diverse and, usually, conflicting requirements. In particular, exhaust emissions must meet increasingly stringent Federal and California standards while fuel economy must meet customer expectations and contribute to Federally mandated Corporate Average Fuel Economy (CAFE) imperatives. Neither vehicle performance nor reliability may be compromised in attaining these goals (emission performance must be guaranteed for 100,000 miles). Of course, all these objectives must be achieved at the lowest possible cost and with the minimum number of sensors and actuators.

The conventional method of reducing feedgas oxides of nitrogen (NO_x) emissions is by the application of exhaust gas recirculation (EGR). Formation of NO_x during the combustion process is influenced by air-fuel ratio (*A/F*) and temperature. These variables are typically adjusted by directing exhaust gas from the high pressure exhaust manifold via a pulse-width modulated valve to the low pressure induction system where it serves to reduce the combustion temperature and dilute the air-fuel mixture in the cylinder. A desirable side-effect is that the presence of inert exhaust gas raises the intake manifold pressure reducing the pumping losses and improving fuel economy. Although effective, the dynamics associated with this system (transport delay, valve and intake manifold dynamics) can result in vehicle performance deterioration.

The variable cam timing (VCT) system with which this paper is concerned addresses both drivability and emissions performance by utilizing an electro-hydraulic mechanism to rotate the camshaft relative to the crankshaft in order to retard the cam timing with respect to the intake and exhaust strokes of the engine. Variable cam timing operation is illustrated in Fig. 1. By retarding the exhaust valve closing further

into the intake stroke, more exhaust gas is drawn into the cylinder providing internal exhaust gas recirculation. In this manner, the amount of residual gas trapped in the cylinder at the end of the exhaust stroke is controlled by cam timing, suppressing NO_x formation and reducing the pumping losses [1], [9], [14]. Furthermore, this residual contains some unburned hydrocarbons; consequently, retaining it in the cylinder through two combustion cycles also reduces hydrocarbon emissions [8].

In addition to the reduction of NO_x and HC emissions, variable cam timing permits the engine designer to optimize cam timing over a wide range of engine operating conditions, providing both good idle quality (minimal overlap between the intake and exhaust events) and improved wide-open throttle performance (maximum inducted charge). In this article, we will assume that the camshaft position is continuously variable to



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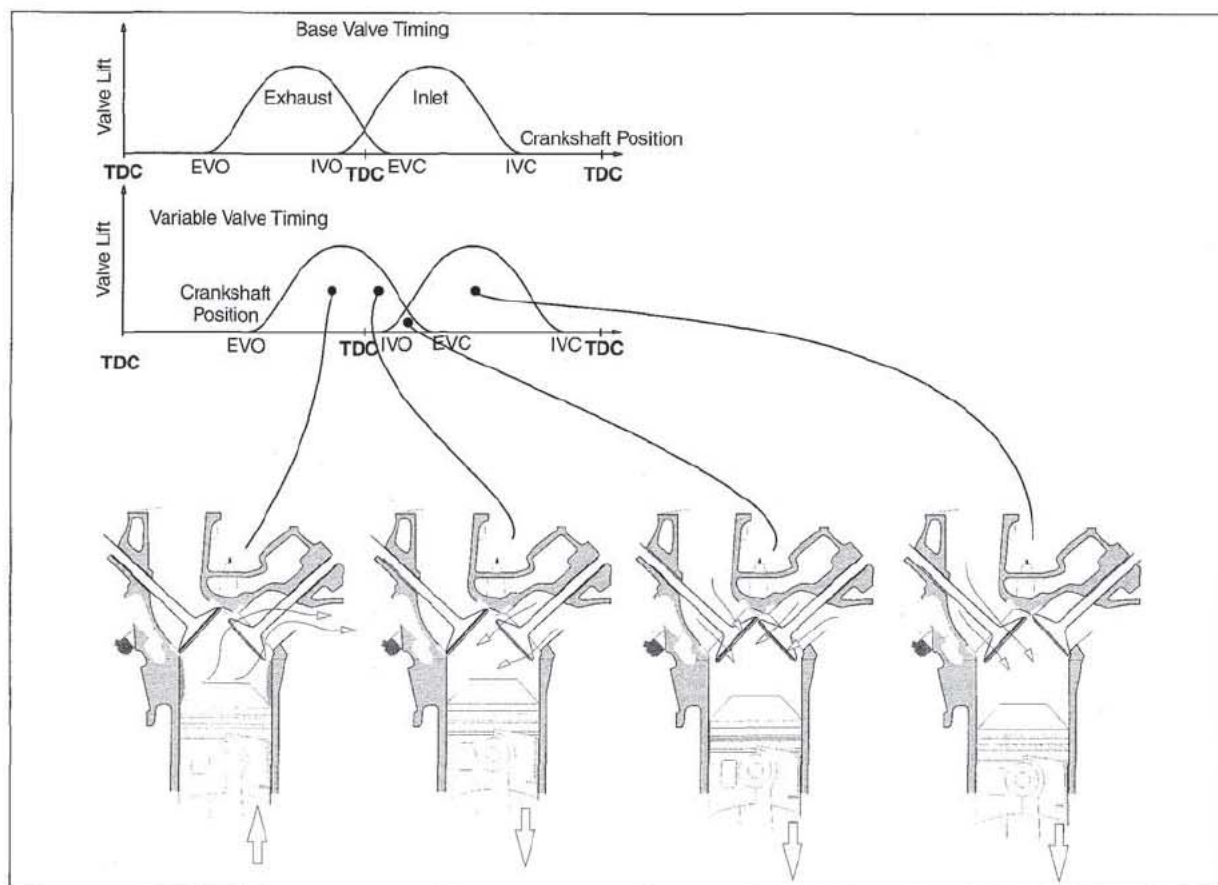


Fig. 1. Valve lift profiles of conventional and VCT engines. By retarding the cam phasing, the exhaust valve stays open during the intake event for a longer time period, retaining otherwise unburned HC and reducing the combustion temperature due to the dilution of the inert gas.

achieve the full benefits in emissions and torque. There are successful examples of simple, two-position VCT systems that address and partially resolve the idle versus wide-open throttle performance trade-off. Obviously, variable cam timing has a substantial effect on the breathing process of the engine. Properly controlled, the variable cam can be used to operate the engine at higher intake manifold pressures, reducing pumping losses at part throttle conditions and providing a fuel economy improvement as well [2], [4], [7].

Four versions of VCT are available for double overhead camshaft (DOHC) engines: phasing only the intake cam (intake only), phasing only the exhaust cam (exhaust only), phasing the intake and exhaust cams equally (dual equal), and phasing the two camshafts independently (dual independent). The dual independent VCT provides the best performance, but it is the most complex and expensive to implement. Of the remaining three, dual equal VCT, where the intake valve timing and exhaust valve timing are advanced or retarded equally, gives the best overall performance in terms of emissions and fuel economy [6]. On the other hand, the dual equal VCT causes the greatest disturbance to cylinder air-charge and air-fuel ratio which may result in drivability problems and increased tailpipe emissions above levels predicted by the steady state analysis [11].

One method of reducing the air-charge variation and improving drivability is to "detune the cam" by slowing down the re-

sponse of the VCT mechanism. However, detuning the cam results in engine operation with lower levels of recirculated exhaust gas and increased NOx emissions. Drivability can also be improved by selecting a less aggressive (steady-state suboptimal) cam timing schedule, but again the price for this is increased NOx emissions and reduced fuel economy. Instead, in this article we pursue an active method of compensation for cylinder air-charge variation due to VCT which employs either an electronic throttle or an air-bypass valve. Electronic control of the throttle provides an additional degree of control capability to affect emissions and fuel economy while providing the performance desired by the driver as interpreted from the accelerator pedal position. The air-bypass valve is a conventional actuator of limited authority used to admit air flow to the engine under closed throttle operation, and to effect idle speed regulation [3], [5].

From the controller design point of view, several prominent characteristics of this problem will dictate our approach:

- The model of the plant is low order, nonlinear, with known nonlinearities available from the dynamometer mapping data.
- The disturbance (cam phasing) is measured.
- The output (cylinder mass air flow, or cylinder air charge, or torque) is not measured.
- Transient response is much more important than steady state accuracy.

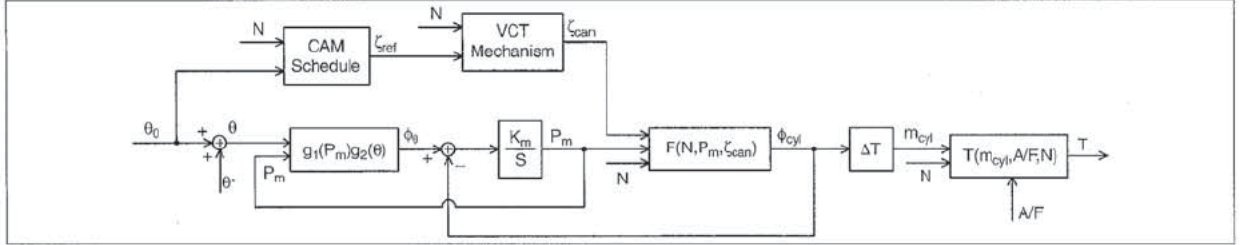


Fig. 2. Engine model with VCT and electronic throttle.

The first three items above suggest that a feedforward compensation may be the most appropriate, so this is the approach we adopt. Our compensator design is based on a low order nonlinear model of a VCT engine derived in [12]. Starting from the nonlinear model we develop a nonlinear control algorithm that rejects the air-charge (torque) disturbance and recovers the drivability of the conventional engine. (Note that the “conventional engine” in this context is the engine which corresponds to fixed cam at 0° (base cam). However, this algorithm requires a degree of control authority which is available only if the engine is equipped with an electronic throttle (ETC). If the engine is equipped only with an air-bypass valve (ABV), the limited control authority of this device forces us to change the design objective: rather than to emulate the conventional engine, our compensation tries to achieve the drivability of an engine with a different characteristic.

Because the relative degree from the disturbance to the output is smaller than the one from the control input to the output (the disturbance is closer to the output than the control), our compensation algorithm uses the derivative of the disturbance (cam phasing). We present two different methods of implementing this term in order to find a trade-off among the performance of the compensator, the sensitivity to measurement noise, and the sensitivity to parameter variation.

VCT Engine Model

The control design engine model described here is a continuous-time nonlinear, low-frequency phenomenological model developed in [10] and modified to incorporate VCT in [12]. Details about the VCT engine model can be found in [13]. A block diagram in Fig. 2 shows the cam timing reference ζ_{ref} scheduled on engine speed and driver demanded throttle θ_0 .

Notation

A/F air-fuel ratio
 N engine speed
 P_m manifold pressure
 m_{cyl} mass of air into cylinder
 T torque
 ζ_{ref} cam phasing command
 ζ_{cam} cam phasing
 θ_0 throttle angle due to driver request
 θ^* additive throttle angle due to compensation
 θ total throttle angle
 ϕ_θ mass air flow rate into manifold
 ϕ_{cyl} mass air flow rate into cylinders
 δ opening of the air-bypass valve
 ΔT duration of the intake event
 σ spark retard from maximum braking torque (MBT) setting.

The engine breathing dynamics describe the filling and emptying of the manifold. The rate of change is proportional to the difference between mass air flow rate into the manifold through the throttle (ϕ_θ) and the mass air flow rate out of the manifold into the cylinders (ϕ_{cyl})

$$\dot{P}_m = K_m(\phi_\theta - \phi_{cyl}). \quad (1)$$

The mass air flow through the throttle body into the manifold is a function of the upstream atmospheric pressure P_a , the manifold pressure, and the throttle angle:

$$\phi_\theta = g_1(P_m / P_a)g_2(\theta, P_a),$$

where

$$g_1(P_m / P_a) = \begin{cases} \gamma^{1/2} \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma+1}{2(\gamma-1)}} & \text{if } \frac{P_m}{P_a} \leq \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}} \\ \sqrt{\frac{2\gamma}{\gamma-1} \left(\left(\frac{P_m}{P_a} \right)^{\frac{2}{\gamma}} - \left(\frac{P_m}{P_a} \right)^{\frac{\gamma+1}{\gamma}} \right)} & \text{if } \frac{P_m}{P_a} > \left(\frac{2}{\gamma+1} \right)^{\frac{\gamma}{\gamma-1}}, \end{cases} \quad (2)$$

with $\gamma = c_p / c_v = 1.4$. $g_2(\theta, P_a)$ is the throttle dependent mass air flow characteristic obtained from static engine data. We assume that $P_a = 1$ bar and suppress the dependence of g_1 and g_2 on P_a .

In this application, the throttle angle is comprised of the throttle position due to the driver's request (θ_0) and an additive term due to the compensation (θ^*)

$$\theta = \theta_0 + \theta^*.$$

It is important to observe in Fig. 2 that the cam is scheduled only on θ_0 . The mass air flow rate into the cylinders can be represented as a function of cam phasing ζ_{cam} , manifold pressure P_m , and engine speed N :

$$\phi_{cyl} = F(N, P_m, \zeta_{cam}). \quad (3)$$

For our design model we approximate $F(N, P_m, \zeta_{cam})$ by a function linear in P_m :

$$\phi_{cyl} = \alpha_1(N, \zeta_{cam})P_m + \alpha_2(N, \zeta_{cam}), \quad (4)$$

where α_1 and α_2 are low order polynomials in N and ζ_{cam} .

The generation of torque depends on the cylinder air charge m_{cyl} , air-fuel ratio A/F , spark retard from MBT σ , and engine speed. This relationship is best captured by curve fitting of experimental data:

$$T = T(m_{cyl}, A/F, \sigma, N), \quad (5)$$

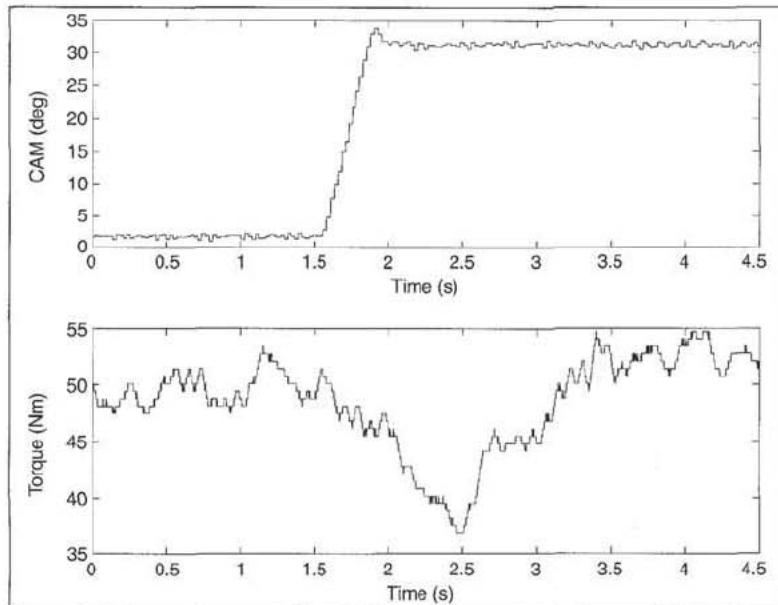


Fig. 3. Engine torque response to a change in cam phasing at fixed throttle and engine speed.

where $m_{cyl} \approx \phi_{cyl} \Delta T$, (ΔT is the duration of the intake event which depends on engine speed).

