

Calculation of Injector Pulse Width. The base pulse width is determined from the required fuel mass flow rate (F_m) and an empirical injector constant. The injector constant is determined by the design of the injector and is a function of the energized time versus the flow volume. This constant is normally determined with a constant differential pressure across the injector (from fuel rail to intake manifold). When the pressure across the injector does not remain constant (i.e., there is no pressure regulator intake manifold vacuum reference), an entire map of injector constants for different manifold pressures may be required.

The effective injector pulse width is a modification of the base pulse width. The base pulse width is adjusted by a number of correction factors depending on operating conditions. For example, a battery voltage correction is required to compensate for the electromechanical characteristics of the fuel injectors. Injector opening and closing rates differ depending on the voltage applied to the injector, which affects the amount of fuel injected for a given pulse width. Other common correction factors may include hot restart, cold operation, and transient operation corrections. Figure 12.2 is a flowchart of a typical injector effective pulse-width calculation method.

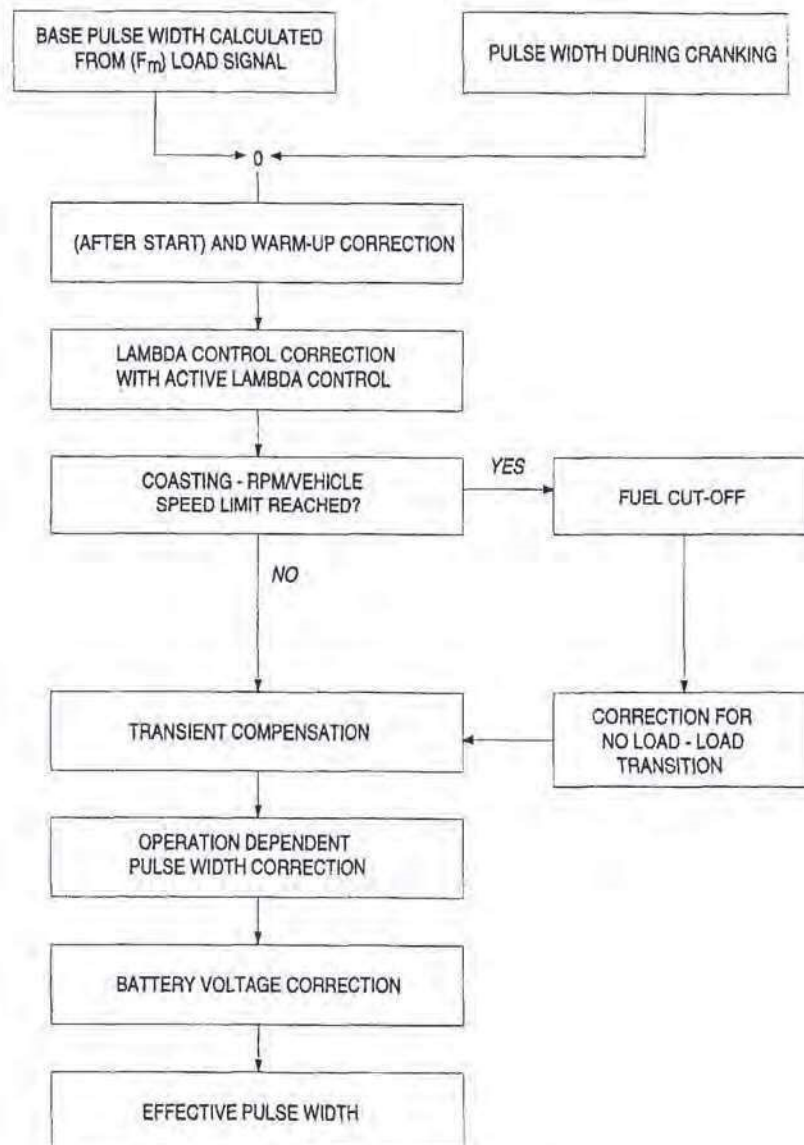


FIGURE 12.2 Determination of effective injector pulse width.

Injection Strategies. There are three commonly used fuel injection strategies for multi-point fuel metering systems: simultaneous injection, group injection, and sequential injection. Figure 12.3 is a diagram of the different strategies. Some engines use simultaneous injection during crank and switch over to sequential after the engine is running. This allows for shorter starting times since no synchronization with the camshaft is necessary before fuel injection begins. A description of each strategy follows.

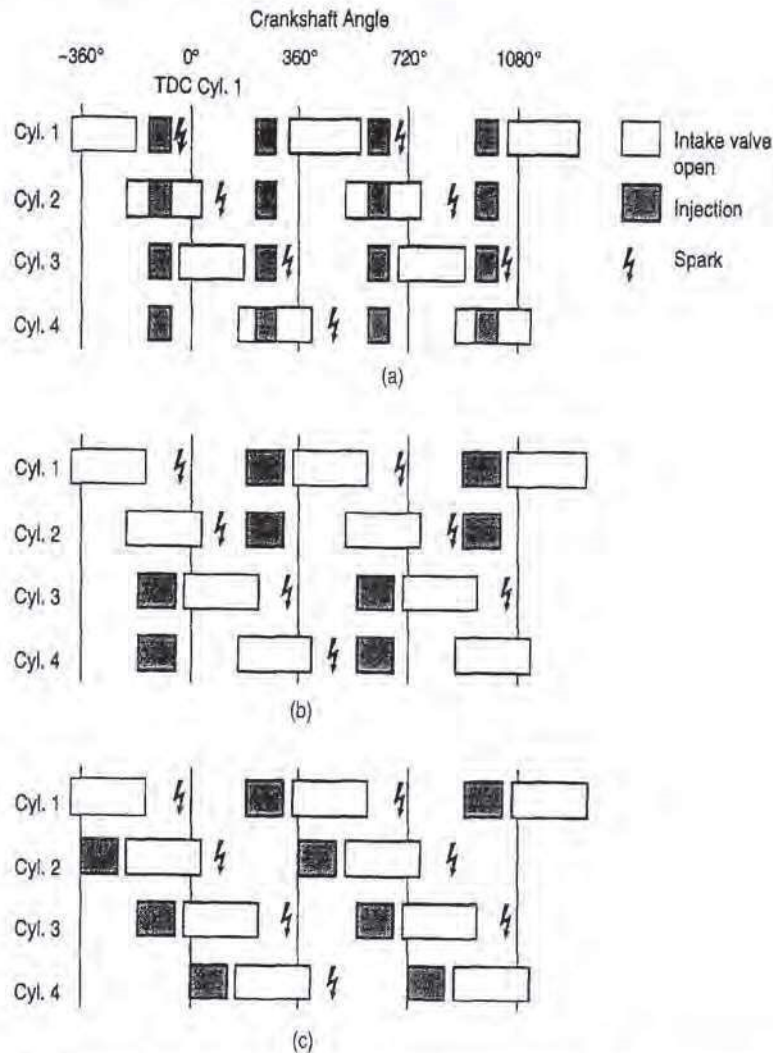


FIGURE 12.3 Fuel injection strategies: (a) simultaneous injection, (b) group injection, and (c) sequential injection.

Simultaneous injection. Injection of fuel occurs at the same time for all cylinders every revolution of the crankshaft. Therefore, fuel is injected twice within each four-stroke cycle. The injection timing is fixed with respect to crank/camshaft position.

Group injection. The injectors are divided into two groups that are controlled separately. Each group injects once per four-stroke cycle. The offset between the groups is one crankshaft revolution. This arrangement allows for injection timing selection that eliminates spraying fuel into an open intake valve.

Sequential injection. Each injector is controlled separately. Injection timing, both with reference to crank/camshaft position and pulse width, can be optimized for each individual cylinder.

Lambda Control. A subsystem of the fuel control system is lambda closed-loop control. Lambda (λ) is defined as the excess-air factor that indicates the deviation of the actual air/fuel ratio from the theoretically required ratio:

$$\lambda = \frac{\text{quantity of air supplied}}{\text{theoretical requirement (14.7 for gasoline)}}$$

The lambda sensor, or exhaust gas oxygen sensor, is installed in the engine exhaust stream upstream of the catalytic converter. The sensor responds to the oxygen content of the exhaust gas. The signal from the lambda sensor serves as feedback to the fuel control system. This provides the fine-tuning needed to remain within the limited catalytic converter window for optimal catalyst performance. (See Sec. 12.1.1 for more discussion on the catalytic converter window.) For a lean mixture ($\lambda > 1$), sensor voltage is approximately 100 mV. For a rich mixture ($\lambda < 1$), the sensor voltage is approximately 800 mV. At roughly $\lambda = 1$ (a stoichiometric mixture), the sensor switches rapidly between the two voltages. The input from the lambda sensor is used to modify the base pulse width to achieve $\lambda = 1$.

Lambda closed-loop control requires an operationally ready lambda sensor, typically one which has reached an operating temperature threshold. Sensor output is monitored by the ECU to determine when the sensor is supplying usable information. An active sensor signal, along with other requirements, such as engine temperature, must be achieved before lambda closed-loop control will be activated.