In spark ignited (gasoline) engines the air-fuel ratio is maintained at stoichiometry (14.3) in order to ensure that the catalytic converter operates at its peak efficiency. Thus, from the expressions (3) and (5) it is clear that the change in cam timing affects the torque through the change in cylinder mass air flow rate. In sonic flow, when $g_1(P_m) = c$ (c is a constant), the change in ζ_{cam} does not change the steady state value of the torque. This can be verified on the model by noting that, in steady state when $\dot{P}_m = 0$, $\phi_{cyl} = \phi_0 = cg_2(\theta)$ independently of ζ_{cam} . However, cam transients do affect the torque as one can see in the dynamometer test data in Fig. 3 that show the response of engine torque to a change in ζ_{ref} with the other variables held constant. If the throttle flow is subsonic, cam changes also affect the steady state values of the cylinder mass air flow and torque.

The cam timing has to change depending on the operating conditions. For this reason, ζ_{ref} is scheduled on engine speed and throttle position. Typically, the schedule reaches maximal cam retards at part throttle which provides maximal internal EGR. While close to idle and at wide open throttle the cam phasing is at zero or slightly advanced. Scheduling cam timing on throttle causes it to change when the pedal is depressed or released. The torque variation caused by the cam transient leads to an undesirable engine response and drivability problems. This effect will be illustrated in subsequent sections.

Electronic Throttle Based Compensator

We design a compensator to remove or reduce the effect of the cam transients on the cylinder mass air flow which employs θ^* as a virtual actuator. We assume that the variables available for measurement are the manifold pressure P_m , engine speed N , and the cam position ζ_{cam} . Our design objective is to recover the drivability of the conventional engine.

The basic idea is to find a control law for θ^* such that the rate of change of ϕ_{cyl} coincides with that of the conventional engine. We start with the equations (1) and (3) which describe the manifold filling and engine pumping rates. By differentiating ϕ_{cyl} we obtain

$$\dot{\phi}_{cyl} = \frac{\partial F(N, P_m, \zeta_{cam})}{\partial N} \dot{N} + \frac{\partial F(N, P_m, \zeta_{cam})}{\partial P_m} \dot{P}_m + \frac{\partial F(N, P_m, \zeta_{cam})}{\partial \zeta_{cam}} \dot{\zeta}_{cam}.$$

To simplify our presentation we shall assume that the $\dot{N}$ term is approximately equal to the corresponding term in the conventional engine; that is, we assume that $\partial F(N, P_m, \zeta_{cam}) / \partial N - \partial F(N, P_m, 0) / \partial N \dot{N} \approx 0$. Hence, we suppress the $\dot{N}$ -terms in our subsequent derivation. If this assumption does not hold we can redesign our compensator in a straightforward way to account for this additional disturbance term. By using the linear (in P_m) approximation of F and expanding $\dot{P}_m$ we obtain

$$\dot{\phi}_{cyl} = \alpha_1(N, \zeta_{cam}) K_m (g_1(P_m) g_2(\theta_0 + \theta^*) - \phi_{cyl}) + \left(\frac{\partial \alpha_1}{\partial \zeta_{cam}} P_m + \frac{\partial \alpha_2}{\partial \zeta_{cam}} \right) \dot{\zeta}_{cam}. \quad (6)$$

Now we can use θ^* to cancel the disturbance term proportional to $\dot{\zeta}_{cam}$. This, however, will not achieve complete disturbance rejection because α_1 and P_m depend on ζ_{cam} . To achieve our objective we have to choose θ^* in such a way that ϕ_{cyl} behaves as if it is a cylinder mass flow rate of the conventional engine:

$$\dot{\phi}_{cyl} = \alpha_1(N, 0) K_m (g_1(\tilde{P}_m) g_2(\theta_0) - \phi_{cyl}), \quad (7)$$

where $\tilde{P}_m$ is a fictitious reference manifold pressure which should be equal to the manifold pressure of the conventional engine driven with the throttle θ_0 and engine speed N . This reference manifold pressure is generated by

$$\dot{\tilde{P}}_m = K_m (g_1(\tilde{P}_m) g_2(\theta_0) - \alpha_1(N, 0) \tilde{P}_m - \alpha_2(N, 0)). \quad (8)$$

The expression for θ^* is simplified if we substitute $\alpha_1(N, \zeta_{cam})$ instead of $\alpha_1(N, 0)$ in (3). This is justified by the fact that between the maximal and the minimal values of ζ_{cam} , α_1 changes by approximately 20–25% and that α_1 affects mostly the speed of response of ϕ_{cyl} .

To achieve (3) with $\alpha_1(N, \zeta_{cam})$ instead of $\alpha_1(N, 0)$, the following equality must be satisfied

$$\alpha_1(N, \zeta_{cam}) K_m g_1(P_m) g_2(\theta_0 + \theta^*) + \left(\frac{\partial \alpha_1}{\partial \zeta_{cam}} P_m + \frac{\partial \alpha_2}{\partial \zeta_{cam}} \right) \dot{\zeta}_{cam} = \alpha_1(N, \zeta_{cam}) K_m g_1(\tilde{P}_m) g_2(\theta_0). \quad (9)$$

Thus, the compensation θ^* should be chosen as

$$\theta^* = g_2^{-1} \left(\frac{\frac{\partial \alpha_1}{\partial \zeta_{cam}} P_m + \frac{\partial \alpha_2}{\partial \zeta_{cam}} \dot{\zeta}_{cam}}{K_n g_1(P_m) \alpha_1} + \frac{g_1(\tilde{P}_m)}{g_1(P_m)} g_2(\theta_0) \right) - \theta_0. \quad (10)$$

Because the function g_1 changes little in the range of manifold pressures between 0.2 to 0.8 bar, another possible simplification is to use the steady state value $P_m^{ss}(\theta_{0,N})$ instead of the dynamically generated pressure $\tilde{P}_m$. In simulations, a slight degradation in performance is noticeable only at high reference manifold pressures.

Robust stability

The above compensation method uses the information of the engine speed, cam phasing, and manifold pressure to modify the throttle and achieve the drivability of the conventional engine. Even though it is designed as a feedforward compensator, because of its dependence on P_m , there is a feedback component which necessitates stability analysis.

If, for fixed N and θ_0 , cam retard causes the steady state manifold pressure to rise into the subsonic region, the compensation will create a different set point at a higher value of manifold pressure. This new set point, denoted by P_m^0 , may be sensitive to modeling uncertainties. Below, we propose a simple modification which makes the value of P_m^0 less sensitive to modeling uncertainties and guarantees that it is a stable equilibrium point.

If the functions g_1 and g_2 are known accurately, the implementation of the control law (3) leads to

$$\dot{P}_m = K_n \left(-\frac{\frac{\partial \alpha_1}{\partial \zeta_{cam}} P_m + \frac{\partial \alpha_2}{\partial \zeta_{cam}} \dot{\zeta}_{cam}}{K_n \alpha_1} + g_1(\tilde{P}_m) g_2(\theta_0) - F(N, P_m, \zeta_{cam}) \right) \quad (11)$$

In steady state, when $\dot{P}_m$ and $\dot{\zeta}_{cam}$ are zero, $\phi_{cyl} = F(N^{ss}, P_m, \zeta_{cam}^{ss}) = g_1(\tilde{P}_m^{ss}) g_2(\theta_0)$, that is, in steady state, the cylinder mass air flow rate of the compensated VCT engine is exactly equal to that of the conventional engine.

To analyze the sensitivity of the compensation method we note that the functions g_1 , g_2 , α_1 and α_2 can be relatively accurately modeled from the static engine data. The only significant source of sensitivity arises because the function g_1 , which is close to 0 at high manifold pressures, appears in the denominator in (10). So we only analyze the effect of modeling errors in g_1 on the steady state pressure P_m^0 and the stability of this equilibrium. To make it distinct from the "true" value g_1 , we use $\hat{g}_1$ to denote the subsonic flow correction factor employed in the compensator (10).

The steady state pressure P_m^0 can be obtained by solving

$$0 = \frac{g_1(P_m)}{\hat{g}_1(P_m)} \hat{g}_1(\tilde{P}_m^{ss}) g_2(\theta_0) - F(N^{ss}, P_m, \zeta_{cam}^{ss}). \quad (12)$$

At higher manifold pressures, the solution of (12) is very sensitive to changes in $\hat{g}_1$. We can reduce this sensitivity by intentionally skewing $\hat{g}_1$ so that $\hat{g}_1(P_m) > g_1(P_m)$ when P_m is high, say $P_m \geq 0.75$ bar. This guarantees the existence of a single solution of (12) and forces it to assume a lower value, in the direction of

smaller sensitivity of (12). The resulting steady state error in the cylinder mass air flow rate does not affect the drivability.

To analyze the stability of the equilibrium at P_m^0 , we define $x = P_m - P_m^0$. With $\dot{\zeta}_{cam} = 0$, the linearization of the manifold pressure dynamics around the equilibrium $P = P_m^0$ results in

$$\dot{x} = \left(\frac{1}{\hat{g}_1^2} \left(\frac{\partial g_1}{\partial P_m} \hat{g}_1 - \frac{\partial \hat{g}_1}{\partial P_m} g_1 \right) \hat{g}_1(\tilde{P}_m^{ss}) g_2(\theta_0) - \alpha_1 \right) x.$$

By our choice of $\hat{g}_1$ we have assured not only that $\hat{g}_1 \geq g_1 \geq 0$, but also that $|\partial \hat{g}_1 / \partial P_m| \leq |\partial g_1 / \partial P_m|$. Since $\partial \hat{g}_1 / \partial P_m$ and $\partial g_1 / \partial P_m$ are both negative, this implies that $(\partial g_1 / \partial P_m) \hat{g}_1 - (\partial \hat{g}_1 / \partial P_m) g_1 \leq 0$. Because $\hat{g}_1$, g_2 , and α_1 are always positive, the equilibrium $x = 0$ is asymptotically stable. Hence, our compensation creates a single equilibrium P_m^0 which is asymptotically stable.

Simulation results for ETC compensator

To demonstrate the effectiveness of the compensation (10) we have tested it on a model which differs from the one used in the controller design. We have used a simpler design model in order to keep the compensator simple. In addition, this provides some indication of the effects of modeling errors on the performance of the compensator.

To implement $\dot{\zeta}_{cam}$ we have chosen an approximate differentiation

$$\dot{\zeta}_{cam} \approx \frac{s}{\tau s + 1} \zeta_{cam}, \quad \tau = 0.04s.$$

If the high frequency measurement noise present in the signal ζ_{cam} is excessive, one can use a model based approximation of $\dot{\zeta}_{cam}$. We shall discuss this issue in the final section.

The compensator, which has been developed in continuous time, has been discretized for simulations. The sampling rate is chosen to be 10ms.

Fig. 4 shows the reduction of the torque fluctuation during cam transients achieved by the compensation. For this simulation run we have varied the reference cam independently from the throttle and engine speed (which are held constant). In this and all other simulations the MBT spark is used, that is $\sigma = 0$. Clearly, the compensator achieves a substantial level of disturbance attenuation. The disturbance rejection is not perfect because of the differences between the design and simulation models, discretization, and filtering used for the derivative of ζ_{cam} . The performance of the compensator appears much better if we look at its effect on drivability. Fig. 5 shows the performance of the compensated VCT engine compared with the conventional engine and uncompensated VCT engine. The input is the throttle step with the engine speed held constant and the cam reference scheduled on throttle and engine speed. The uncompensated VCT engine response is not monotonically increasing, indicating a drivability problem. Our compensation θ^* , which can be seen in the top plot in Fig. 5 as the difference between the solid and dashed curves, brings the response of the VCT engine close to that of the conventional engine, effectively removing this problem. We emphasize that the difference between the steady state values of the conventional engine and the (uncompensated) VCT engine has little effect on drivability. Rather, it is the shape

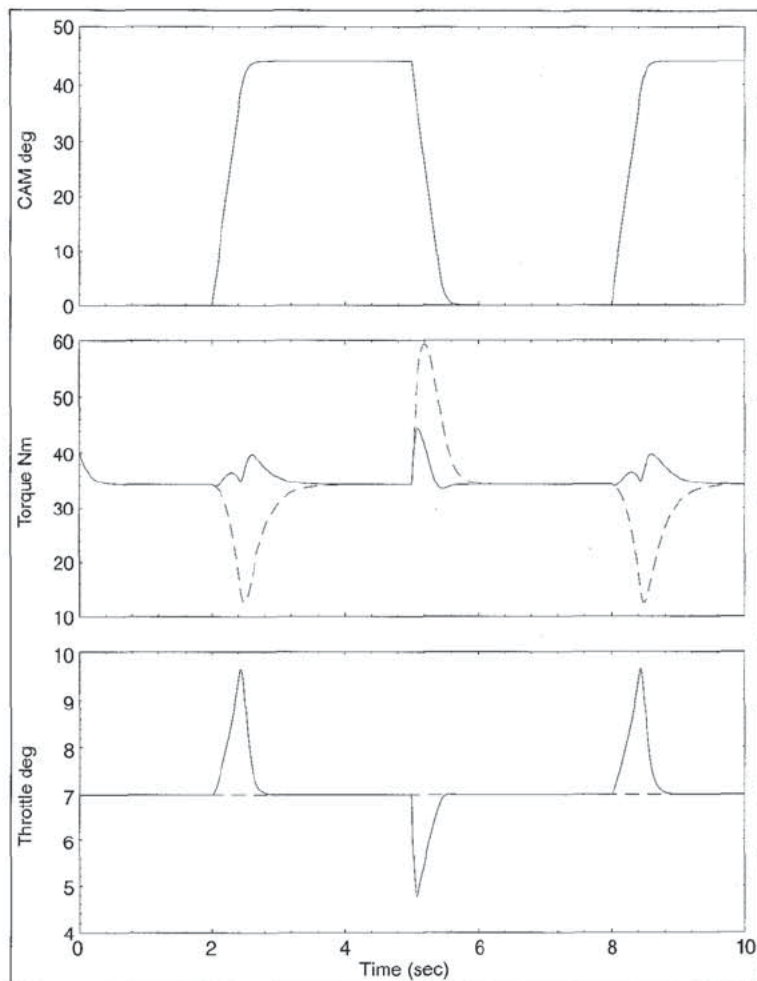


Fig. 4. Torque response of the VCT engine (dashed lines) to cam phase steps (0 to 44 deg.) at fixed engine speed (2000 RPM) and throttle angle (7 deg.); response with active compensation (full lines) shows substantial reduction in torque variation.

of the transient response which affects it the most. This observation will be used to modify our design objective in the next section, because low control authority of the air-bypass valve prevents us from recovering the torque response of the conventional engine.

Compensation with the Air-Bypass Valve

In the previous section we showed an effective way to modulate the electronic throttle to make a VCT engine respond like the conventional one. In most cases the engine will be equipped only with an air-bypass valve (ABV) which is used to control the idle speed by modifying the amount of air entering the intake manifold. We employ the ABV to affect the transients of the VCT engine and improve drivability.