Under steady state conditions, the lambda control system oscillates between rich and lean around the lambda window. As the lambda sensor switches, the injector pulse width is adjusted by the amount determined by a control factor until the lambda sensor switches again to the opposite condition. The control factor can be defined as the allowable increase or decrease in the commanded fuel injector pulse width. The frequency of oscillation is determined by the gas transport time and the magnitude of the control factor. The gas transport time is defined as the time from air/fuel mixture formation to lambda sensor measurement.

Under transient conditions, the gas transport time results in a delay before the lambda sensor can indicate that the operating conditions have changed. Using only the lambda sensor for closed-loop fuel control would result in poor driveability and exhaust emissions because of this delay. Therefore, the engine control unit uses an anticipatory control strategy that uses engine load and RPM to determine the approximate fuel requirement. The engine load information is provided by the manifold pressure sensor for speed density systems and by the air meter for air flow and air mass measurement systems and by the throttle valve position sensor. The engine control unit contains data tables for combinations of load and RPM. This allows for rapid response to changes in operating conditions. The lambda sensor still provides the feedback correction for each load/RPM point. The data used for these data tables are largely developed from system modeling and engine development testing.

Due to production variations in engines, variations in fuel and changes due to wear and aging, the control system must be able to adapt to function properly for every engine over the engine's life. Therefore, the electronic control unit has a feature for adapting changes in the fuel required for the load/RPM points. At each load/RPM point, the lambda sensor continuously provides information that allows the system to adjust the fuel to the commanded A/F ratio. The corrected information is stored in RAM (random access memory) so that the next time the engine reaches that operating point (load/RPM), the anticipatory value will require less correction. These values remain stored in the electronic control unit even after the engine is shut off. Only if power to the electronic control unit is disrupted (i.e., due to a dead battery), will the correction be lost. In that case, the electronic control unit will revert back to the original production values that are written in ROM (read-only memory).

Lambda sensors do not switch symmetrically from lean to rich and rich to lean. Because of this, the control strategy is modified to account for the asymmetry. This can be accomplished either by delaying the modification by the control factor after the sensor switches or by using control factors of different magnitudes for rich-to-lean and lean-to-rich switching.

Ignition Timing Control. The goal of the engine control system for ignition timing is to provide spark advance which optimizes engine torque, exhaust emissions, fuel economy, and driveability, and which minimizes engine knock. Data tables with the base ignition timing, depending on engine load and RPM, are stored in ROM in the electronic control unit. The values in these tables are optimized for fuel economy, exhaust emissions, and engine torque. They are developed through engine experimentation, usually with an engine dynamometer. Corrections to the base timing values are needed for temperature effects, EGR, hot restart, barometric pressure, and engine knock. In addition, some systems use ignition timing to vary the engine torque for improvement in automatic transmission shift quality or for idle speed control. Figure 12.4 is a flowchart of a typical ignition timing calculation method.

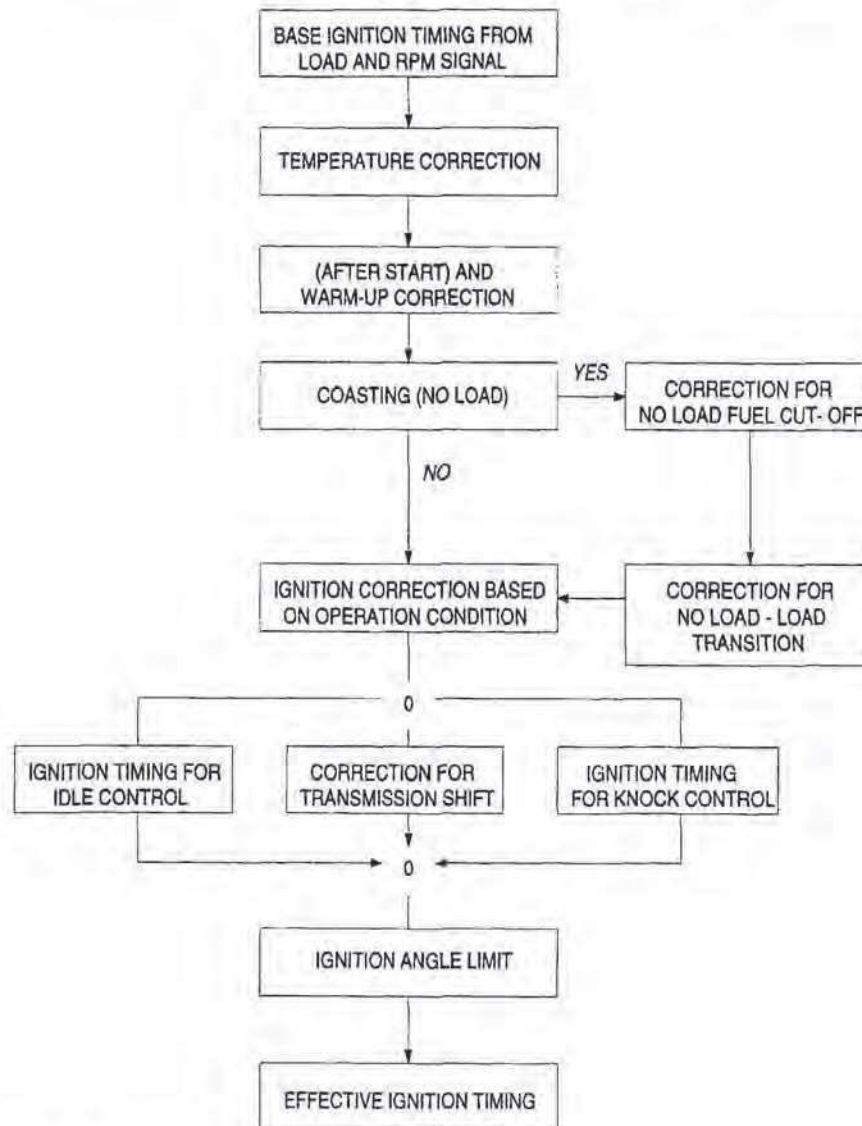


FIGURE 12.4 Determination of effective ignition timing.

Dwell Angle Control. The dwell angle performance map stored in the electronic control unit controls the charging time of the ignition coil, depending on RPM and battery voltage. The dwell angle is controlled so that the desired primary current is reached at the end of the charging time just prior to the ignition point. This assures the necessary primary current, even with quick transients in RPM. A limit on the charge time in the upper RPM ranges allows for the necessary spark duration.

Knock Control. The ignition timing for optimization of torque, fuel economy, and exhaust emissions is in close proximity to the ignition timing that results in engine knock. Engine knock occurs when the ignition timing is advanced too far for the engine operating conditions and causes uncontrolled combustion that can lead to engine damage, depending on the severity and frequency. If a factor of safety was used when developing the base timing map for all conditions that contribute to knock, such as fuel quality and variations in compression ratio, the ignition timing would be significantly retarded from the optimum level, resulting in a significant loss in torque and fuel economy. To avoid this, a knock sensor (one or more) is installed on the engine to detect knocking (see Chap. 8). Knock sensors are usually acceleration sensors that provide an electric signal to the electronic control unit. From this signal, the engine control unit algorithm determines which cylinder or cylinders are knocking. Ignition timing is modified (retarded) for those cylinders until the knock is no longer detected. The ignition timing is then advanced again until knock is detected. (See Fig. 12.5.) Information on the amount of spark retard required to eliminate the knock for each cylinder under each load/RPM condition is saved in the electronic control unit RAM. This allows for quick access to the appropriate "learned" ignition timing for each condition. With this control system, the base timing can be more advanced for improved fuel economy and torque.

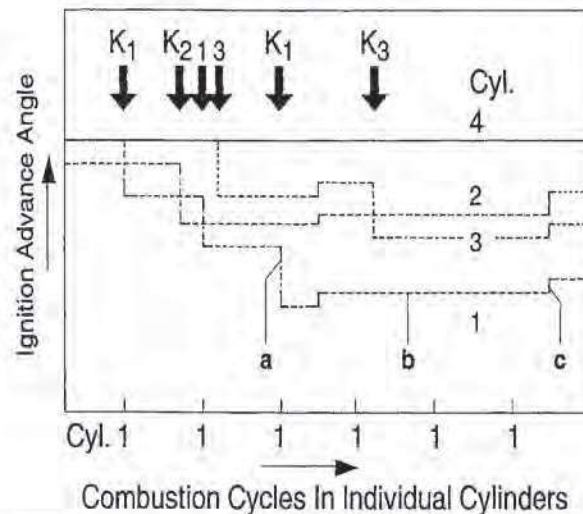


FIGURE 12.5 Knock control. Control algorithm for ignition adjustments for a four-cylinder engine. $K_{1...3}$ (knock in cylinders 1...3), cylinder number four (no knock), (a) (ignition retard), (b) (delay before return to original point), (c) (spark advance).

Evaporative Emissions Control. Hydrocarbon (HC) emissions in the form of fuel vapors escaping from the vehicle, primarily from the fuel tank, are closely regulated by federal statutes. There are two principal causes of fuel vapor in the fuel tank: increasing ambient temperature and return of unused hot fuel from the engine. In order to control the release of these emissions to the atmosphere, the evaporative emissions control system was developed.

Evaporative Emissions Control System. A vapor ventilation line exits the fuel tank and enters the fuel vapor canister. The canister consists of an active charcoal element which absorbs the vapor and allows only air to escape to the atmosphere. Only a certain volume of fuel vapor can be contained by the canister. The vapors in the canister must therefore be purged from and burned by the engine so that the canister can continue to store vapors as they are generated. To accomplish this, another line leads from the charcoal canister to the intake manifold. Included in this line is the canister purge solenoid valve. Figure 12.6 shows a layout of a typical evaporative emissions control system.

During engine operation, vacuum in the intake manifold causes flow through the charcoal canister because the canister vent opening, at the charcoal filter end, is at atmospheric pres-

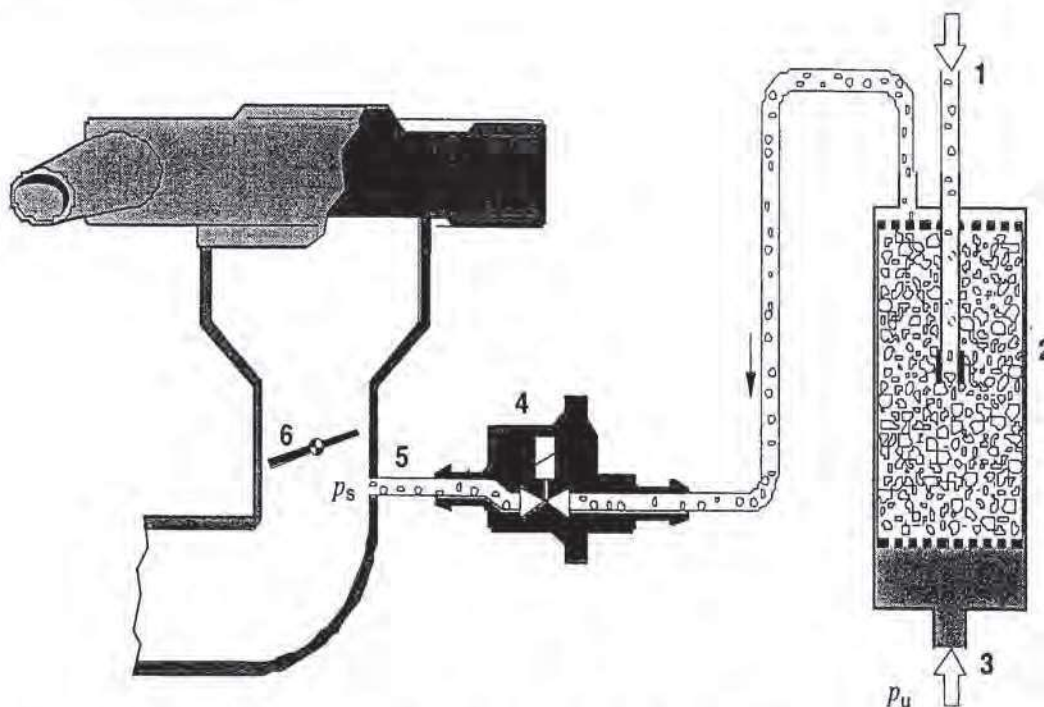


FIGURE 12.6 Evaporative emission control system: fuel vapor from fuel tank (1), charcoal canister (2), ambient air (3), canister purge control valve (4), purge line to intake manifold (5), throttle valve (6), p_s is intake manifold vacuum, and p_u is atmospheric pressure.

sure. The canister purge valve meters the amount of flow from the canister. The amount of fuel vapor in the canister and, therefore, contained in the flow stream, is not known. Therefore, it is critical that the lambda control system is operating and adjusting the fuel requirement as the vapors are being purged. Purge vapors could otherwise result in up to a 30 percent increase in air/fuel mixture richness in the engine.

Purge Valve Control. Control of the purge valve must allow for two criteria:

- There must be enough vapor flow so that the charcoal canister does not become saturated and leak fuel vapors to the atmosphere
- Purge flow must generally occur under lambda closed-loop control so that the effect of the purge vapors on A/F ratio can be detected and the fuel metering corrected

When the electronic control unit commands the purge valve to meter vapor from the canister, it requests a duty cycle (ratio of ON time to total ON and OFF time). This allows the amount of vapor flow to be regulated depending on the engine operating conditions. When lambda control is not operating, only low duty cycles and, therefore, small amounts of purge vapors, are allowed. Under deceleration fuel cutoff, the purge valve is closed entirely to minimize the possibility of unburned HCs in the exhaust.

Turbocharger Boost Pressure Control. The exhaust turbocharger consists of a compressor and an exhaust turbine arranged on a common shaft. Energy from the exhaust gas is converted to rotational energy by the exhaust turbine, which then drives the compressor. The compressed air leaves the compressor and passes through the air cooler (optional), throttle valve, intake manifold, and into the cylinders. In order to achieve near-constant air charge pressure over a wide RPM range, the turbocharger uses a circuit that allows for the bypass of the exhaust gases away from the exhaust turbine. The valve that regulates the bypass opens at a specified air charge pressure and is known as the wastegate.

Engines that have turbochargers benefit significantly from electronic boost pressure control. If only a pneumatic-mechanical wastegate is used, only one boost pressure point for the entire operating range is used to divert the exhaust gas. This creates a compromise for part-load conditions, which results in increased exhaust backpressure, more turbocharger work, more residual exhaust gas in the cylinders, and higher-charge air temperatures.

By controlling the wastegate with a pulse-width modulated solenoid valve, different wastegate opening pressures can be specified, depending on the engine operating conditions (Fig. 12.7). Therefore, only the level of air charge pressure required is developed. The electronic control unit uses information on engine load from either manifold pressure or the air meter and RPM and throttle position. From this information, a data table is referenced and the proper boost pressure (actually a duty cycle of the control valve) is determined. On systems using manifold pressure sensors, a closed-loop control system can be developed to compare the specified value with the measured value.

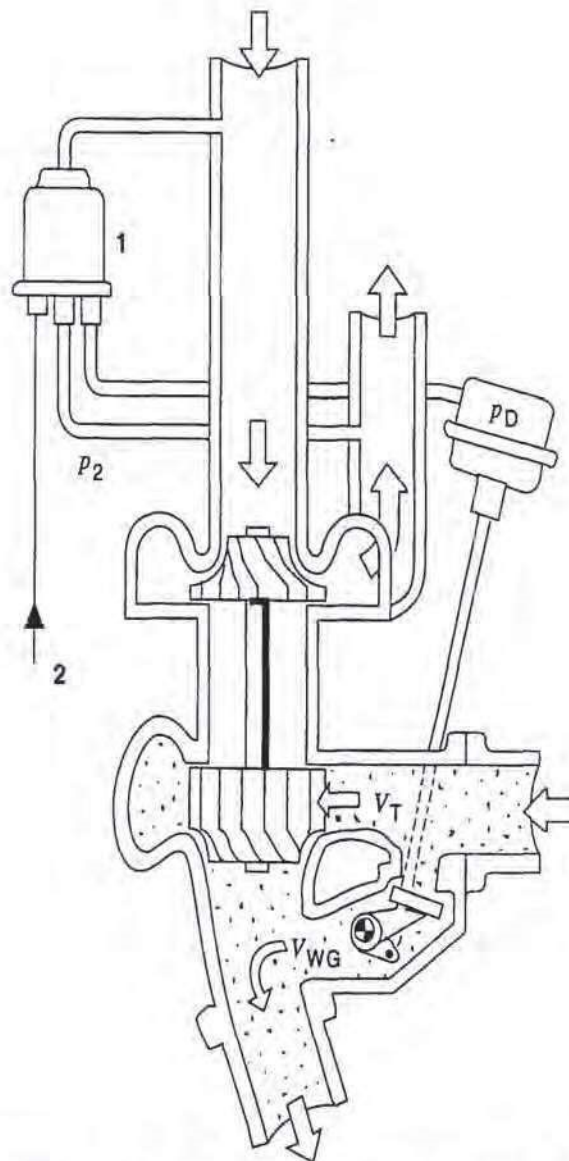


FIGURE 12.7 Electronic turbocharger boost control: solenoid valve (1), control signal from ECU (2), boost pressure (p_D), volume flow through turbine (V_T), volume flow through wastegate (V_{WG}).

The boost pressure control system is usually used in combination with knock control for turbocharged engines. When the ignition timing is retarded due to knock, an increase in already high exhaust temperatures for turbocharged engines occurs. To counteract the temperature increase, the boost pressure is reduced when the ignition timing is retarded past a predetermined threshold.

Engine/Vehicle Speed Control. Using the inputs of engine RPM and vehicle speed to the electronic control unit, thresholds can be established for limiting these variables with fuel cut-off. When the maximum speed is achieved, the fuel injectors are shut off. When the speed decreases below the threshold, fuel injection resumes.

EGR Control. By mixing a portion of the exhaust gas with the fresh intake air/fuel mixture, oxides of nitrogen (NO_x) can be reduced by lowering the peak combustion temperatures. However, the addition of exhaust gas can degrade driveability by causing combustion instability, especially at idle and low speeds and with a cold engine. The ECU references an engine RPM/load table of optimal EGR valve openings. The data table is developed on the engine dynamometer by analyzing the exhaust emissions. With increasing EGR, a point is reached where hydrocarbon (HC) emissions begin to increase. The optimal percent of EGR is just prior to that point.