Because an ABV has much smaller control authority than an ETC, we have to change our design objective: instead of trying to perfectly match the response of the conventional engine, we shall only try to modify the torque transient to have the same shape as that of a conventional engine. The steady state torque will be different, equal to that of the uncompensated VCT engine.

To explain this point, we refer to Fig. 6. The dashed and dotted curves show the static throttle-to-load (load is the normalized cylinder air charge) characteristics at a fixed engine speed for conventional ($\zeta_{cam} = 0^\circ$) and fixed cam ($\zeta_{cam} = 35^\circ$) engines. By scheduling the cam on the throttle and engine speed, we create a new static characteristic for the VCT engine shown by the solid curve in Fig. 6. Now our objective is to modify the transients of the VCT engine, using the ABV, so that the torque response is the same as that of a (fictitious) conventional engine which has a static throttle-to-load characteristic shown by the solid curve.

The new objective creates a problem for the design because now we do not have simple reference manifold pressure dynamics. In fact, as we change the throttle, the scheduled ζ_{ref} changes, and our reference manifold pressure model changes. We emphasize that it is not sufficient to introduce $F(N, P_m, \zeta_{ref})$ instead of $F(N, P_m, 0)$ in the manifold pressure dynamics (8). The problem is that, at the step change in ζ_{ref} , not only must the dynamics for the reference pressure change, but also reference pressure value at that instant must change (jump). The generation of the desired reference manifold pressure can be done systematically at the expense of significant complexity. Instead, as a reference pressure we use the steady state value of the manifold pressure corresponding to the current values of N, θ_0 , and ζ_{ref} . Simulation results have confirmed that the simplified algorithm performs very well.

With the air-bypass valve, the ideal gas law has to take into account the additional air coming through the ABV:

$$\dot{P}_m = K_m [g_1(P_m)(g_2(\theta_0) + g_3(\delta)) - F(N, P_m, \zeta_{cam})], \quad (13)$$

where $\delta = \delta_0 + \delta^*$ is the opening of the air-bypass valve and g_3 is its static flow characteristic. We assign $\delta = 0$ to the completely closed and $\delta = 1$ to the completely open valve. Following the same steps as for the electronic throttle compensation, we obtain

$$\delta^* = g_3^{-1} \left(-\frac{\frac{\partial \alpha_1}{\partial \zeta_{cam}} P_m + \frac{\partial \alpha_2}{\partial \zeta_{cam}}}{K_m g_1(P_m) \alpha_1(N, \zeta_{cam})} \dot{\zeta}_{cam} + \frac{g_1(\bar{P}_m^{ss})}{g_1(P_m)} [g_2(\theta_0) + g_3(\delta_0)] \right) - \delta_0,$$

where, as we have said above, the reference pressure $\bar{P}_m^{ss} = \bar{P}_m^{ss}(N, \theta_0, \delta_0, \zeta_{ref})$ is the steady state value of the manifold pressure corresponding to the present values of the variables N, θ_0, δ_0 , and ζ_{ref} . The evaluation of the reference pressure $\bar{P}_m^{ss}$ in real time can be done by evaluating two functions of two variables (two look-up tables).

The potential for instability for the compensation with the ABV is much smaller than with the ETC. The reasons are (i) at

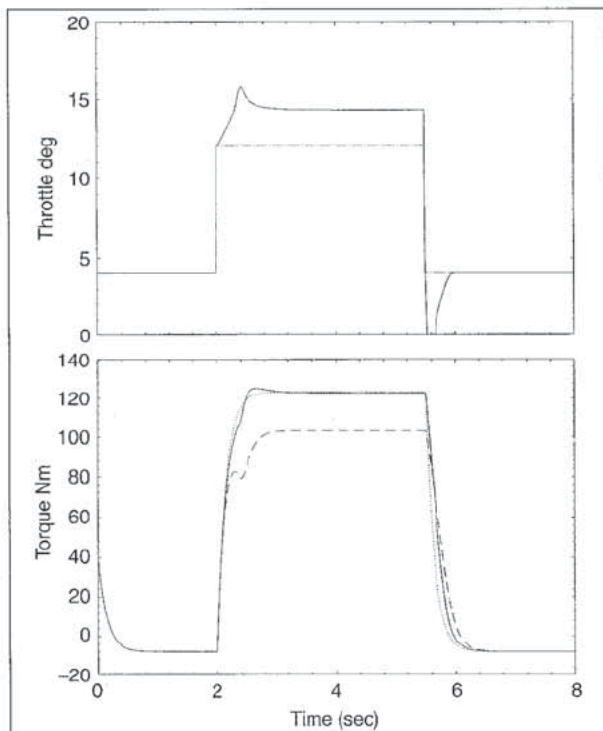


Fig. 5. Comparison of the throttle step torque response of the conventional engine (dotted lines), VCT engine without compensation (dashed lines), and VCT engine with compensation (full lines); engine speed is 2000 RPM, throttle is stepped from 4 to 12 and back to 4 deg.

the high end, $\bar{P}_m^{ss} < P_m^0$, and (ii) at the operating regimes with high manifold pressure, the ABV has little control authority. Nevertheless, we have used the skewed value $\hat{g}_i$ in the implementation.

The simulations are performed with the same simulation and design models as in the case of electronic throttle except that both models now include the air-bypass valve as an actuator. The air-flow characteristic of the valve is modeled as a sigmoidal function for the simulation model and as a straight line for the design model.

Plots in Fig. 7 show the air-bypass valve opening δ and the torque responses to throttle steps with the cam timing scheduled on throttle angle and engine speed. The torque of uncompensated VCT engine (dashed line) flares at both tip-ins and tip-outs indicating a drivability problem. Our compensation algorithm results in the torque response which closely resembles a scaled down response of the conventional engine (full line). The dynamics of the air-bypass valve have been neglected in the simulation model; the traces of δ for the uncompensated VCT engine show the dash-pot functionality of the air-bypass valve. The difference between the steady state values for two torques in Fig. 7 is due to the difference between the simulation and design models. It has no consequence on drivability and can be removed with a simple high-pass filter if desired.

The algorithm has shown very good performance in simulations. The compensator has performed well over a variety of speed and load conditions. At high loads (high values of throttle) the air-bypass valve loses its control authority and the torque re-

sponses of the compensated and uncompensated engines are the same. However, this happens in the region where the VCT does not cause drivability problems.

Implementation of $\dot{\zeta}_{cam}$

Our compensation algorithms require that the derivative of ζ_{cam} be available. Two methods to obtain this derivative are explored. One method is to use the approximate differential of the available measurement as we have done for the simulations, the other one is a model based estimator. For the approximate differential, we use the a second order, low-pass Butterworth filter. The model based estimator uses ζ_{ref} and N as inputs to the model of the VCT mechanism (c.f. Fig. 2) which is itself a feedback loop consisting of the VCT actuator controlled by a PID regulator.

The VCT actuator system consists of a control valve and the actuator itself. This control oriented model captures the dominant behavior of the VCT actuator system. These dominant dynamics can be reduced to a single integrator. Adding higher frequency components does not add much to the accuracy of this model. The static control valve behavior is captured in the function $F(\cdot)$. It is purely a function of the control input. The gain of the VCT actuator depends heavily on the engine speed N and is described by the static function $K(N)$. A block diagram of the VCT mechanism model is shown in Fig. 8.

In Fig. 9, the estimated cam position and the cam position measured on the engine are the solid and dotted curve, respectively. The plot shows that the model captures the VCT mecha-

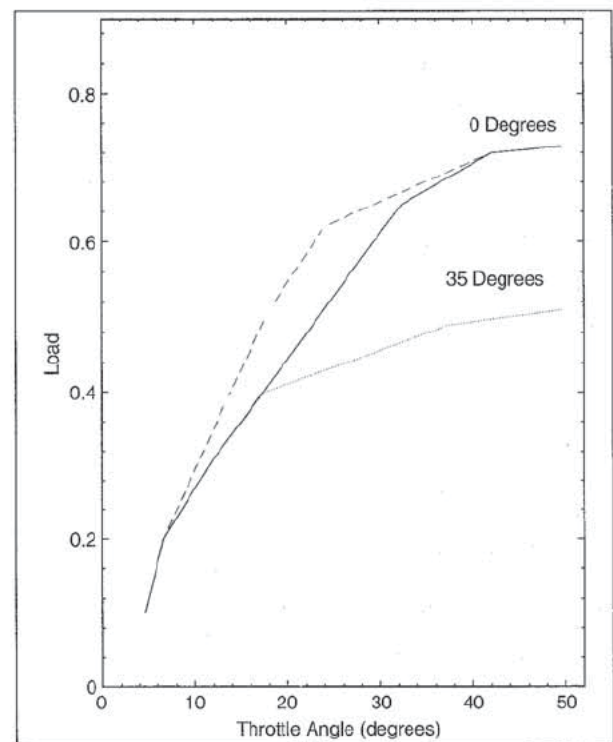


Fig. 6. Load vs throttle characteristic for the conventional engine (dashed line), an engine with cam fixed at 35° retard (dotted line), and the VCT engine with cam scheduled on throttle and engine speed (full line).

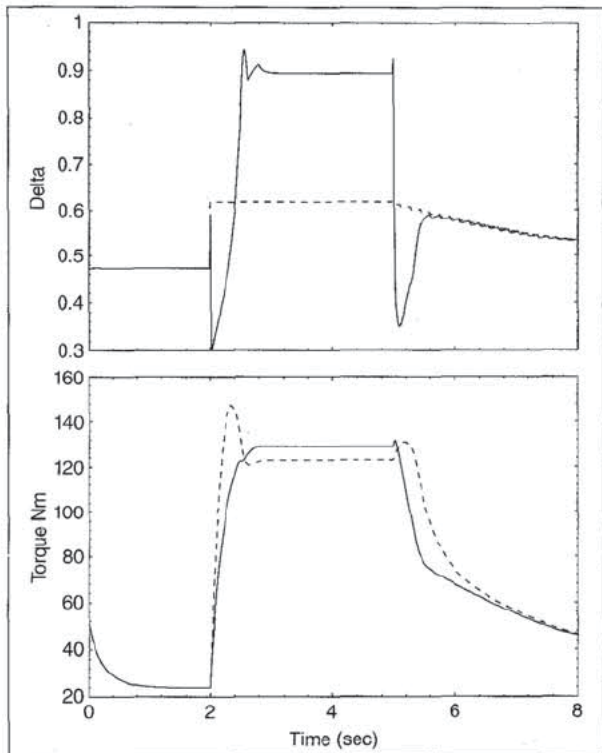


Fig. 7. Normalized air-bypass valve opening δ and torque response for the VCT engine without compensation (dashed lines) and with compensation (full lines); throttle steps 0 to 9 to 0 deg., engine speed 1250 RPM.

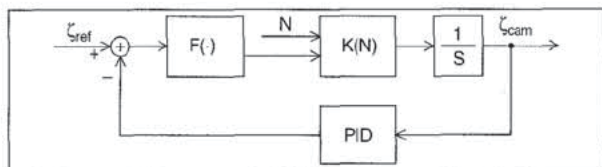


Fig. 8. Simplified controls-oriented model of the VCT mechanism.

nism dynamics very well. It also shows high frequency noise content in the measured signal which forces us to use a low bandwidth Butterworth filter for approximate differentiation.

The estimates of ζ_{cam} obtained by two methods are shown in Fig. 10. The one obtained by differentiating the measured cam position lags the model-based one because of the low pass filtering and measurement delay. It appears that the model is sufficiently accurate to show a clear advantage over an approximate differentiation method. One must keep in mind, however, that this conclusion applies for the particular hardware implementation of the VCT sensor and actuator.

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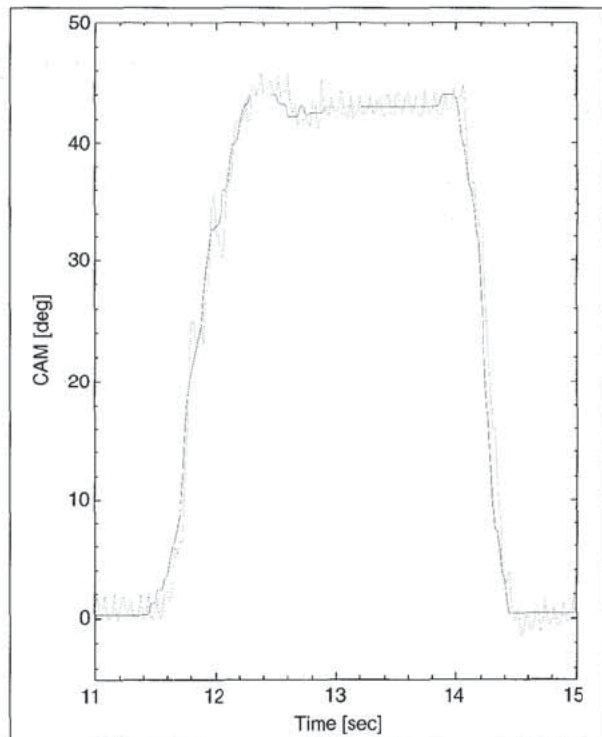


Fig. 9. Comparison of the model based ζ_{cam} estimate (solid) and ζ_{cam} measured (dotted).

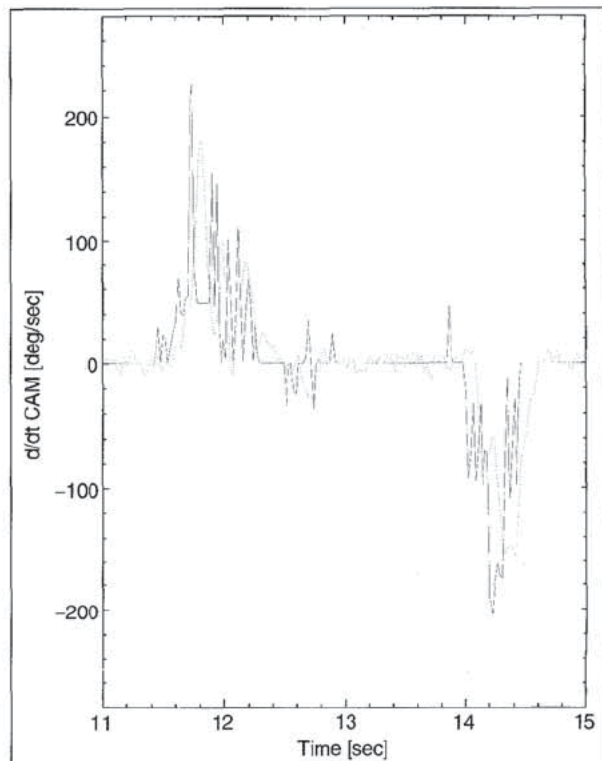


Fig. 10. Comparison of ζ_{cam} estimate obtained by approximate differentiation (dotted) and the model based one (solid).