The electronic control unit regulates a pneumatic- or solenoid-type valve to meter a certain quantity of exhaust gas back to the intake manifold. Typically, an engine coolant temperature threshold is also required before EGR is activated to avoid poor driveability. Under acceleration and at idle, EGR is deactivated.

Camshaft Control. There are two types of camshaft controls: phasing (i.e., overlap or intake/exhaust valve opening point) and valve lift and opening duration.

Camshaft Phasing Control. Valve overlap is a function of the rotation of the intake camshaft with respect to the exhaust camshaft. Overlap can be controlled by an electrohydraulic actuator. At idle and at high RPM, it is desirable to have the intake valves open and close later, which reduces the overlap. For idle, this reduces the residual exhaust gases that return with the fresh charge air and improves idle stability. At high RPM, late closing of the intake valve provides the best condition for maximum cylinder filling and, therefore, maximum output. For partial loads, a large valve overlap, where the intake opens early, is desirable. This allows for an increase in residual exhaust gas for improved exhaust emissions (Fig. 12.8).

Valve Lift and Opening Duration Control. Control of the valve lift and opening duration is accomplished by switching between two camshaft profiles. An initial cam specifies the optimal lift and duration for the low to middle RPM range. A second cam profile controls a higher valve lift and duration for high-RPM operation. By monitoring engine load and RPM, the ECU actuates the electrohydraulic device that switches from one cam profile to the other (Fig. 12.9).

Variable Intake Manifold Control. The goal of the engine design is to achieve the highest possible torque at low engine RPM as well as high output at high engine RPM. The torque curve of an engine is proportional to the air charge at any given engine speed. Therefore, a primary influence on the torque is the intake manifold geometric design. The simplest type of air charging uses the dynamics of the drawn-in air. The standard intake manifolds for multipoint engines consist of several intake runners and collectors converging at the throttle valve.

In general, short intake runners result in a high output at high RPM with a simultaneous loss of torque at low RPM. Long intake runners have the opposite effect. Due to intake valve and piston dynamics, pressure waves occur that oscillate within the intake manifold. Proper selection of runner lengths and collector sizes can result in the pressure waves arriving at the intake valves just before they are closing. This has a supercharging effect. The limitation of this method is that, for a given intake manifold configuration, the tuning peak can only occur at one operating point.

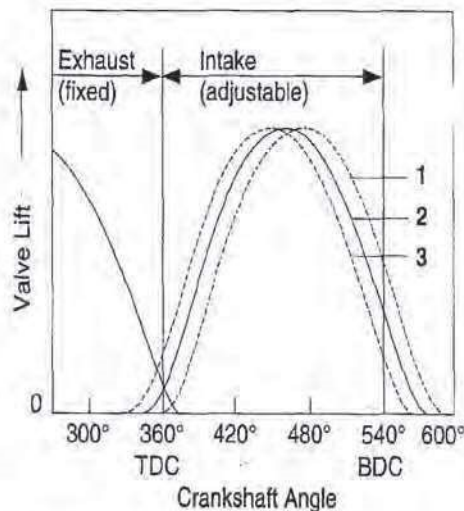


FIGURE 12.8 Adjustment angle for intake camshaft: retard (1), standard (2), advance (3).

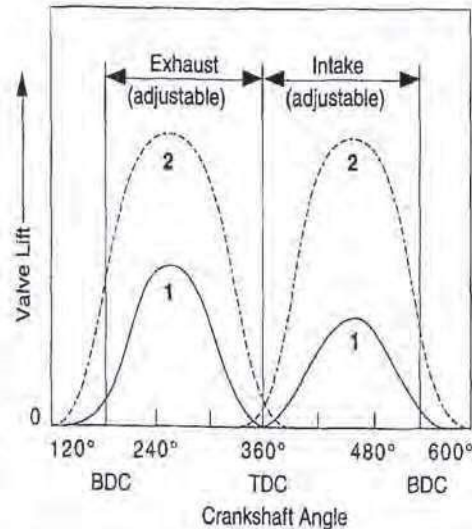


FIGURE 12.9 Selective camshaft lobe actuation: base cam lobe (1), auxiliary cam lobe (2).

Variable Intake Systems. To optimize the benefits of intake manifold charging, several systems have been developed that allow for changes in runner length and collector volume, depending on engine operating conditions. This allows for tuning peaks at more than one operating point. One method developed uses electronically controlled valves to close off areas of the intake manifold (Fig. 12.10). Inputs of engine load, RPM, and throttle angle determine the position of the valves.

12.2.2 Engine Control Modes

Engine Crank and Start. During engine cranking, the goal is to get the engine started with the minimal amount of delay. To accomplish this, fuel must be delivered that meets the requirements for starting for any combination of engine coolant and ambient temperatures. For a cold engine, an increase in the commanded air/fuel ratio is required due to poor fuel vaporization and "wall wetting," which decreases the amount of usable fuel. Wall wetting is the condensation of some of the vaporized fuel on the cold metal surfaces in the intake port and combustion chamber. It is critical that the fuel does not wet the spark plugs, which can reduce the effectiveness of the spark plug and prevent the plug from firing. Should plug wetting occur, it may be impossible to start the engine.

Fuel Requirement. Within the ECU ROM there are specific data tables to establish cold-start fuel based on engine coolant temperature. For two reasons, the lambda sensor output cannot be used during crank: the lambda sensor is below its minimum operating temperature and the air/fuel ratio required is outside the lambda sensor control window.

Many starting sequences use a front-loading strategy for fueling whereby the quantity of fuel is reduced after a speed threshold (RPM) is achieved, after a certain number of revolutions or at a defined time after the initial crank. Some systems also switch over from simultaneous injection to sequential injection after a speed threshold is achieved. For cold temperature starting, the fuel mixture may remain richer than $\lambda = 1$ after starting, due to the continuing poor mixture formation in the cold induction system.

Ignition Timing Requirement. Ignition timing is controlled by the ECU during crank and is determined by engine coolant temperature and cranking speed. For a cold engine with low cranking speeds, ideal timing is near TDC. For higher cranking speeds, a slightly more advanced timing is optimal. Timing advance must be limited during cranking to avoid igniting

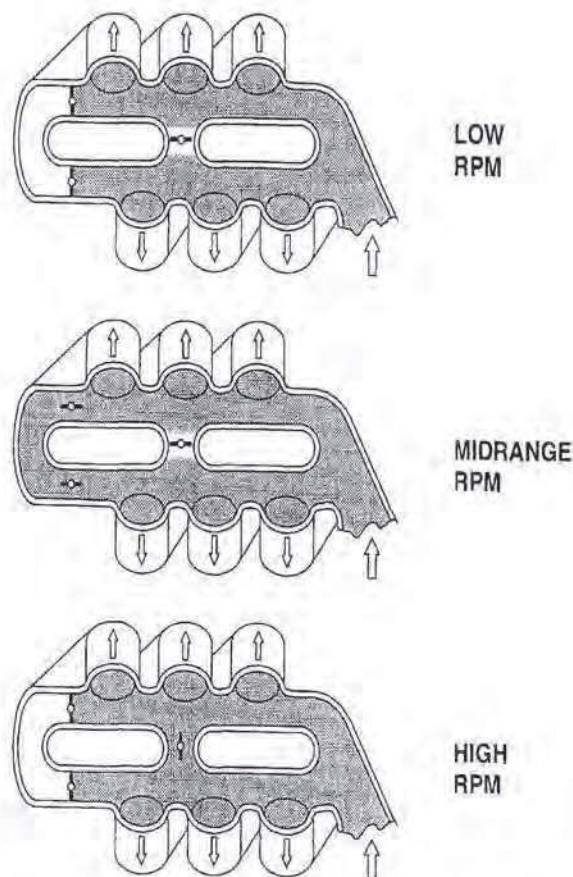


FIGURE 12.10 Variable configuration intake manifold.

the air/fuel mixture before the crankshaft reaches top dead center (TDC). A damaging torque reversal could occur that would damage the starter. After the engine starts, ignition timing is advanced to improve cold engine running as well as to reduce the need for fuel enrichment.

Engine Warm-Up. During the warm-up phase, there are three conflicting objectives: keep the engine operating smoothly (i.e., no stalls or driveability problems), increase exhaust temperature to quickly achieve operational temperature for catalyst (light-off) and lambda sensor so that closed-loop fuel control can begin operating, and keep exhaust emissions and fuel consumption to a minimum. The best method for achieving these objectives is very dependent on the specific engine application.

If the engine is still cold, fuel enrichment will be required to keep the engine running smoothly due, again, to poor fuel vaporization and wall wetting effects. The amount of enrichment is dependent on engine temperature and is a correction factor to the injector pulse width. This enrichment, combined with secondary air injection, also helps achieve the desired increase in catalyst temperature. To provide secondary air injection, an external air pump delivers fresh air downstream of the exhaust valves for a short time after start. The excess air causes oxidation (burning) of the excess HC and CO from the rich mixture in the exhaust manifold, which rapidly increases the temperature of the catalytic converter. The oxidation also removes harmful pollutants from the exhaust stream.