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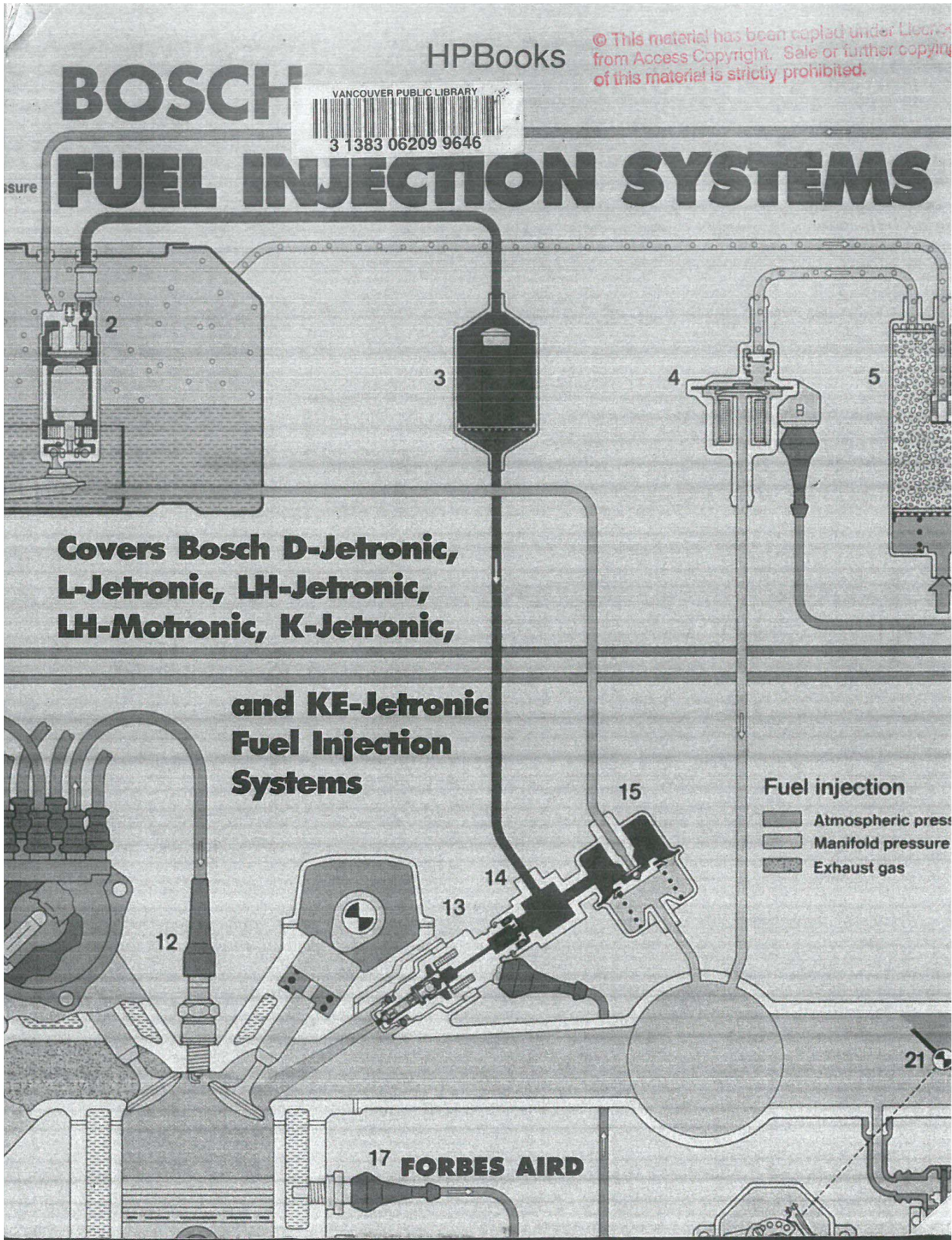
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VW EX1009
U.S. Patent No. 6,557,540

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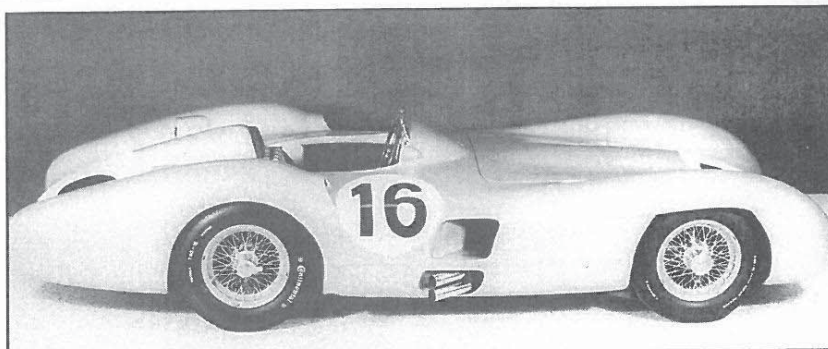
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FUEL INJECTION: THEN AND NOW

Despite the fact that there were a thousand and one other problems to be dealt with in the early days of the automobile, auto pioneers were quick to focus attention on the shortcomings of carburetors. The drawbacks included the possibility of ice forming within the venturi in cold, damp air, and a tendency to vapor lock in hot weather when volatile fuels were used. Conversely, carburetors were unsuitable for use with fuels that did not readily vaporize (remember, the automobile came first—gasoline followed, almost anything that would burn was used as a fuel at one time or another in the earliest days). Also, the venturi restricts airflow, and thus power. Because fuel delivery depends on the level in the float bowl, a carburetor is sensitive to the inclination of the car, such as when climbing hills. In addition, there was some dawning awareness of the problems of feeding multiple cylinders from a single "central mixer," as described in the previous chapter. Accordingly, some of the critics set about designing an alternative, specifically what we now call fuel injection.

Early Production FI Systems

Although there were even earlier experiments, perhaps the first production application of FI was by the German company, Deutz, who built about 300 fuel-injected stationary engines around the turn of the century. Hampered by a lack of precision manufacturing facilities, Deutz abandoned the scheme soon after. (Deutz is still in the business of manufacturing industrial and marine diesel engines.) Perhaps aware of Deutz's work, Robert Bosch began experiments with fuel injection in 1912. In 1927, Bosch bought the Acro company, and the patent rights of an Acro employee, Franz Lang,



who had successfully developed the equipment needed for high-pressure fuel injection. Shortly afterward, Bosch began manufacturing diesel injection equipment of a pattern that has changed little in the 70-odd years since (see sidebar "The Real Jerk").

The adaptation of this system to gasoline-fuelled engines began in the 1930s, initially for use in aircraft. In this application, three advantages of fuel injection over carburetors stand out: freedom from problems of vapor lock, no risk of icing, and a complete indifference to the attitude of the vehicle. Vapor lock is a significant problem at high altitudes, because the reduced atmospheric pressure there makes it so easy for the fuel to evaporate that it is inclined to boil, forming bubbles of vapor in all sorts of awkward places that can effectively prevent any fuel from getting to the engine. Carburetor icing, while merely an annoyance in an automobile, can be potentially fatal in an aircraft. And a fuel system that can keep the engine running no matter what the attitude of the aircraft becomes a matter of first importance during extreme maneuvers, such as inverted flight. Most, if not all, Daimler Benz engines fitted to German aircraft during World War II enjoyed this advantage over their Allied opponents.

Early Racing FI Systems

If we substitute severe cornering for

The straight eight engine of Mercedes Benz's first post-war Formula One race car—this streamlined W196—**injected fuel directly into each cylinder, a system developed from Diesel injection principles. (Mercedes Benz).**

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THE REAL JERK

In a diesel, the cylinder at the end of the compression stroke is filled with nothing but air; combustion cannot begin until the fuel is injected. Thus, the moment of introduction of the fuel is equivalent to the ignition timing on a spark-ignition gasoline engine. For that reason, control is needed over not just the quantity of fuel injected, but also the timing of that injection.

Because the fuel has to be kept separate from the air until the correct instant for ignition, the fuel is injected directly into the combustion chamber—the injector nozzle lies inside the cylinder. As a result of that, the injector is exposed to full cylinder pressure during the power stroke. To prevent combustion pressure from blowing the fuel backward through the fuel lines, the injector is fitted with a very stiff spring-loaded check valve. This check valve also performs a number of other important functions. It prevents fuel from dribbling out of the injectors between timed squirts, and ensures that the start and end of each injection is clearly defined. Without this valve, the fuel spray would taper off gradually toward the end of each injection. The check valve, in conjunction with similar check valves at the pump outlets, also ensures that the fuel lines leading from the injection pump remain filled with fuel at full pressure, to ensure that the pressure pulse from the pump is transmitted immediately to the injector nozzle. To overcome the resistance of these check valves, the entire system has to operate at a very high pressure—as much as 3000psi!

The design of the classic diesel injection pump, and the gasoline injection pumps directly derived from that design, follows from the three requirements: to control the quantity of the fuel delivery, and its timing, and to provide a very high pressure output. In construction, it consists of an approximately rectangular body, bored with a number of small cylinders—one for each engine cylinder—each of which is lined with a cylindrical sleeve. The whole thing thus roughly resembles a miniature engine block, a similarity that extends to the "bottom end" of the pump, which has a shaft running lengthwise through it, analogous to an engine's crankshaft. Instead of crank throws, however, this shaft has a number of individual cams, one for each cylinder. Each cylinder in the pump contains a piston-like plunger that is driven up its sleeve by the corresponding cam lobe as the pump shaft rotates, driven by the engine at half crank speed, like an ordinary ignition distributor.

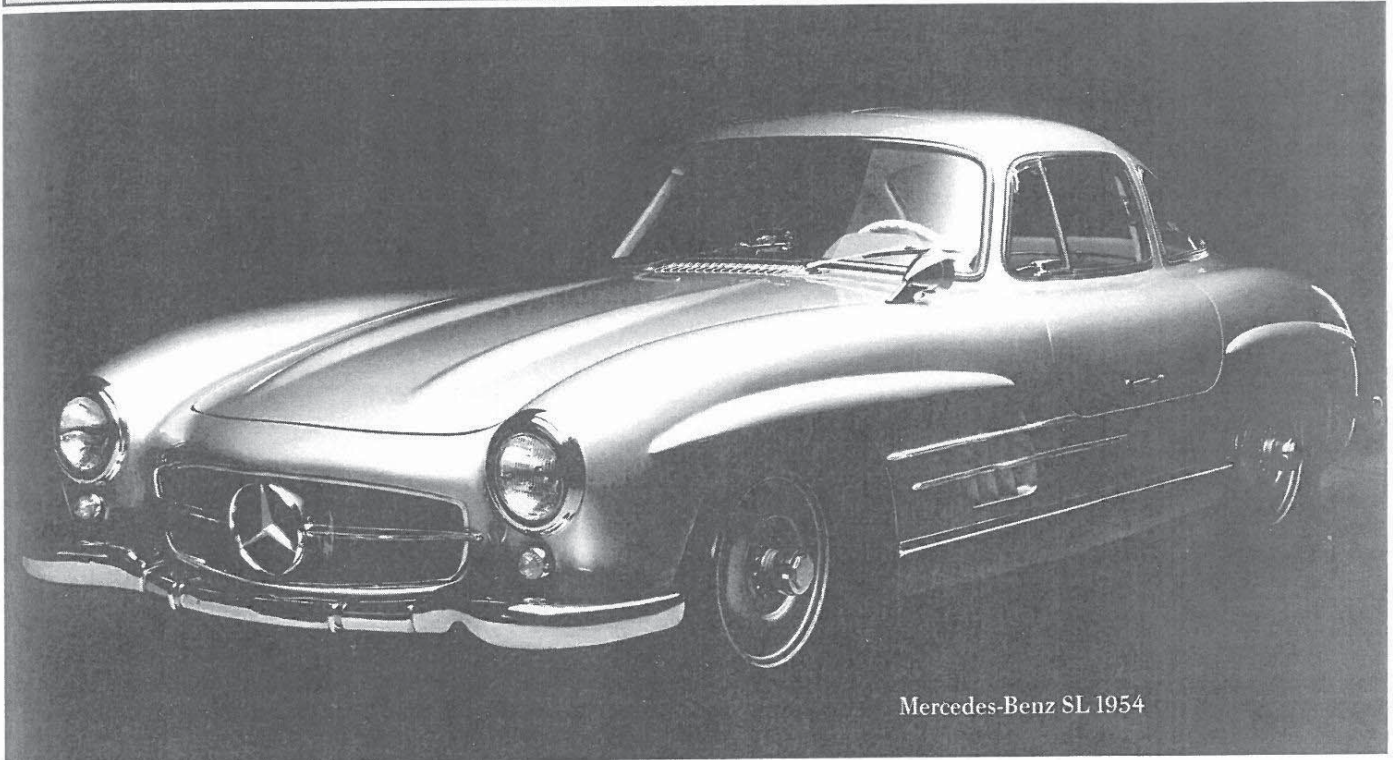
The mechanical drive thus provides the timing and the necessary pressure, leaving the problem of control of fuel quantity. To achieve this, each sleeve has a "spill" port drilled through its side, while the upper edge of each plunger has its side cut away to form a sort of spiral ramp. The sleeve fits in its bore with a very slight clearance, and so is free to rotate. Rotation of the sleeve in its bore thus covers or exposes the spill port, according to the angular position of the sleeve relative to the spiral cutaway on the plunger. With the sleeve rotated so that the spill port is covered even when the plunger is at the bottom of its stroke, the only route out of the cylinder is through a hole at the top, where the fuel lines to the injectors connect. A single stroke of the piston will thus expel the entire volume of fuel held in the cylinder; this represents the maximum capacity of the pump and thus corresponds to full power.

At anything less than wide-open throttle, a lesser amount of fuel per revolution is obviously needed, and this reduction is achieved by rotating the sleeve around, relative to the plunger, so that the spill port lies some distance above the plunger's spirally formed top edge. Upward movement of the plunger thus initially causes fuel to be discharged from this port, from where it is fed back to the tank, until the plunger rises far enough to close off the port, and injection commences.

The rotation of the sleeve(s) is accomplished by a toothed rack that meshes with gear teeth cut onto the outside of each sleeve. The rack is connected directly to the throttle linkage, so moving the pedal rotates the sleeves within the pump, thus controlling the quantity of fuel delivered with each stroke of the plunger.

Although such a mechanical injection pump is, in fact, quite a simple device, and contains few parts—just two per engine cylinder served, plus the camshaft and rack—the quantity of fuel delivered per stroke is very small, so all those parts have to be made with extreme precision, involving much hardening, grinding and precision gaug-

ing. Accordingly, mechanical injection pumps—known widely and for fairly obvious reasons as "jerk" pumps—are mighty expensive. When adapted for use with gasoline, rather than the kerosene used in diesels, there is the additional problem that gasoline is a "dry" fuel, lacking the lubricating properties of diesel fuel, thus demanding the use of extremely hard, wear-resistant alloys, which further adds to the machining difficulty and expense.