It is possible to increase the exhaust temperature by increasing the idle speed during warm-up. The increased idle speed may also be combined with a slightly retarded ignition timing, which increases temperatures in the exhaust, thereby promoting rapid warm-up of the catalyst.

Transient Compensation. During transitions such as acceleration or deceleration, the objective of the engine control system is to provide a smooth transition from one engine operating condition to another (i.e., no hesitations, stalls, bumps, or other objectionable driveability concerns), and keep exhaust emissions and fuel consumption to a minimum.

Acceleration Enrichment. When an increase in engine load and throttle angle occurs, a corresponding increase in fuel mixture richness is required to compensate for the increased wall wetting. The sudden increase in air results in a lean mixture that must be corrected swiftly to obtain good transitional response. The rate of change of engine load and throttle angle are used to determine the quantity of fuel during acceleration enrichment. The amount of fuel must be enough to provide the desired performance, but not so much as to degrade exhaust emissions and fuel economy.

During acceleration enrichment, the ignition timing is set for maximum torque without knocking. Additionally, when a large change in engine load occurs, some systems delay the ignition timing advance briefly to prevent engine knock, which may arise from a momentary lean mixture or from transient ignition timing errors.

Deceleration Enleanment. During deceleration modes, such as coasting or braking, there is no torque requirement. Therefore, the fuel may be shut off until either an increase in throttle angle is detected or the engine speed falls to a speed slightly above the idle RPM. Fuel shut-off or cutoff can decrease exhaust emissions by eliminating unburned HC and CO and may also improve fuel consumption. Fuel cutoff is also used to protect the catalytic converter from extreme high temperatures during extended overrun conditions. During transition to fuel cutoff, the ignition timing is retarded from its current setting to reduce engine torque and to assist in engine braking. The fuel is then shut off. During the transition, the throttle bypass valve or the main throttle valve may remain open for a short period to allow fresh air to oxidize the remaining unburned HC and CO to further reduce exhaust emissions. During development of the fuel cutoff strategy, the advantage of reduced emission effects and catalyst temperature control must be balanced against driveability requirements. The use of fuel cutoff may change the perceived amount of engine braking felt by the driver. In addition, care must be taken to avoid a "bump" feel when entering the fuel cutoff mode, due to the change in torque.

Full Load. Under steady state full-load conditions, such as for climbing a grade, it is desirable to control the air/fuel mixture and ignition timing to obtain maximum power and to also limit engine and exhaust temperatures. The best engine torque is typically delivered at about $\lambda = 0.9$ to 0.95. When the ECU determines the engine is operating at full load via the throttle valve sensor (at WOT), the commanded air/fuel mixture, if required, can be enriched. The lambda sensor signal cannot be used to provide correction to the air/fuel mixture because the rich operating point lies outside the lambda control window.

The ignition timing at full load is set to achieve the maximum torque without knocking. This initial value is determined through engine dynamometer testing. With a knock control system (see Sec. 12.2.1), the ignition timing is modified (retarded) when engine knock occurs. The modification required to eliminate the knock may be saved in the ECU so that the next time that engine RPM/load point occurs, less knocking will occur and less correction will be required.

Idle Speed Control. The objectives of the engine control system during idle are:

- Provide a balance between the engine torque produced and the changing engine loads, thus achieving a consistent idle speed even with various load changes due to accessories (i.e., air conditioning, power steering, and electrical loads) being turned on and off and during engagement of the automatic transmission. In addition, the idle speed control must be able to compensate for long-term changes in engine load, such as the reduction in engine friction that occurs with engine break-in.
- Provide the lowest idle speed that allows smooth running to achieve the lowest exhaust emissions and fuel consumption (up to 30 percent of a vehicle's fuel consumption in city driving occurs during idling).

To control the idle speed, the ECU uses inputs from the throttle position sensor, air conditioning, automatic transmission, power steering, charging system, engine RPM, and vehicle speed. There are currently two strategies used to control idle speed: air control and ignition control.

Air Control. The amount of air entering the intake manifold is controlled either by a bypass valve or by an actuator acting directly on the throttle valve. The bypass valve uses, for example, an electronically controlled motor controlled by the ECU that opens or closes a fixed amount. For large throttle valves, it may be desirable to use a bypass valve because a small change in throttle angle may result in a large change in air flow and, therefore, idle speed may be difficult to control. Using engine RPM feedback input, the ECU adjusts the air flow to increase or decrease the idle speed. A disadvantage to air control is that the response to load changes is relatively slow. To overcome this, air control is often combined with ignition timing control to provide acceptable idle speed control. The fuel quantity required at idle is determined by engine load and RPM. During closed-loop operation, this value is optimized by the lambda sensor closed-loop control.

Ignition Timing Control. Engine torque may be increased or decreased by advancing or retarding the ignition timing within an established window. This principle can be employed to help control idle speed. Ignition timing control is particularly desirable for responding to idle load changes because engine torque output changes more rapidly in response to a change in ignition timing than to a change in air valve position. Using the same inputs as for air control, the ECU adjusts the spark advance to either raise or lower the idle speed.

Anticipating Accessory Loads. Specific electric inputs to the ECU, such as a pressure switch located in the power steering system, are used to anticipate accessory loads so that the idle control system can compensate more quickly. This "feed forward" strategy allows better idle control than a strictly feedback system which does not respond until the idle speed begins to fall. When an accessory can be controlled by the ECU, further improvement in idle speed control is obtained. By delaying the load briefly after it is requested, the compensation sequence can begin before the load is actually applied. Such a load delay strategy is effective for controlling air conditioning compressor loads, for example. In this case, when the air conditioner is requested, the ECU begins to increase the idle speed first and then activates the A/C compressor.

12.2.3 Engine Control Diagnostics

The purpose of system diagnostics is to provide a warning to the driver when the control system determines that a malfunction of a component or system has occurred and to assist the service technician in identifying and correcting the failure (see Chap. 22). In many cases, to the driver, the engine may appear to be operating correctly, but excessive amounts of pollutants may be emitted. The ECU determines that a malfunction has occurred when a sensor signal received during normal engine operation or during a system test indicates there is a problem. For critical operations such as fuel metering and ignition control, if a required sensor input is faulty, a substitute value may be used by the ECU so that the engine will continue to operate, but likely not at optimal performance. It is also possible to apply an emergency measure if the failure of a component may result in engine or emission system damage. For example, if repeated misfires are detected in one cylinder, perhaps due to an ignition failure, the fuel injector feeding that cylinder can be shut off to avoid damage to the catalytic converter. When a failure occurs, the malfunction indicator light (MIL), visible to the driver, is illuminated. Information on the failure is stored in the ECU. A service technician can retrieve the information on the failure from the ECU and correct the problem.

Air Mass Sensor. For air mass measurement systems, the pulse width of the fuel injectors is calculated in the ECU from the air mass sensor input. As a comparison, the pulse width is also calculated from the throttle valve sensor and the engine RPM. If the pulse width values devi-

ate by a predetermined amount, the discrepancy is stored in the ECU. Then, while the vehicle is being driven, plausibility tests determine which input is incorrect. When this has been determined, the appropriate failure code is saved in the ECU.

Misfire Detection. Misfiring is the lack of combustion in the cylinder. Misfiring can be caused by several factors including fouled or worn spark plugs, poor fuel metering, or faulty electrical connections. Even a small number of misfires may result in excessive exhaust emissions due to the unburned mixture. Increased misfire rates can damage the catalytic converter.

To determine if the engine is experiencing a misfire, the crankshaft speed fluctuation is monitored. If a misfire occurs, no torque is created during the power stroke of the cylinder(s) that is misfiring. A small decrease in the rotational speed of the crankshaft occurs. Because the change in speed is very small, highly accurate sensing of the crankshaft speed is required. In addition, a fairly complicated calculation process is required in order to distinguish misfiring from other influences on crankshaft speed. As was mentioned previously, if a cylinder repeatedly misfires, it is possible to shut off the fuel to that cylinder to prevent damage to the catalytic converter.

Catalytic Converter Monitoring. During the useful life of a catalytic converter, its efficiency decreases. If subjected to engine misfire, the decrease in efficiency occurs more rapidly. A loss in efficiency results in an increase in exhaust pollutants. For this reason, the catalytic converter is monitored. A properly operating catalytic converter transforms O_2 , HC, CO, and NO_x into H_2O , CO_2 , and N_2 . The incoming air/fuel ratio oscillates from rich to lean due to the lambda closed-loop control strategy discussed in Sec. 12.2.1. Only a properly functioning catalytic converter is able to dampen these oscillations by storing and converting the incoming components. As the catalyst ages, this storage effect is diminished. To monitor the catalytic converter, an additional lambda sensor is installed downstream of the catalyst. The ECU compares the signal of the lambda sensor upstream with the lambda sensor downstream and determines if the catalytic converter is operating properly. If not, the ECU illuminates the malfunction indicator light (MIL) and stores a failure code.

Lambda Sensor Monitoring. To minimize exhaust emissions, the engine must operate within the catalytic converter window for air/fuel ratio (see Sec. 12.1.1 for a detailed description of the catalytic converter window). Output from the lambda sensor serves as feedback to the ECU to control the fuel within that window. When a lambda sensor is exposed to high heat for a long period of time, it may respond more slowly to changes in the air/fuel mixture. This can cause a deviation in the air/fuel mixture from the window, which would affect the exhaust emissions.

If the upstream lambda sensor operation is determined to be too slow, which can be detected by the system operation frequency, the ECU illuminates the malfunction indicator light (MIL) and a failure code is stored. Additionally, the ECU compares the output signal of the additional lambda sensor downstream of the catalytic converter with the lambda sensor signal upstream. With this, the ECU is able to detect deviations of the average value in air/fuel ratio.

For heated lambda sensors, the electric current and voltage of the heater circuit is monitored. To accomplish this, the heater is directly controlled by the ECU, not through a relay.

Fuel System Monitoring. To provide the correct air/fuel ratio, the ECU uses a preset data map with the optimal fuel required for each load and RPM point. The lambda closed-loop control system (see Sec. 12.2.1) provides feedback to the ECU on the necessary correction to the preset data points. The corrected information is stored in the ECU's RAM so that the next time that operating point is reached, less correction of the air/fuel ratio will be required. If the ECU correction passes a predetermined threshold, it is an indication that some component in the fuel supply system is outside of its operating range. Some examples are defective pressure regulator, defective manifold pressure sensor, intake system leakage, or exhaust system leakage. When the ECU determines a problem exists, the MIL is illuminated and a code is stored in the ECU.

Exhaust Gas Recirculation (EGR) Monitoring. There are currently two methods used to monitor EGR operation. One method confirms that hot exhaust gases are returning to the intake manifold during EGR operation by use of a temperature sensor in the intake manifold. The second method requires the EGR valve to be fully opened during coast operation, where high intake manifold vacuum occurs. The exhaust gas flowing into the manifold causes a measurable increase in pressure. Thus, if a measured increase in pressure does not occur, the EGR system is not operating.

Evaporative Emissions Control System (EECS) Monitoring. In general, a valve will be installed at the atmospheric side of the purge canister. During idle, this valve would close and the purge valve would open. Intake manifold vacuum would occur in the entire EECS. A pressure sensor in the fuel tank would provide a pressure profile during this test to the ECU, which would then determine if a leak existed in the system.

12.2.4 Fuel Delivery Systems

Overview. Fuel management in the spark ignition engine consists of metering the fuel, formation of the air/fuel mixture, transportation of the air/fuel mixture, and distribution of the air/fuel mixture. The driver operates the throttle valve, which determines the quantity of air inducted by the engine. The fuel delivery system must provide the proper quantity of fuel to create a combustible mixture in the engine cylinder. In general, two fuel delivery system configurations exist: *single-point* and *multipoint* (Fig. 12.11).

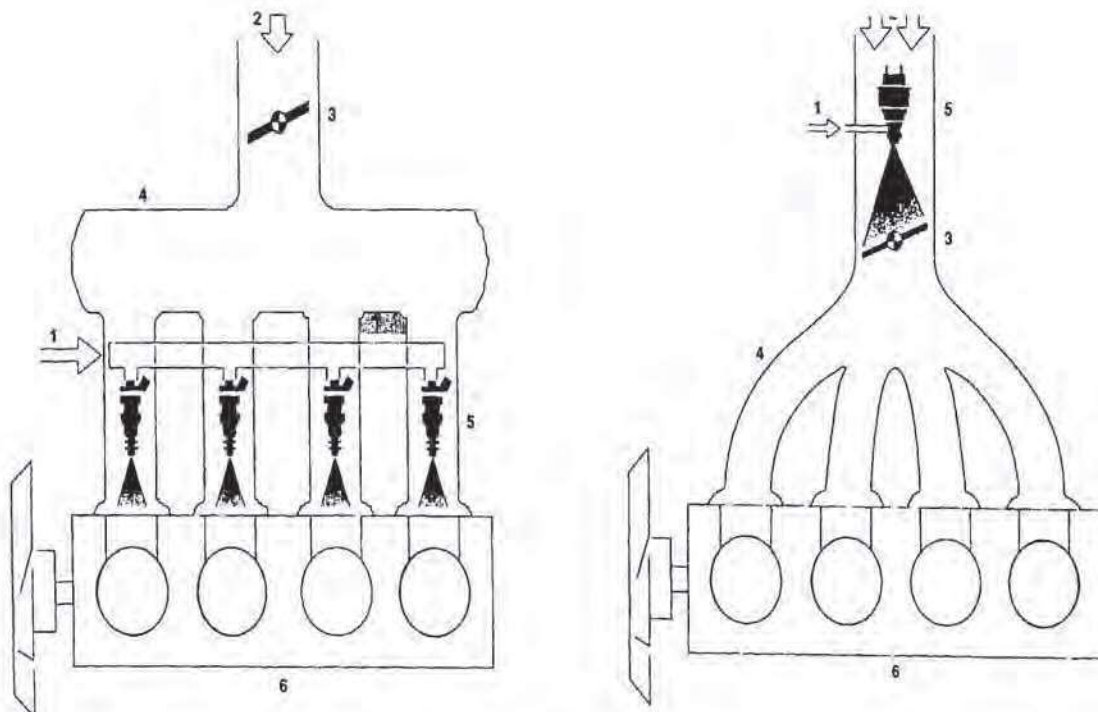


FIGURE 12.11 Air-fuel mixture preparation: right, single-point fuel injection; left, multipoint fuel injection with fuel (1), air (2), throttle valve (3), intake manifold (4), injector(s) (5), and engine (6).

For single-point systems such as carburetors or single-point fuel injection, the fuel is metered in the vicinity of the throttle valve. Mixture formation occurs in the intake manifold. Some of the fuel droplets evaporate to form fuel vapor (desirable) while others condense to form a film on the intake manifold walls (undesirable). Mixture transport and distribution is a function of intake manifold design. Uniform distribution under all operating conditions is difficult to achieve in a single-point system.

For multipoint systems, the fuel is injected near the intake valve. Mixture formation is supplemented by the evaporation of the fuel on the back of the hot intake valve. Mixture transport and distribution occurs only in the vicinity of the intake valve. The influence of the intake manifold design on uniform mixture distribution is minimized. Since mixture transport and distribution is not an issue, the intake manifold design can be optimized for air flow.

Single-Point Injection Systems A single-point injection system uses one or, in some cases, two electronic fuel injectors to inject fuel into the intake air stream. The main component is the fuel injection unit which is located upstream of the intake manifold.

Component Description. An electric fuel pump provides fuel at a medium pressure (typically 0.7 to 1.0 bar) to the electronic fuel injection unit (Fig. 12.12). The fuel injection unit houses the solenoid-operated fuel injector, which is located in the intake air flow above the throttle valve. This allows for homogeneous mixture formation and distribution. The injector spray pattern is designed to allow fuel to pass between the throttle valve and the throttle bore. To prevent vapor lock of the injector, fuel flows through the injector at all times. Fuel not used by the engine is returned to the fuel tank. The injector is activated in relation to the speed of the engine, typically once per ignition event. The length of the pulse width determines the quantity of fuel provided.

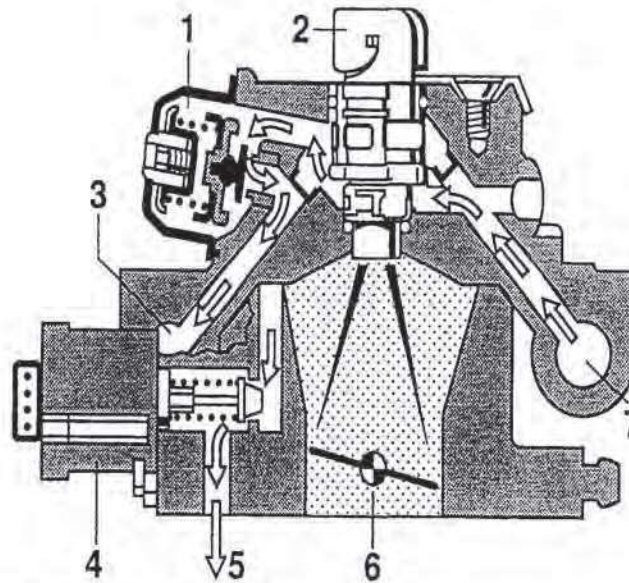


FIGURE 12.12 Single-point injection unit: pressure regulator (1), injector (2), fuel return (3), stepper motor for idle speed control (4), to intake manifold (5), throttle valve (6), and fuel inlet (7).

The electronic injection unit also houses the throttle position sensor and, in some cases, an inlet air temperature sensor which provides operating condition information to the ECU. The throttle valve actuator and fuel pressure regulator are also mounted on the injection unit. In addition, some units contain an air bypass valve for idle speed control. Engine temperature, battery voltage, and engine speed via the ignition system are all inputs to the ECU. The single-

point injection system also uses lambda closed-loop fuel control to optimize fuel metering within the lambda control window (see Sec. 12.2.1).