Mercedes-Benz SL 1954

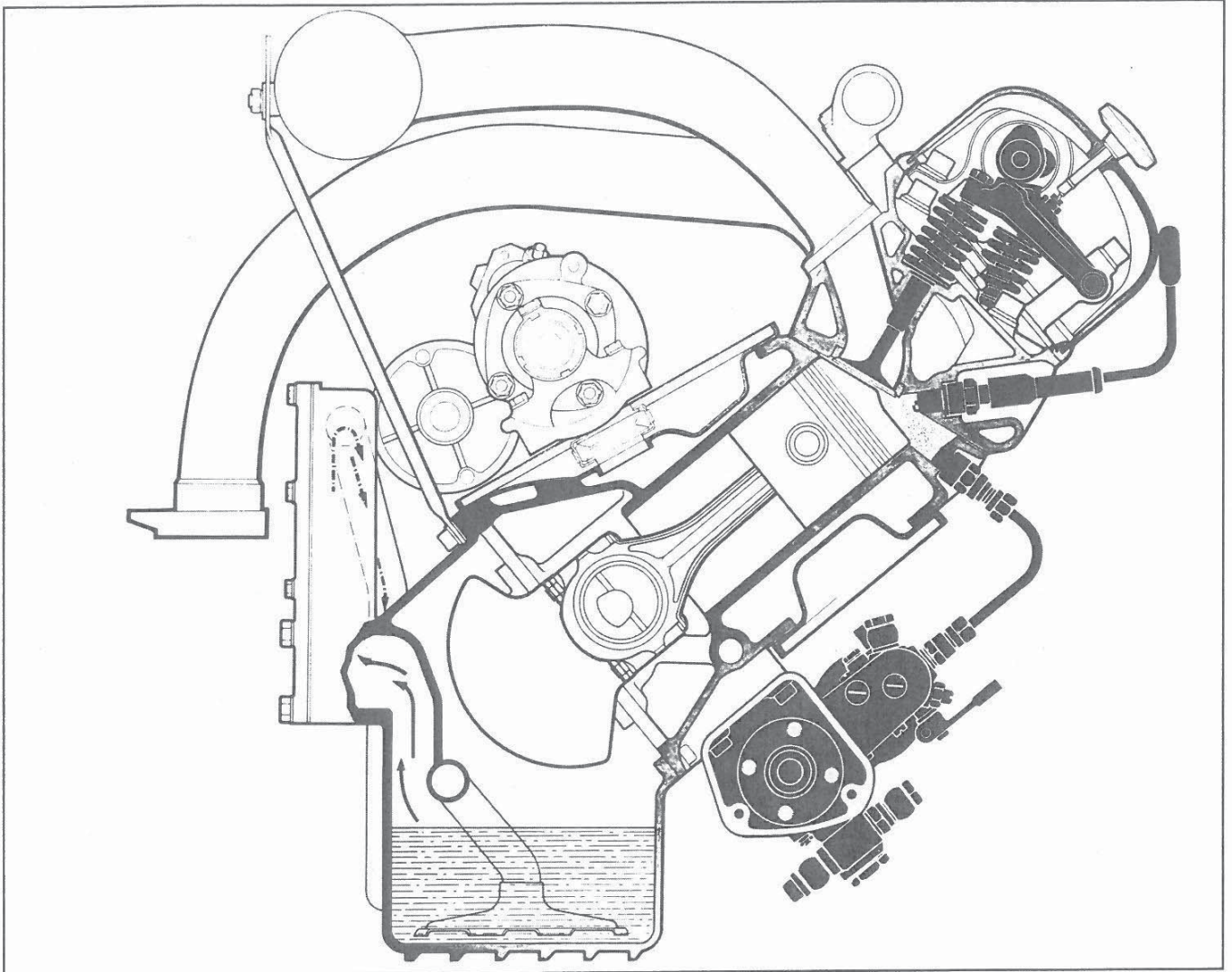
The famous Mercedes 300SL "gull-wing" coupe introduced fuel injection to the street. Like the W196, a system of direct injection was used. (Mercedes Benz).

inverted flight, most of the advantages listed are also highly applicable to race cars. Accordingly, by 1937 Mercedes Benz had tested a single cylinder mockup for a racing engine, using high-pressure injection of fuel directly into the combustion chamber. In the meantime, a handful of European makers of small, two-stroke engined passenger cars had introduced this "direct" fuel injection, in an attempt to tame the notorious thirst of two-strokes that results from something like one quarter of the air/fuel mixture whistling straight out the exhaust ports during the intake/exhaust event, when both inlet ("transfer") and exhaust ports are open

together. Because these engines were two-strokes, the injection pump had to run at crank speed, rather than half-speed, as for a four-stroke. This provided Bosch with valuable lessons in running their injection equipment at high pump rpm. Mercedes's first post-war Formula One race engine made direct use of the experience so gained, both their own and that of Bosch.

In both the M196 Formula One engine that appeared in 1952 and in the engine for the famous 300SL "gull-wing" sports car, first exhibited in New York in 1954, Mercedes retained the direct (into the combustion chamber) injection scheme used in

Bosch Fuel Injection Systems



The location of the injector—screwed right into the cylinder head—is apparent in this cross-section of the 300SL engine. (Daimler-Chrysler Archive)

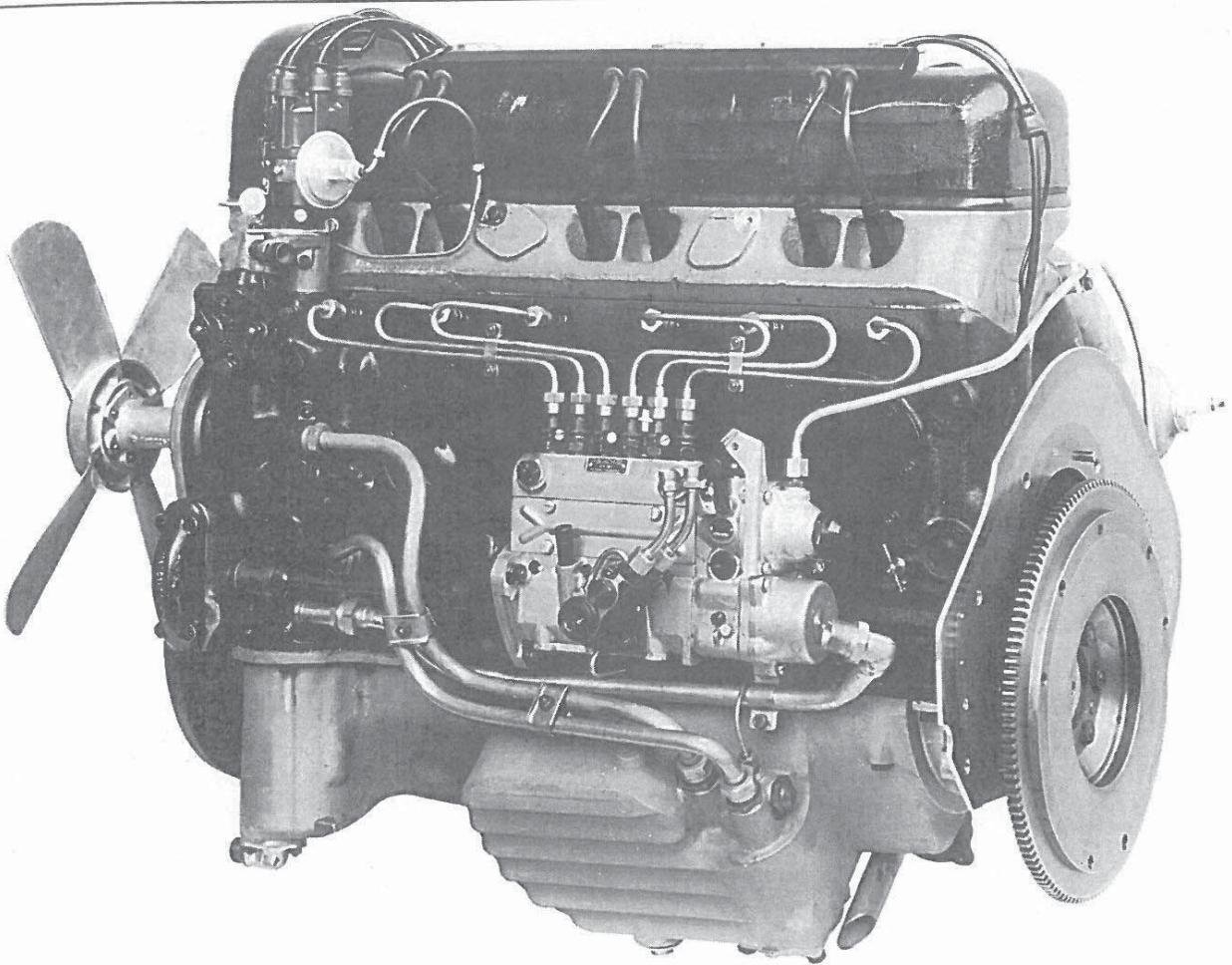
diesels and in the earlier experiments with gasoline fuel. Combined with suitable injection timing, this allowed radical valve timing and experiments with "tuned" intake pipes, without introducing the problem of poor fuel economy from portions of the intake charge being lost out the exhaust. Until injection occurred, after the exhaust valve had closed, the engine was inhaling only air.

Diesel Jerk Pump

Given the existence of an established technology for injecting fuel, it is hardly surpris-

ing that the diesel "jerk" pump was adapted in this way for gasoline injection. The adaptation, however, required hurdling a major difficulty—the problem of regulating the quantity of fuel delivered throughout the full range of engine operating speeds and loads.

Unlike gasoline engines where power is controlled by closing off ("throttling") the air supply, yet where a very narrow range of air/fuel ratios must be maintained, on a diesel there is no "throttle" as such. At any given speed, the engine inhales the same full load of fresh air no matter what the position of the gas pedal; varying the power is sim-



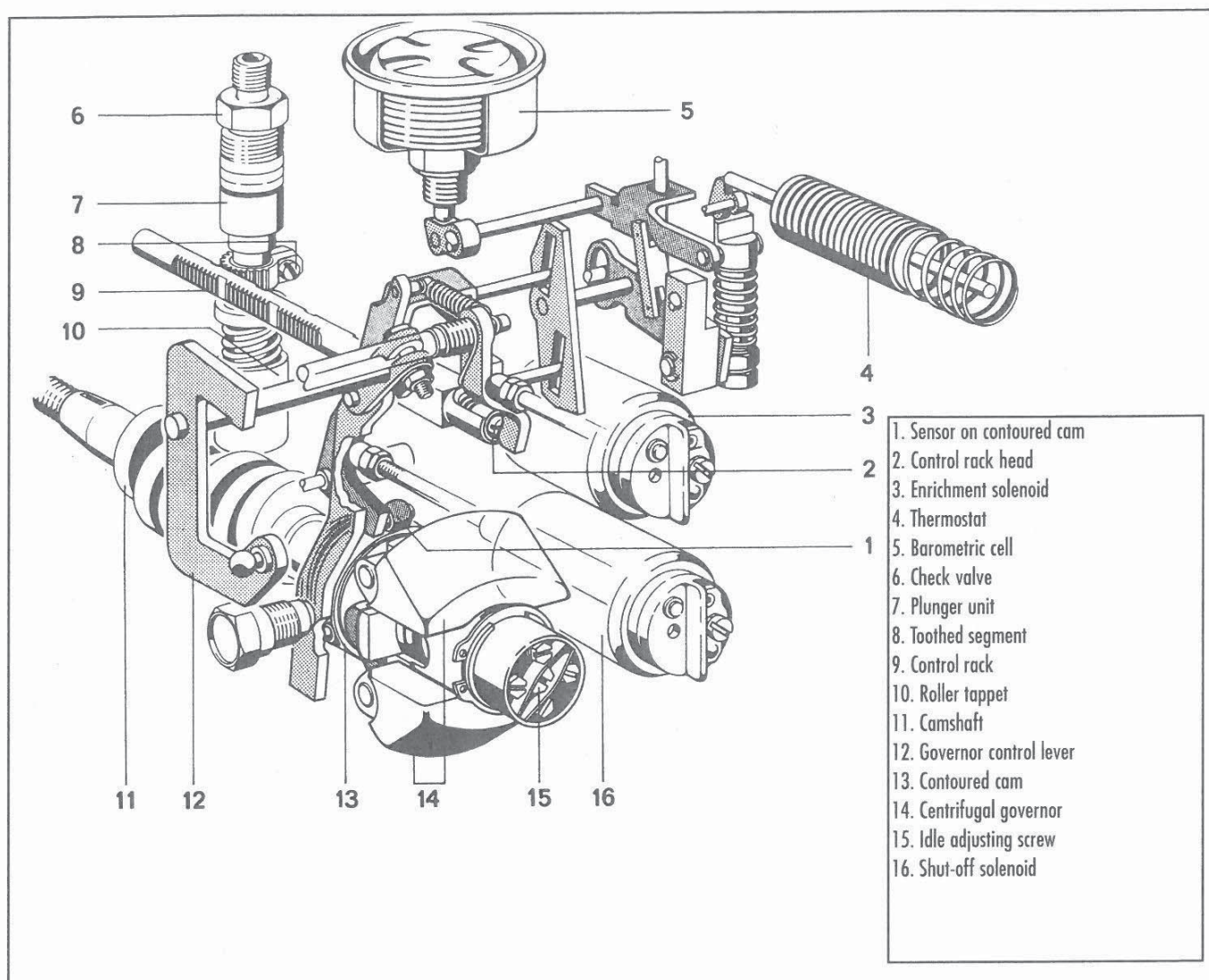
The resemblance to Diesel engine practice is clear in this exterior shot of the 300SL engine. Gasoline's lower lubricity, compared to diesel fuel, compelled the use of expensive, wear-resistant alloys in the injection pump. (Daimler-Chrysler Archive)

ply a matter of injecting more or less fuel. Power is regulated, in other words, by adjusting the mixture strength. In the simpler diesel systems, at least, this is directly controlled by the driver's right foot—the engine runs extremely lean at light throttle, and rich to the point that it visibly smokes when maximum power is demanded. In fact, it is the smoky exhaust that sets the limit for the rated power of a diesel engine; more power would be available simply by injecting yet more fuel—if we were prepared to put up with the smoke. (Even at full power, the amount of fuel injected falls short of a

stoichiometric mixture—diesels are always running "lean," which is one reason they can never produce as much power as a gasoline engine of the same displacement.)

The inefficient breathing of any piston engine, gas or diesel, at speeds well away from the rpm at which torque peaks means that the mass of air inhaled per revolution, at any given throttle setting, will most definitely not be constant across the speed range. Because of the gasoline engine's finicky appetite, the amount of fuel mixed in with that air also has to be varied on the basis of the quantity of air inhaled, and not just

Bosch Fuel Injection Systems



The complexity of the mechanical controls for the Bosch system used on the 300SL is mind-boggling. Fuel delivery of the pump (only the camshaft is shown, 11) is controlled by the rack (9). The position of the rack is governed by engine speed, via the centrifugal weights (14) acting on a contoured cam (13), and by throttle position. Other factors are accounted for by a barometric capsule (5) and a temperature sensitive device (4). (Robert Bosch Corporation)

according to speed and throttle position.