Adaptation to Operating Conditions. For cold-start and engine warm-up, the ECU uses engine temperature information to determine the correct amount of fuel and commands the fuel injector via a pulse width. Due to wall wetting and poor fuel vaporization when the engine is cold, an increase in mixture richness is required. As the engine warms up to operating temperature, the commanded pulse width is reduced.

During an acceleration transition, the ECU adds a correction factor (an increase) to the commanded injector pulse width. The sudden increase in air results in a lean mixture which must be corrected swiftly to obtain good transitional response. During a deceleration transition, the fuel can be shut off by simply not providing a pulse width signal to the injector to minimize exhaust emissions and fuel consumption.

During full-load operation, the air/fuel mixture can be enriched ($\lambda < 1$) to deliver maximum torque. The ECU determines full-load operation by the throttle position sensor (at or near wide-open throttle) and adds a correction to the injector pulse width to achieve the desired air/fuel mixture richness.

The single-point system can control the idle speed by ECU control of either a throttle valve actuator or a bypass valve. Idle speed is a function of engine operating temperature, whether the transmission is in drive, and what accessories are in use. Fuel metering at idle is determined by engine RPM and load as well as lambda closed-loop control.

Multipoint Fuel Injection Systems. A multipoint fuel injection system supplies fuel to each cylinder individually via a mechanical or solenoid-operated fuel injector located just upstream of the intake valve. Advantages of this system type compared to SPI systems are numerous:

- *Increased fuel economy.* On an SPI engine, due to the intake manifold configuration, mixture formation will differ at each cylinder. To provide adequate fuel for the leanest cylinder, too much fuel must be metered overall. In addition, during engine load changes, a film of fuel is deposited on the intake manifold walls. This leads to further variations in mixture from cylinder to cylinder. Multipoint injection provides the same quantity of fuel to each cylinder.
- *Higher power output.* With the fuel being injected near the intake valve, the rest of the intake manifold can be optimized for maximum air flow. The result is increased torque.
- *Improved throttle response.* Because the fuel is injected onto the intake valves, responses to increases in throttle position are swift. With an SPI system, the increased fuel required must travel the length of the intake manifold before entering the cylinder.
- *Lower exhaust emissions.* As was discussed for fuel economy, mixture variation in an SPI system creates increased exhaust emissions. Metering of the fuel at the intake valve decreases this variation. In addition, the system transport time is reduced, increasing the frequency at which the lambda closed-loop control system can switch air/fuel ratio. Catalytic converter efficiency is increased.

Although there are numerous advantages of the MPI systems over the SPI systems, there is still one important advantage the SPI systems have over the MPI systems. In general, SPI systems have better fuel preparation, similar to a carburetor.

Mechanically Controlled Continuous Injection System. This type of system meters the fuel as a function of the intake air quantity and injects it continuously onto the intake valves. This is accomplished by measuring the air flow as it passes through the air flow meter by means of deflection of a meter plate. The fuel is supplied through a fuel accumulator to the fuel distributor by an electric fuel pump. A primary-pressure regulator in the fuel distributor maintains constant fuel pressure. The fuel distributor, through its interface with the air flow meter and warm-up regulator, meters fuel to the continuously flowing fuel injectors.

Component Description

Mixture control unit. The mixture control unit houses the air flow meter and the fuel distributor. In the air flow meter, the measurement of the intake air serves as the basis for

determining the amount of fuel to be metered to the injectors. The air flow meter is located upstream of the throttle valve so that it measures all the air entering the engine. It consists of an air funnel, in which a sensor plate is free to pivot. Intake air flowing through the funnel causes a deflection of the sensor plate. The sensor plate is mechanically linked to a control plunger and movement of the plate results in movement of the control plunger. The control plunger movement determines the amount of fuel to be injected.

In the fuel distributor, the control plunger moves up and down in a cylindrically shaped device (barrel) with rectangular openings (metering slit), one for each engine cylinder. Increased air flow causes the control plunger to move upward, uncovering a larger area of the metering slit and increasing the fuel metered. Downstream of each metering slit is a differential-pressure valve that maintains a constant pressure drop across the metering slits at different flow rates. Due to the constant pressure, the fuel flow through the slits is directly proportional to the position of the control plunger (Fig. 12.13).

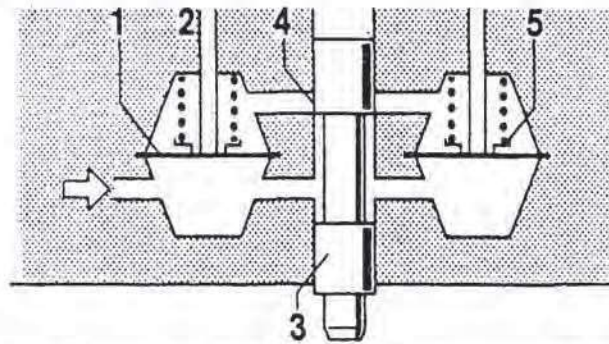


FIGURE 12.13 Fuel distributor for mixture control unit: diaphragm (1), to injector (2), control plunger (3), metering slot (4), differential pressure regulator (5).

Warm-up regulator. The warm-up regulator is used to richen the fuel mixture under cold engine conditions. It consists of a diaphragm valve and an electrically heated bimetallic spring. Under cold conditions, the warm-up regulator lowers the control pressure on the control plunger. The control pressure acts on the opposite end of the plunger from the air flow meter plate. A lower control pressure results in a lower force required to move the meter plate. Therefore, the same air flow causes the meter plate and control plunger to move a greater distance and additional fuel is metered to the injectors.

Fuel injectors. The injectors open at a pressure of approximately 3.6 bar. Atomization of the fuel occurs through oscillation (audible chatter) of the valve needle caused by the fuel flowing through it. The injectors remain open as long as fuel is provided above the opening pressure. Fuel is injected continuously into the intake port. When the intake valve opens, the mixture is drawn into the cylinder.

Auxiliary air valve. The auxiliary air valve provides additional air to the engine by bypassing the throttle valve during cold engine operation. This creates an increase in the idle speed needed during cold operation.

Thermo-time switch. The thermo-time switch controls the cold start valve as a function of time and engine temperature. Fuel enters the intake manifold from the cold start valve and further enriches the mixture to improve cold-starting at low ambient temperatures. When the engine is warm, the contacts in the thermo-time switch open and the cold-start valve is not used in starting the engine.

Lambda sensor. With the addition of a lambda sensor in the exhaust stream, a frequency valve, a modified fuel distributor, and an electronic control unit, the mechanically controlled fuel system can operate under lambda closed-loop control. The lambda sensor sig-

nal is read by the ECU. The ECU outputs electric pulses to an electromagnetic (frequency) valve. The frequency valve modulates the pressure to the lower chambers of the differential-pressure valves in the fuel distributor. This results in a modification of the pressure drop across the metering slits, effectively increasing or decreasing the amount of fuel injected. Figure 12.14 is a schematic of a typical mechanically controlled continuous injection system.

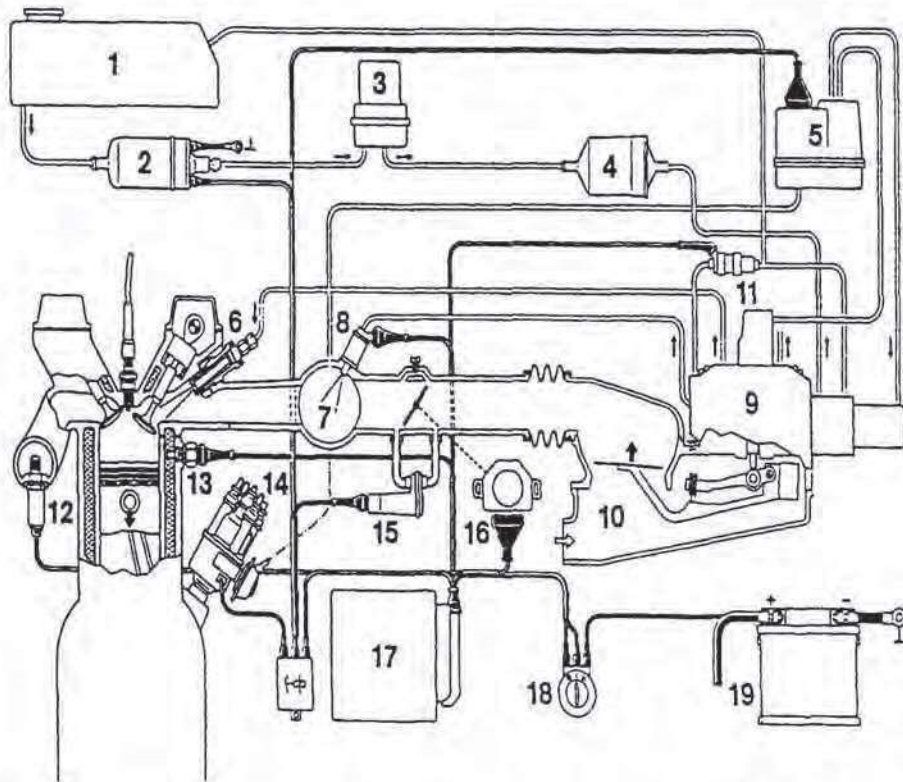


FIGURE 12.14 Schematic of mechanically controlled continuous injection system: fuel tank (1), electric fuel pump (2), fuel accumulator (3), fuel filter (4), warm-up regulator (5), injector (6), intake plenum (7), cold-start valve (8), fuel distributor (9), air flow sensor (10), electrohydraulic pressure actuator (11), lambda sensor (12), thermo-time switch (13), ignition distributor (14), auxiliary air valve (15), throttle switch (16), ECU (17), ignition switch (18), and battery (19).