Adapting to Gas Engines—To adapt a diesel jerk pump to gasoline operation, then, means that the rack (see sidebar, page 20) cannot simply be directly hooked up to the loud pedal; the relationship between the two has to be modulated by some other control(s). The usual way this is done is by adding a set of centrifugal weights, somewhat like those in a traditional ignition distributor, to provide a signal proportional to engine speed, plus a diaphragm that "reads"

manifold vacuum, a fairly close approximation of engine load. These additional control devices are connected to the linkage controlling the rack via a system of cams and links, with the shape of the cams tailored according to the idiosyncrasies of the engine's breathing.

The monkey motion of these additional levels of control added even more to the substantial cost of the pump itself, demanded meticulous adjustment, and its complexity mocked the essential simplicity of the

basic pump. Indeed, one observer at the time mused aloud that if all engines had worn fuel injection all along, then someone invented the carburetor, he would have been hailed as a genius.

Injector Location—Apart from the mechanical complexity of the "add-on" speed- and load-sensing mechanisms at the pump, another problem confronted this adaptation from diesel to gasoline—that of shielding the injector nozzles from the heat and fury of combustion. Surprising as it may seem, gasoline combustion chambers see higher peak temperatures than diesels. On their M196 Formula One engine, Mercedes dealt with this by fitting the injector into the side of the cylinder, where it was masked by the piston at TDC and thus shielded from the worst of the inferno.

Certainly, timed direct injection deals with the potential issue of fuel consumption, especially with radical valve timing, and the high pressure spray of these systems is favorable for atomization, but against these advantages were set the problems noted above, plus the issue of the power required to drive a pump that has enough grunt to blow open the stiff check valves in the injectors and at the pump outlets, used respectively to prevent combustion pressure from blowing back through the injection lines and to keep the lines fully charged. Nevertheless, the fact that engines operate quite happily on carburetors made it clear from the outset that there was no absolute need for precision timing of the fuel delivery in a spark-ignition engine.

Granted, a carbureted engine would probably make a little less power than a similar one with direct injection, but some part of this might be attributed to the fact that a separate injector for each cylinder eliminates the problems of a central mixer, rather than to the timing. Of course, the same thing can be achieved with carburetors if a separate carb is provided for each cylinder. That, however, would be heavier, probably more

expensive and arguably even more complex, depending on the number of cylinders. Thus, while a handful of other Formula One teams and upmarket European sports car manufacturers also used Bosch gasoline injection, to the best of this writer's knowledge, all these others left the injector just outside the combustion chamber, aimed at, and spraying against, the head of the intake valve.

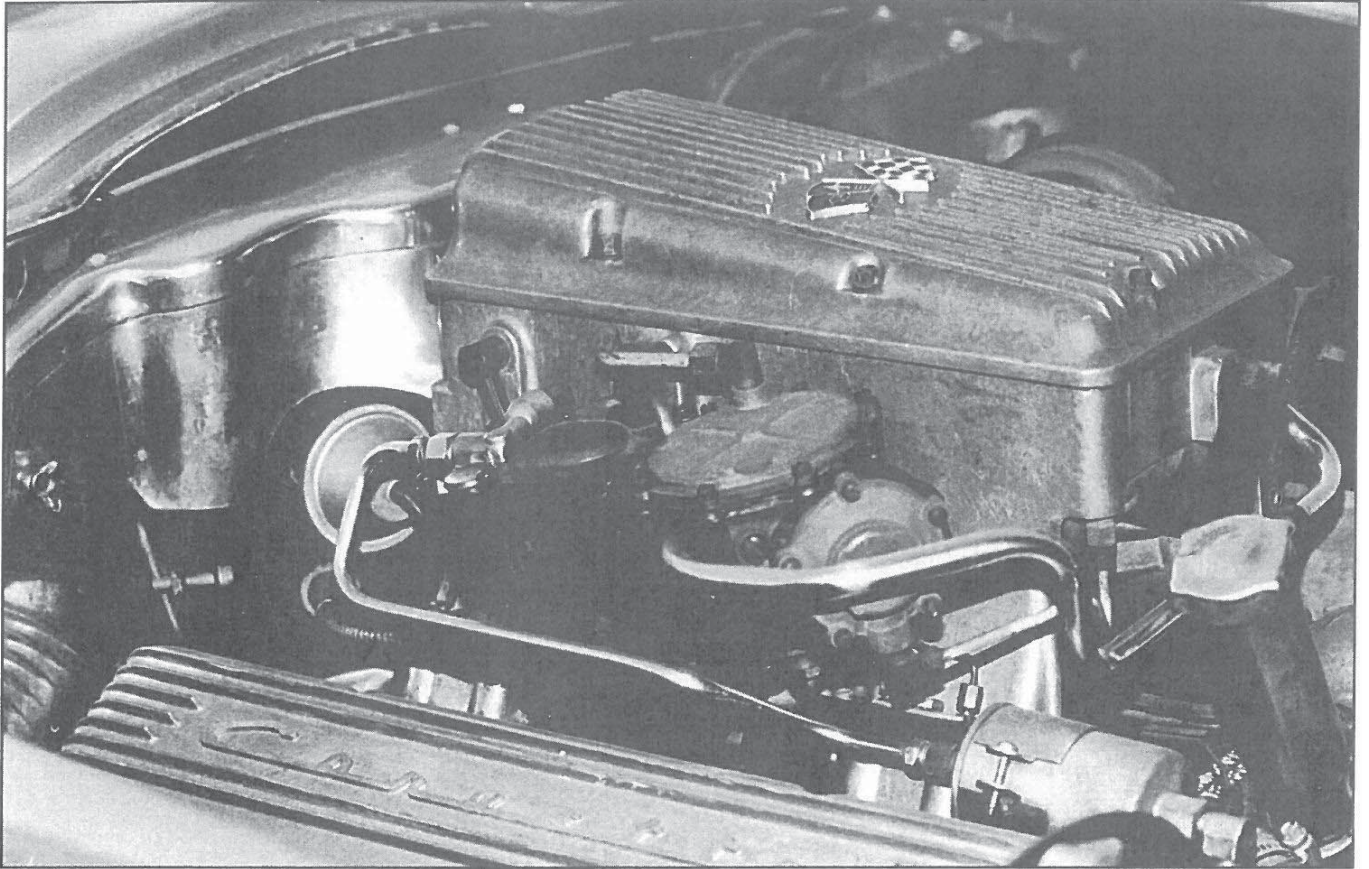
But if you decide to locate the injectors outside the combustion chamber, then you don't need to fight combustion pressure, so you don't need a very high pressure pump. Also, if the injector lies outside the cylinder, then you don't need to time the injection so it occurs with both valves closed. In that case, why bother to time it at all? And if the timing of discrete squirts is not crucial, then why does the fuel have to be delivered in individual pulses?

Early Continuous Injection Systems

These questions had occurred to a good many people, and an alternate form of fuel injection came into being. In these, fuel is injected continuously, rather than in pulses, and at very much lower pressure than a "jerk" pump provides, though still much higher than the small pressure difference that exists between a carburetor venturi and the atmosphere.

The Wright brothers used a primitive form of this continuous injection, as the arrangement is called, on the engine of their first successful aircraft. An engine-driven pump of the "positive displacement" type—one which supplies a fixed quantity of fuel at each revolution—was used, to continuously spray fuel directly into the engine air intake. It should be clear that while a satisfactory mixture strength might be arranged at full throttle—presumably after some tinkering with the size and/or drive ratio for the pump—with no means to reduce the pump output the mixture must have become hopelessly rich at anything less than full throttle.

Bosch Fuel Injection Systems



Although Stu Hilborn had proved, years earlier, that such a system could work on the track, Rochester were the first to demonstrate the practicality of the simpler, continuous flow fuel injection system, as on this early "fuelie" Corvette. (Dave Emanuel)

But then the Wrights' had no immediate interest in operation at part throttle!

Winfield's FI System—About the same time that Bosch and Mercedes were beginning their experiments with timed, high-pressure, direct gasoline injection in the mid-1930s, Ed Winfield developed a low pressure, continuous injection system, intended for Indy cars. Winfield's design had provision for varying the fuel flow with throttle position, but a lack of interest from potential customers forced him to eventually abandon the scheme, and he allowed his patents to lapse. It was not until fifty years after the Wrights that racers had access to a continuous injection system that dealt with the part-throttle problem.

Hilborn Injection—Stu Hilborn achieved this by incorporating a controllable spill valve, connected to the throttle linkage, that

tapped off (and returned to the tank) some portion of the pump's output, according to how wide the throttle was opened.

On the face of it, this might appear to fill the bill, but the assumption built-in here is that an engine, at any given throttle setting, needs a fixed quantity of fuel per crank revolution, and we have already seen that this is mistaken, because of the way an engine's ability to take a full breath varies across its speed range. As a result, gasoline FI systems that use just engine speed and throttle position to determine fuel quantity can be arranged to provide a suitable mixture strength only under very limited circumstances, and must inevitably miss the mark by a large margin under others.

Still, schemes such as Hilborn's have a long and honorable history when used with methanol fuel. As discussed in more detail

in Chapter 8, methanol will tolerate wide variations in mixture strength with little effect on power output, which helps mask the inherent shortcomings of such a simple system. In this case, rpm and throttle position alone provide a sufficiently accurate measure of fuel flow requirements, so it is not necessary to measure the rate of airflow at all. Since there is no need to measure it, there is no need for the restriction of a venturi or any other airflow sensor stuck in the airstream. This improves engine breathing and so promises more power just for that reason alone.

Rochester FI System

For a street engine running on gasoline, however, any purely mechanical system, whether injecting continuously or in a series of pulses, needs to somehow measure or compute the rate of airflow into the engine, and to use that information to modulate the delivery of fuel so as to keep the air/fuel ratio appropriate to the circumstances. We have discussed how this can be achieved with a jerk pump, but what about a continuous gasoline injection system?

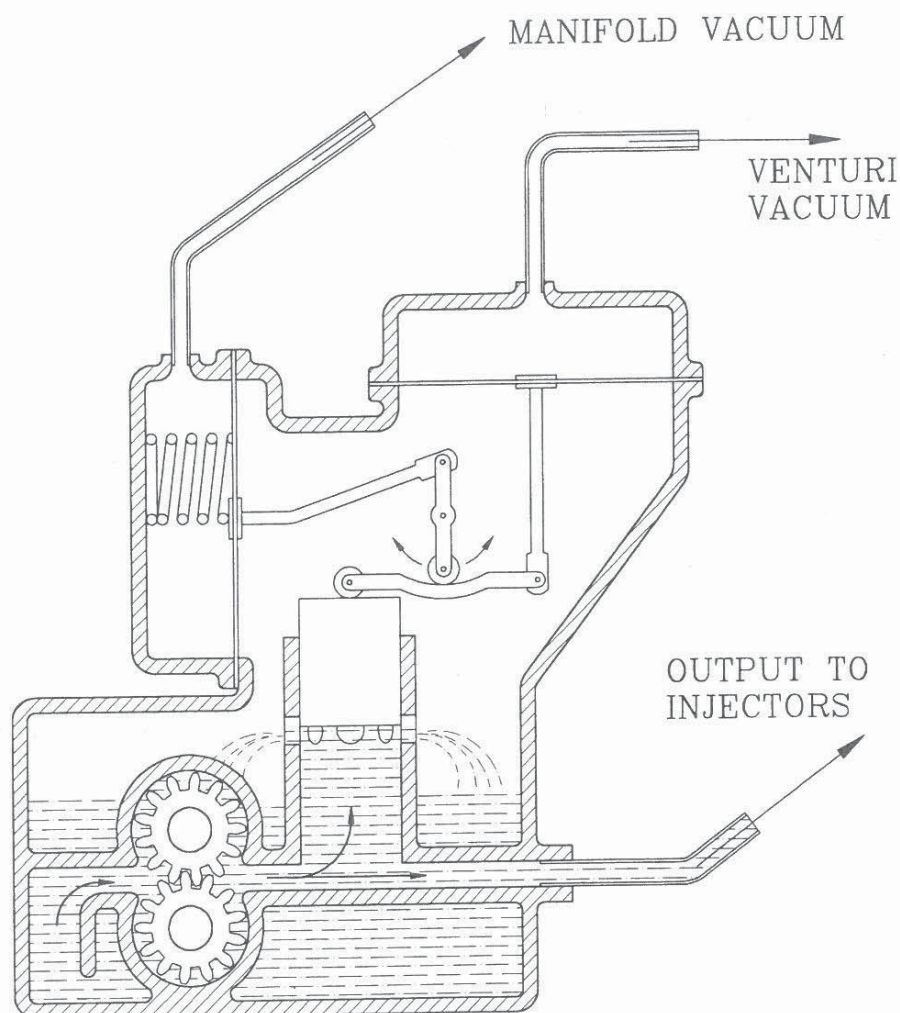
There have been many such systems throughout automotive history; one of the more recent and more widely used of these is the Bosch K-Jetronic system, described in Chapter 6. Nevertheless, the example most of us remember best is likely the Rochester system used on "fuelie" Corvettes from 1957 to 1965. The heart of the Rochester system is a spill valve that serves as a regulator of fuel quantity—indeed, it is the only mixture control element in the whole system. Fuel is fed to the spill valve by an engine-driven gear-type pump that simply serves as a source of fuel at medium-high pressure (200psi at maximum), with the pressure tending to rise with increasing rpm. The spill valve consists of a sleeve with a number of ports arranged radially in its side, plus a plunger that covers or exposes these ports, according to its position. The position

of the plunger, in turn, depends on a balance between the fuel pressure and a force applied to the top of the plunger by a mixture control mechanism.

Increasing fuel pressure tends to raise the plunger, thus uncovering the sleeve ports and so bleeding off some of the pressure. The mixture control device, shown schematically in the following illustration, amounts to a simple mechanical computer that applies a spring force to the top of the plunger, modified by the forces acting on two diaphragms—one connected to the intake manifold, and one to a venturi through which all the air entering the engine has to pass. The eight individual port injectors downstream of the spill valve thus see a fuel pressure that depends on a balance between the supply pressure and the forces acting on the diaphragms, and their output varies accordingly.

The Rochester system is strikingly simple, especially when compared to a jerk pump having the controls needed for street use on gasoline, and none of its parts require exceptional precision or expensive materials for their manufacture. Although the injectors themselves had to be individually calibrated and installed as a matched set, given sufficiently large production numbers it seems likely that the system overall would have been little (if any) more expensive to make than a pair of four barrel carburetors. Nevertheless, it remained a somewhat rare and costly performance option, and justified its price premium by offering more power. Partly this was because the near-elimination of cylinder-to-cylinder variations in mixture strength permitted a higher compression ratio (CR) than was safe with carburetors, but an advantage still showed even when the CR was unchanged. We might attribute this to three sources: nearly identical air/fuel ratios at each cylinder, elimination of manifold heating, and a reduction in the breathing restriction imposed by the venturi, compared with a carburetor. Although a venturi

Bosch Fuel Injection Systems



The heart of the Rochester system's control unit was a spill valve that returned to the tank a certain fraction of the fuel delivered by the gear-type pump. The position of the piston in the spill valve was determined by two diaphragms—one hooked to a manifold vacuum, one to a single large metering venturi that fed the air box. A system of links and rollers linked movement of the diaphragms to the spill valve piston.

was needed to measure air quantity, the amount of pressure drop needed to provide a useful signal to the mixture control unit was much less than that required to lift gasoline out of a carburetor's discharge tube.

In its day, the Rochester FI was regarded by the auto industry and the public alike as very much a high-tech piece, and possibly the final word in fuel delivery. Ironically, at about the same time that GM was introducing this system, another completely different form of fuel injection was being unveiled, to much less public acclaim, yet this one would

eventually sweep away not only the carburetor, but most other forms of fuel injection as well.

The First Use of Solenoid Valves

The basis of this little-hailed breakthrough was the use of an electrically operated solenoid valve as a fuel injector. Supplied with fuel at a modest constant pressure, it squirts at a constant rate as long as an electric current is applied to the solenoid windings, and stops when the current is turned off. Thus, the quantity of fuel injected depends simply

on how long the juice is turned on. If one electrical pulse is delivered every second revolution of the crankshaft (for a four-stroke engine), then the air/fuel ratio of the mixture delivered to the engine can be controlled simply by varying the length of each pulse.

The idea of using a solenoid valve as a fuel injector is a surprisingly old one. Such a scheme had been developed by an engineer named Kennedy at the Atlas Imperial diesel Engine Co, in 1932, and an installation appeared on a marine engine exhibited at the New York Motor Boat Show in 1933, around the same time that Bosch and Mercedes in Germany, and Winfield in the U.S. were beginning their experiments. Given the primitive state of the electrical sciences back then, it is mildly astonishing that the thing worked at all, yet in 1934 a similar but smaller engine was installed in a truck and driven successfully from Los Angeles to New York and back.

The "Electrojector"—A quarter century later, A.H. Winkler and R.W. Sutton of the Eclipse Machine Division, Bendix Aviation Corporation, announced the fruits of four years' work at Bendix to develop an FI system for gasoline-fuelled automobiles using such a solenoid-controlled valve as an injector, in conjunction with an electronic control unit, or (as they termed it) "brain box." They described this combination of components as "electronic fuel injection"—surely the first time this phrase was ever used. The resulting "Bendix Electrojector" comprised a separate port injector for each cylinder, each supplied with fuel from a common rail at a regulated 20 psi. Sensors responsive to manifold pressure, engine speed, atmospheric pressure, and air and coolant temperature fed their various outputs to the "brain box"—the electronic control unit (ECU)—which then calculated the instantaneous fuel requirements of the engine. On the basis of that calculation, the ECU delivered a pulse of current to a rotary distributor which fed

that current pulse to each solenoid valve in turn, causing each successively to open and supply fuel to the cylinder it served. The quantity of fuel delivered by each injection was simply a matter of the duration of the pulse—as long as it was "on," the injector would deliver fuel at a constant rate. At idle, the pulses were of very short duration; at maximum load, the injectors were open for as much as 150 crankshaft degrees.

At the beginning of the development of the Electrojector, about 1951 or 1952, the triggering signal for the ECU was provided by a mechanical commutator—a simple wiper making intermittent contact with a segmented brass ring, as the engine-driven ring swept past. That soon gave way to triggering by an extra set of contact points in a housing sandwiched under the ignition distributor. In at least the three earliest versions, the working bits in the ECU were . . . vacuum tubes! One such unit was installed on the V8 engine of a 1953 Buick, and used to demonstrate the system to the auto industry.

By 1957, when it became available to the public in very limited numbers, the electronics were, thankfully, all transistorized. This eliminated the need to wait for the tubes to warm up before the system became functional, shrank the size of the ECU and substantially reduced the electrical power required to run it, and doubtless greatly improved the system reliability—vacuum tubes do not take kindly to be rattled around! The Electrojector was listed by American Motors as an option for the 1957 Rambler Rebel, but very few cars so equipped were ever sold. A year later, Chrysler offered it on the 300D and DeSoto Adventurer, and several hundred with the option were manufactured in 1958 and 1959.

Despite occasional problems from external electrical fields causing the brain box to go haywire, and a degree of unreliability of the injectors themselves, on the whole the system worked satisfactorily and matched Bendix's claims of modest (5 percent)

improvements in both fuel economy and power, compared to a similar engine fitted with a pair of four-barrel carburetors. The teething troubles were eventually worked out, and by 1965 Bendix had a fully developed production version ready to go.

At this point, rather than try to produce and market the Electrojector themselves, Bendix decided instead to enter into an agreement with the Robert Bosch Co., which gave Bosch access to the Bendix patents. From a business point of view, this made a good deal of sense. In view of their reputation in the auto industry as an established OEM supplier—one with wide experience both in fuel injection and in electrical and electronic equipment—Bosch was clearly in a better position to exploit the potential market. And fuel prices in Europe were many times what they were in the U.S. of the 1950s, justifying a much higher initial cost there, in exchange for the fuel economy benefits alone.

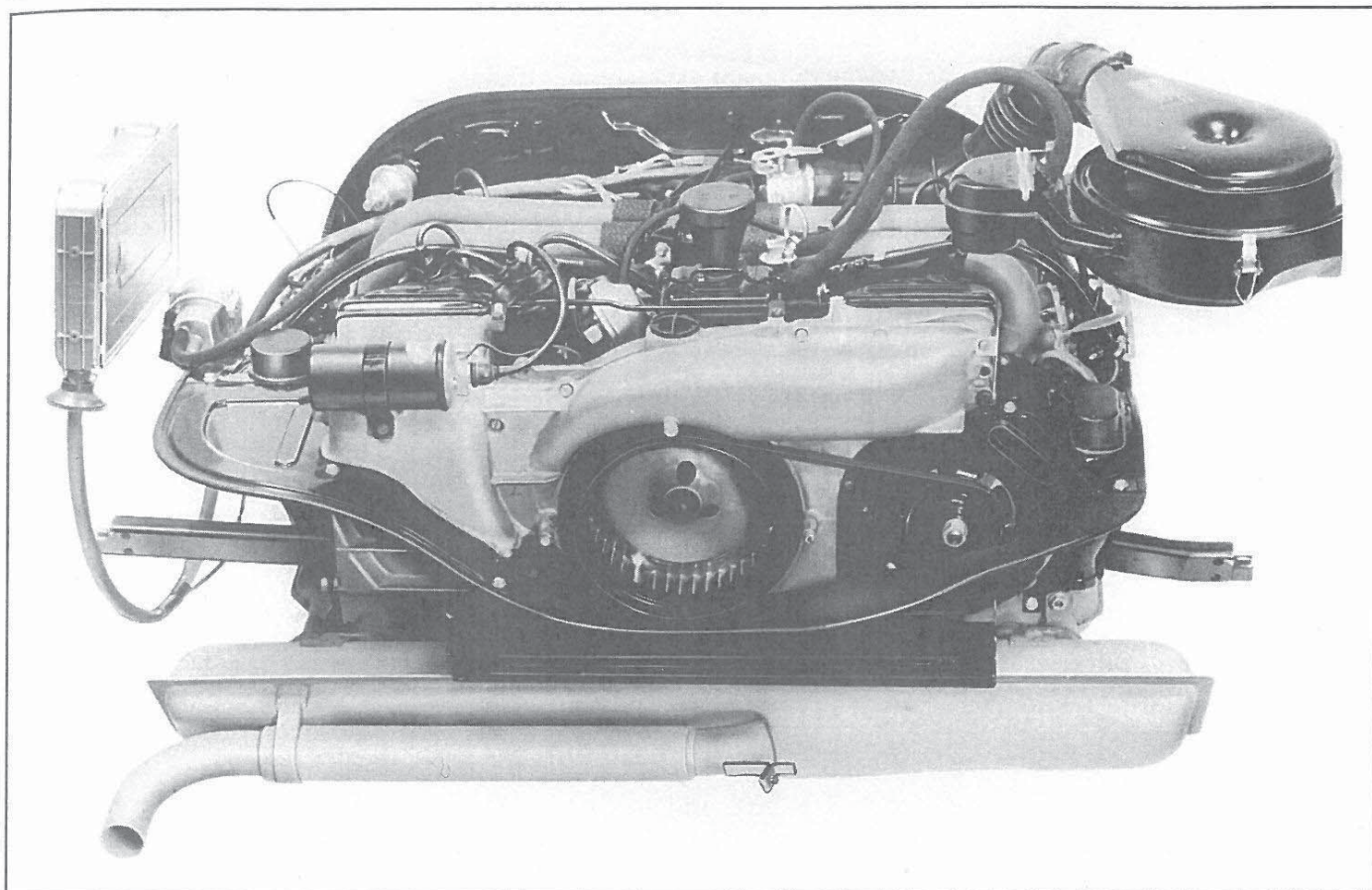
Bosch Jetronic FI

In most respects, the Bosch "Jetronic" FI that first appeared on the 1967 Volkswagen 1600, was functionally identical to the Electrojector system described by Winkler and Sutton ten years earlier. Apart from taking advantage of advances in electronics to produce a more compact ECU, this system—since identified as D-Jetronic, to distinguish it from later Jetronic versions—differed from the Bendix prototype in one respect. In the Electrojector, the housing that was piggybacked onto the ignition distributor and which contained the triggering contact points also contained a rotor, just like an ignition rotor; this distributed the timed pulse to each injector sequentially. In the D-Jetronic as applied to the four cylinder VW, there was no rotor, but there were two sets of triggering contacts, one for each pair of cylinders. Each set of contact points drove two injectors simultaneously, timed so that one cylinder received its injection while its

intake valve was open; for the other cylinder, the injection occurred with valves closed. Fuel thus accumulated in the port, awaiting that cylinder's next intake stroke.

We have mentioned earlier that the mere fact that a carbureted engine will function in a satisfactory way should make it obvious that injection timing is, to say the least, not critical, and the power gains realized with a few continuous injection systems tended to confirm this view. The matter was pretty much clinched in 1959, when Mercedes successfully used a Bosch jerk pump with just two plungers, each feeding three engine cylinders, on its six cylinder 220SE. This must all have improved the comfort level of the Bosch and VW engineers who adopted the hop-skip timing used on the D-Jetronic, an arrangement that allowed eliminating the rotor used in the Electrojector.

The Electrojector-based, intermittent injection "Jetronic" systems from Bosch have undergone successive improvements and refinements over the years since the D-Jetronic's introduction. Apart from the electronic innards of the control unit gaining computing power while shrinking ever further in size and weight, and the use of all-electronic triggering (rather than contact points), the major differences between one variant and another have been in the way the quantity of air supplied to the engine is measured. These basic differences between systems are reflected in the nomenclature used: L-Jetronic, LH-Jetronic, etc. In addition, most later systems accept inputs from additional sensors, most notably an oxygen sensor, also called an O₂ sensor or lambda sensor. As mentioned in the previous chapter, this is a device fitted to the exhaust system that "sniffs" the exhaust gas to gauge its oxygen content, thereby providing a direct measure of what the air/fuel ratio actually is at any given moment, rather than obliging the system to depend totally on a set of pre-programmed "maps" or "look-up tables" that predict what the ratio should be for a



With the rear-mounted "pancake" engine nestled under the apparent floor of the trunk, earlier carburetted versions of the VW 1500/1600 were notorious for vapor lock problems. This is the later 1600 engine with Bosch D-Jetronic injection. The ECU is visible at the far left. (VW Canada).

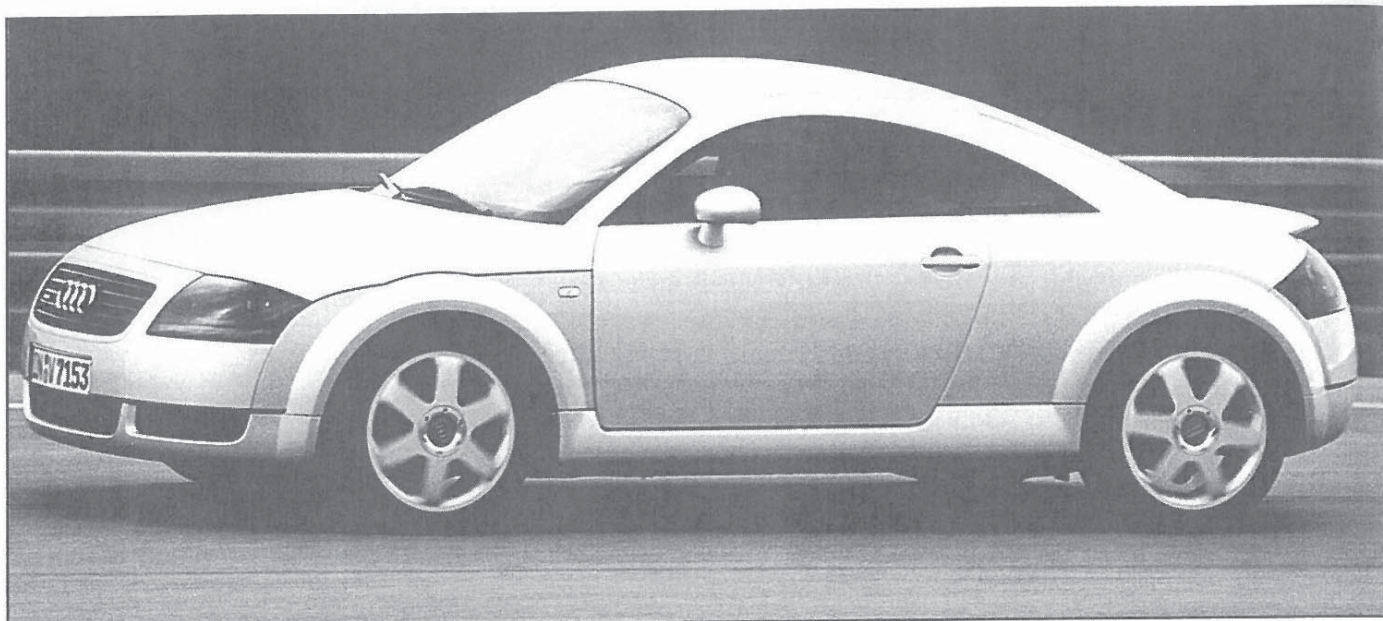
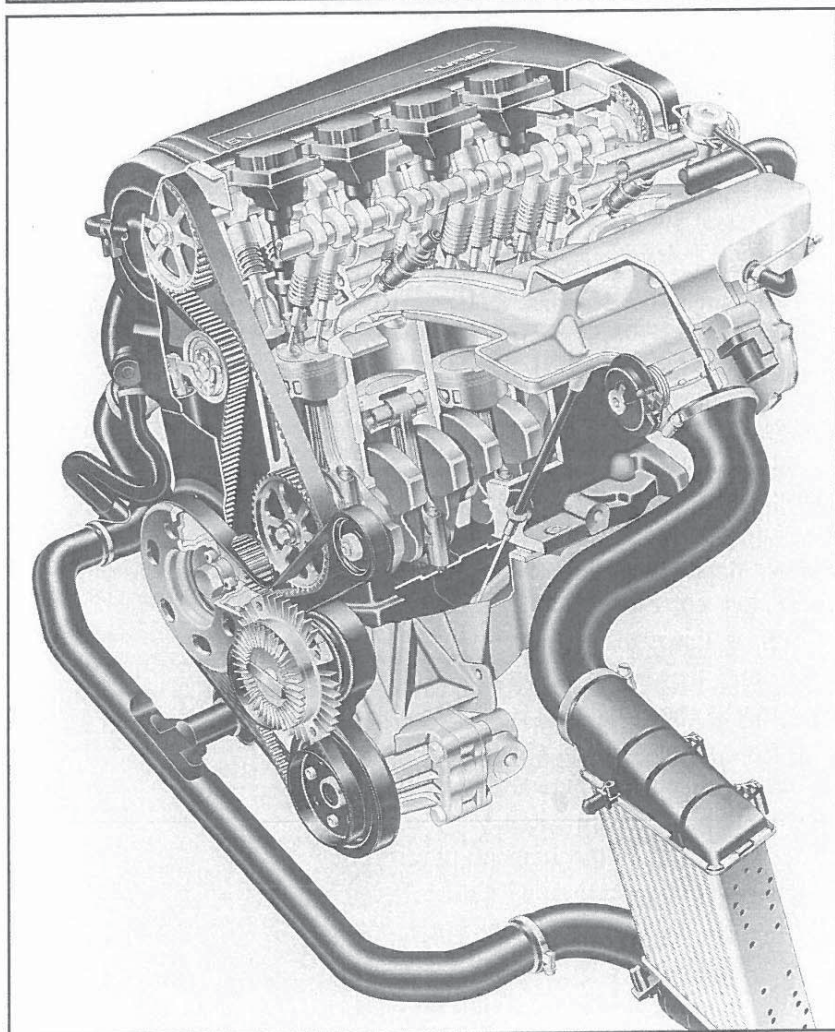
given set of operating conditions. (See also the sidebar "The Oxygen Battery.")

In many cases, control over ignition timing has been integrated into the ECU of these systems, in which case they are termed "Motronic," again with variants according to the method of air metering. Over time, an increasing number of functions have been combined into the Motronic system, including integrated control over automatic transmission shift points and torque converter lockup, the waste-gate on some turbocharger-equipped cars, selection among different length intake ports on some others, and authority over the variable valve lift and/or timing found on a few vehicles. We discuss these Motronic systems in some detail in Chapter 4.

K-Jetronic—At the start of the new mille-

nium, the principle of intermittent injection using electronically controlled, electrically powered solenoid injectors is now universal on domestic passenger cars, and near-universal on models imported from Europe and Asia. Yet not everyone was an immediate convert to this plan. It seems ironic now that even after the pulsed (intermittent) Jetronic system was commonplace, Bosch felt obliged to introduce a continuous injection system, the K-Jetronic (The "K" stands for *kontinuierlich*, German for continuous). Early versions of this were purely mechanical; in response to ever-tightening emissions standards, later ones—such as KE-Jetronic and corresponding Motronic versions—added a degree of electronic control. In part this apparent preference for an all-mechanical system may have been due to a powerful conservative

Bosch Fuel Injection Systems



Intermittent electronic fuel injection allows modern cars to combine high power with driveability, economy, and low emissions. (VW Canada).

streak among some manufacturers and their engineers. In part, too, it may have been a matter of economics—the intermittent systems may have been more expensive at the OEM level. Although there were a few hold-outs until the early 1990s, most of the various K-Jetronic systems were history by the mid 1980s. These now-superseded (not to say obsolete) systems are dealt with in Chapter 6.

We will not be dealing with a handful of sub-variants of Jetronic and Motronic FI that are used on vehicles not imported into North America.

To summarize, then, the essential difference between carburetors and fuel injection systems is that a carburetor uses the pressure drop within its venturi both to measure the airflow and as the means to propel the fuel into the intake air stream, whereas FI mechanically forces the fuel into the intake air in an amount usually determined by a separate airflow or manifold pressure sensor, acting on an electronic or mechanical system that actually controls the quantity of fuel injected.

Basic Types of Fuel Injection

All fuel injection systems can be divided up into two basic types: those that inject fuel

continuously and those that inject intermittently, in a series of pulses. Again, in either case control over how much fuel is injected can be achieved either mechanically or electronically. In the present context, only the old, diesel-type jerk pump uses mechanical control over intermittent injection; all Bosch intermittent systems use electronic control. Bosch K-Jetronic continuous injection systems—sometimes called CIS—may use purely mechanical control, as in the original K-Jetronic, or may include some degree of electronic fine-tuning on top of the basic mechanical computation.

Many FI applications, especially in newer cars and in older high-performance ones, use a separate injector for each cylinder. This greatly reduces the quantity of liquid fuel laying on the walls of the ports. If the injectors are located very close to the intake valves and the intermittent injections are sequential, the possibility of maldistribution of the mixture between one cylinder and another is essentially eliminated.

Many others, though, are of the "throttle body" type, in which the fuel is sprayed into and mixed with the air in one centralized location, then conducted to the cylinders through a thus "wet" manifold, just like that of a carbureted engine. This would seem to put it on the same footing as a single carb in terms of mixture distribution, but the fact that the fuel is injected under pressure means that it emerges from the injector nozzles as a very fine spray, with droplets so

small that they have a much better chance of evaporating than is often the case with the lumpy soup coming from a carburetor. While a TBI system is thus likely to be better in this regard than a carb, an individual injector at each intake port can do even better.

One of the further advantages sometimes argued for FI is that it achieves the maximum possible engine power because it does away with the venturi. That is not quite true, because it is a basic principle of science that you cannot measure something without affecting the thing you are measuring. Certainly a venturi measures airflow, and in the process it also restricts that flow, but electronic fuel injection (EFI) systems that use a spring-loaded flap valve or a "hot wire" airflow sensor (the current needed to keep the wire at a constant temperature is used as the air-quantity signal) both offer at least some restriction to flow, too, although a hot wire (or hot film) sensor will impede airflow much less than any carburetor venturi.

The ultimate sophistication is to provide an oxygen sensor (O_2 sensor, lambda sensor) in the exhaust pipe and to correct the mixture strength from moment to moment, based on that information; an excess of O_2 calls for a richer mixture, and vice versa. For all these reasons, EFI with an injector for each cylinder is unarguably the state-of-the-art for fuel-air mixing and delivery.

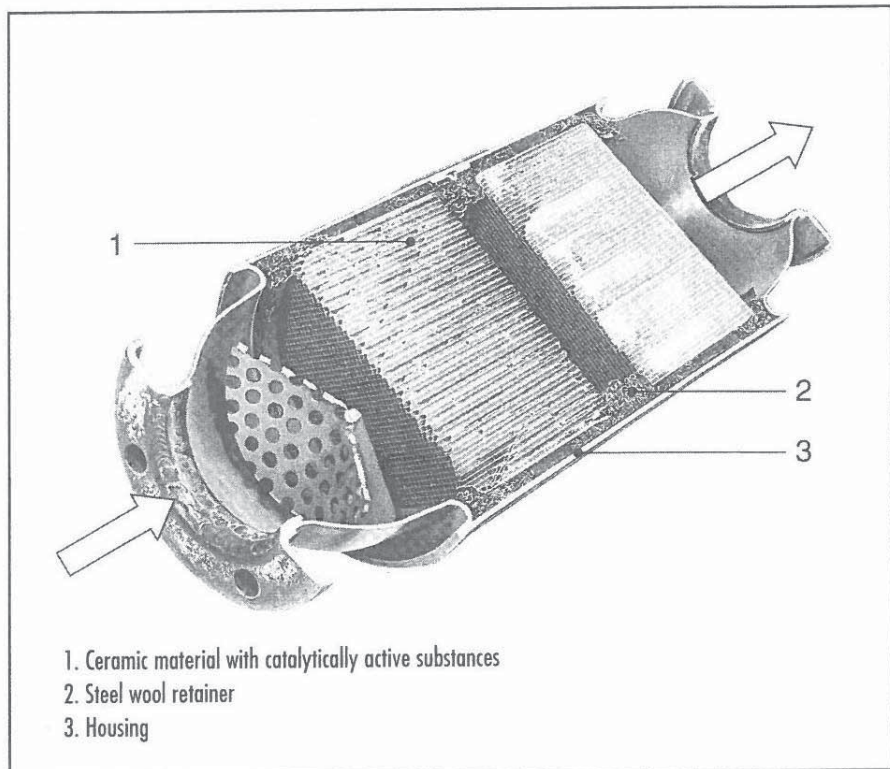
THE OXYGEN BATTERY

Rich mixtures, we explained in Chapter 1, mean that some of the hydrocarbon molecules in the fuel entering the engine pass out into the exhaust unreacted—there simply aren't enough oxygen molecules available for them to combine with. These unburned hydrocarbons are a source of air pollution, and legislation severely limits the quantity of them that is allowed to escape into the atmosphere. The shortage of oxygen also means that the reaction with the hydrocarbons that do burn is chemically incomplete—instead of the hydrocarbon-oxygen pairings forming CO_2 and H_2O , many of them wind up as CO —carbon monoxide. Emissions of carbon monoxide are also strictly controlled by law.

Lean mixtures would keep down the CO levels, and as long as they were just a little lean, the HC levels, too. Ironically, excessively lean mixtures actually raise HC levels above those found with a stoichiometric mixture, because as the mixture approaches the "lean limit"—the air/fuel ratio at which the mixture simply won't ignite—some cylinders misfire, so all of the HC in that particular cylinderful goes straight out the exhaust unburned.

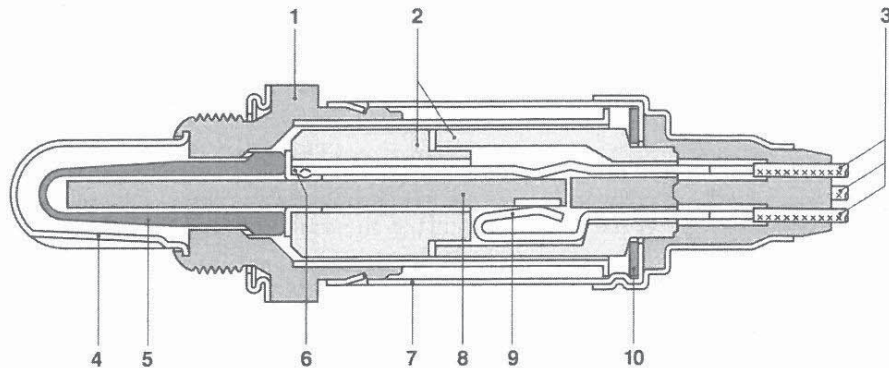
But even mixtures just lean enough to minimize CO and HC create high combustion temperatures. Too-hot combustion encourages some of the nitrogen (N) in the air, normally thought of as inert, to combine with some of the excess oxygen, to form various oxides of nitrogen— NO_x . NO_x is also a pollutant—it is the principle source of photochemical "smog"—so the amount of it, too, is limited by law. (Damned if you do; damned if you don't!) A compromise setting won't work, either. Anything lean enough to meet HC and CO limits would bust the NO_x limit, and anything rich enough to avoid excessive NO_x would lead to illegal levels of HC and CO . (Damned even if you sit on the fence!)

The solution arrived at by auto makers, beginning about 1975, has been the "three-way catalytic converter." In essence, this device splits the oxygen from the NO_x and helps it combine with the CO to make CO_2 , and with the HC to make H_2O , and more CO_2 . This works well enough to satisfy the G-men . . . provided that the mixture was neither too rich nor too lean in the first place. And the three-way catalytic converter is mighty fussy indeed about just how close to the stoichiometric ideal the ingoing mixture has to be; mixtures that are less than one percent either



Construction of a three-way catalytic convertor. Convertors are essential to meet current emissions limits, but can only do their job if the air/fuel mixture fed to the engine is held very close to the chemically correct ratio. And to achieve that, a lambda (oxygen) sensor is essential. (Robert Bosch Corporation).

1. Probe housing
2. Ceramic shield
3. Electrical connections
4. Shield tube w/slits
5. Active ceramic sensor layer
6. Contact
7. Shield



Lambda sensors have a voltage output that rises from near zero in the absence of oxygen in the engine exhaust to a bit less than one volt when small traces of it are present. This characteristic enables an electronic fuel injection system to control fuel delivery on the basis of the sensor's reading of the exhaust gas. The sensor only works when hot, however. To reduce the time between engine start-up from cold and the sensor reaching its operating temperature, some versions are electrically heated. (Robert Bosch Corporation).

side of the chemical ideal render the converter next to useless.

No fuel injection system yet developed (and certainly no carburetor!) can hope to control the mixture strength that closely, based simply on a set of values derived from the characteristics of some similar engine in a test lab. The device that enables EFI to "know" just what the mixture strength needs to be at any instant for any particular engine out in the real world is the "lambda sensor."

In construction, the lambda sensor consists of a hollow ceramic bulb, with both inner and outer surfaces covered with a microthin layer of platinum. For protection, the outer surface is surrounded by a shield. This bulb is inserted into the exhaust system so that its exterior surface is washed by the exhaust gas stream, while its hollow interior is exposed to the atmosphere. In operation, the lambda sensor acts like an oxygen-powered battery: a difference in the oxygen level of the gasses surrounding the two surfaces of the sensor causes it to generate a small voltage.

As you might expect, there is very little oxygen in the exhaust of an engine running on a stoichiometric mixture, yet there is some—about 0.5 percent. Surprisingly, even with a mixture that is about five percent rich, there is still about 0.2 percent oxygen. Equally surprising, a mixture that is five percent lean will only contain about one percent oxygen. A particularly useful feature of the lambda sensor is that its output voltage changes very dramatically right around the point of stoichiometry; very small changes (a few tenths of one percent) in oxygen content lead to large changes in the sensor output. While the sensor will put out about 900 millivolts (0.9v)

when the exhaust oxygen content corresponds to a lambda of about 1.2 (that is, the mixture is about twenty percent lean—an air/fuel ratio of a bit less than 12:1), and while its output drops to near zero at about $\lambda=0.8$ (an air/fuel ratio of near 18:1), almost all of that voltage change occurs within one percent of stoichiometry. The voltage output of the sensor is fed back to the ECU, which slightly reduces or increases the duration of the on time for the injectors, according to whether the sensor reports the mixture is richer or leaner than stoichiometric. This "closed-loop" control maintains the air/fuel ratio within 0.1 percent of the chemical ideal, thus allowing the converter to do its job.

In practice, an output from the sensor of 450mv is used by the ECU as the threshold of the lean limit and 500mv as the rich limit. This means that the air/fuel ratio actually oscillates back and forth (though always within 0.1 percent of stoichiometry) as the ECU responds to the voltage signal from the sensor. The time required for the sensor output voltage to vary between these limits depends on its temperature. At idle, when the gas temperatures are low—say 300° C (about 570° F)—that time interval might be as much as a couple of seconds; under typical road loads, the exhaust temperature is more like 600° C (1100° F), and the "cycle time" is reduced to less than 50 milliseconds (1/20 second).

The sensor is essentially inoperative at temperatures below 300° C, so during engine startup and initial warm-up the ECU ignores the sensor voltage and falls back on a set of internal "maps," until such time as the signals it is getting from the sensor make sense. Accordingly, emissions are inevitably increased during this warm-up period, so to reduce the wait for the sensor to reach operating temperature, some sensors have an internal heating element to bring it on-line as quickly as possible.

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