Depending on the engine temperature, the cold-start valve injects extra fuel into the intake manifold for a limited period during cold start. The injection period is determined by a combination of time and temperature and is controlled by the thermo-time switch. As the engine temperature increases, this additional enrichment is no longer required and the thermo-time switch turns off the cold-start valve. For repeated start attempts or long cranking, the thermo-time switch turns off the cold-start injector after a given time. This minimizes engine flooding when engine start has not occurred.

As the engine continues to warm up, wall wetting and poor fuel vaporization still occur and mixture enrichment is still required until the engine reaches operating temperature. This enrichment is controlled as a function of temperature by the warm-up regulator. As the temperature increases, the warm-up regulator commands less and less additional fuel by increasing the control pressure.

For acceleration response, the air flow sensor "overswings" during quick throttle increases. This causes an additional quantity of fuel to be injected for acceleration enrichment. For full-load enrichment to achieve maximum power, a special warm-up regulator

that uses intake manifold pressure is required. At increased manifold pressures, i.e., during wide-open throttle, the warm-up regulator lowers the control pressure, which results in an increase in fuel delivery. Deceleration fuel shutoff is accomplished by diverting all intake air through an air bypass around the air flow sensor plate. With no air flow past the air flow sensor plate, the fuel pressure to the injectors is decreased below the opening pressure.

Idle speed for the cold-running engine is increased by the auxiliary air valve. The amount of additional air varies with engine temperature until the auxiliary air valve is closed and the idle speed is then controlled only by the air passing the throttle valve.

Electronically Controlled Continuous Injection. The basis of the electronically controlled continuous injection is still the mechanical hydraulic injection system discussed previously. This is supplemented by an electronic control unit (ECU) that allows for an increase in flexibility and the use of additional functions. This system incorporates additional sensors for detecting the engine temperature, the throttle valve position (load signal), and the air flow sensor plate deflection. This information is processed by the ECU, which then commands an electrohydraulic pressure actuator to adapt the injected fuel quantity for the present operating conditions.

In contrast to the mechanical system mentioned previously, the control pressure or counterpressure on the control plunger is not varied by a warm-up regulator. The control pressure remains constant and is the same as the primary pressure. The function of the warm-up regulator is now handled by the ECU and the electrohydraulic pressure actuator. Figure 12.15 is a schematic of a typical electronically controlled continuous injection system.

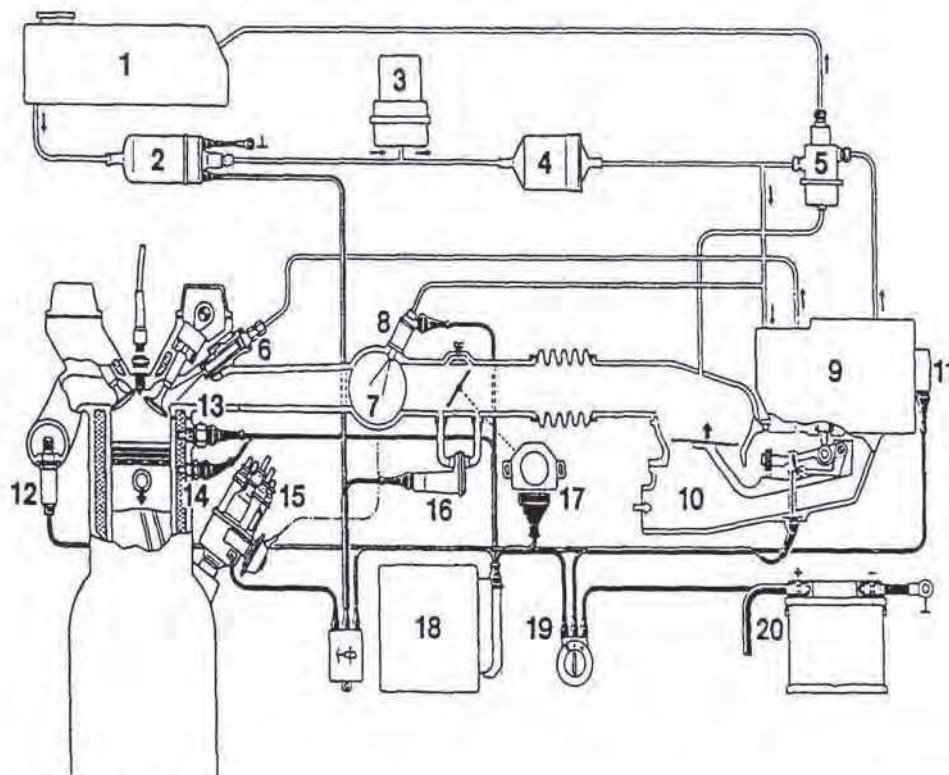


FIGURE 12.15 Schematic of an electronically controlled continuous fuel injection system: fuel tank (1), electric fuel pump (2), fuel accumulator (3), fuel filter (4), fuel pressure regulator (5), injector (6), intake plenum (7), cold-start valve (8), fuel distributor (9), air flow sensor (10), electrohydraulic pressure actuator (11), lambda sensor (12), thermo-time switch (13), coolant temperature sensor (14), ignition distributor (15), auxiliary air valve (16), throttle valve switch (17), ECU (18), ignition switch (19), and battery (20).

Component description—electrohydraulic pressure actuator. The main difference in the componentry between the purely mechanical system and the electronically controlled system is the addition of the electrohydraulic actuator and the elimination of the warm-up regulator. With the addition of the ECU control of fuel metering, the purely mechanical warm-up regulator is no longer required. Depending on the signal received from the ECU, the electrohydraulic pressure actuator varies the pressure in the lower chambers of the differential pressure valves. This changes the amount of fuel delivered to the injectors.

Lambda closed-loop control. As with the mechanical system, the lambda sensor signal is processed by the ECU to determine mixture composition. The difference is that the ECU now commands the electrohydraulic actuator to modify the fuel metered, as opposed to the separate frequency valve, which is no longer necessary.

Adaptation to operating conditions. Depending on the engine temperature, the cold-start valve injects extra fuel into the intake manifold for a limited period during cold-start. The quantity to be injected is controlled by the ECU and is a function of engine temperature (from the engine temperature sensor). The thermo-time switch controls how long the cold-start valve remains active, depending on engine temperature and time.

Acceleration enrichment is controlled by the ECU. Input from the air flow sensor plate position sensor provides the ECU with information on how quickly the engine load has increased. The ECU commands additional enrichment via the electrohydraulic pressure actuator. For full-load enrichment for maximum power, the ECU receives input from the throttle position sensor that the throttle is wide open. The ECU then commands additional enrichment via the electrohydraulic pressure actuator. Deceleration fuel shutoff is also controlled by the ECU when the throttle valve switch indicates the throttle is closed and the engine speed is above idle RPM. The ECU signals the electrohydraulic pressure actuator to interrupt fuel delivery to the injectors.

Idle speed control can be a closed-loop function with the addition of the idle actuator valve. This valve is ECU-controlled and the RPM signal from the ignition, combined with the engine temperature signal, is used to determine its position for the correct idle speed.

Pulsed Fuel Injection Systems. Pulsed fuel injection systems are a further enhancement of the continuous injection systems. Today, most continuous injection systems have been replaced with pulsed fuel injection systems. Instead of injecting fuel continuously and controlling the quantity of fuel by modifying the delivery volume flow rate, the fuel quantity is controlled by the open time of the solenoid-operated injectors. The injectors are controlled directly by the ECU. For most systems, the fuel pressure drop across the injector, from the fuel rail to the intake manifold, is kept constant by using intake manifold air pressure to compensate the fuel pressure regulator. This type of system allows for still greater precision of fuel control and is usually coupled with an equally precise ignition timing control system.

Component description. Several multipoint pulsed injection systems exist in various configurations. The components discussed here serve as a general outline of this system type. Figure 12.16 is a schematic of a typical pulsed fuel injection system.

- *Inlet air sensing.* The inlet air charge can be measured directly using either an air flow meter or a mass air flow meter. The air flow meter is a vane-type meter, which uses the force of the incoming air to move a flap through a defined angle. The angular movement is converted by a potentiometer to a voltage ratio. The air flow meter requires an air inlet temperature sensor to correct for air density changes. The air mass flow meter measures the air mass directly by hot-wire or hot-film element. As the inlet air flow passes the heated element, a bridge circuit keeps the element at a constant temperature above the inlet temperature. The heating current required by the bridge circuit to maintain the element at a constant temperature is measured and converted to an air density value.

The air charge can also be measured indirectly by measuring the inlet air temperature, intake manifold pressure, and engine RPM and then calculating the air charge (see Sec. 12.2.1 for further discussion on the calculation method which is called speed density air measurement).

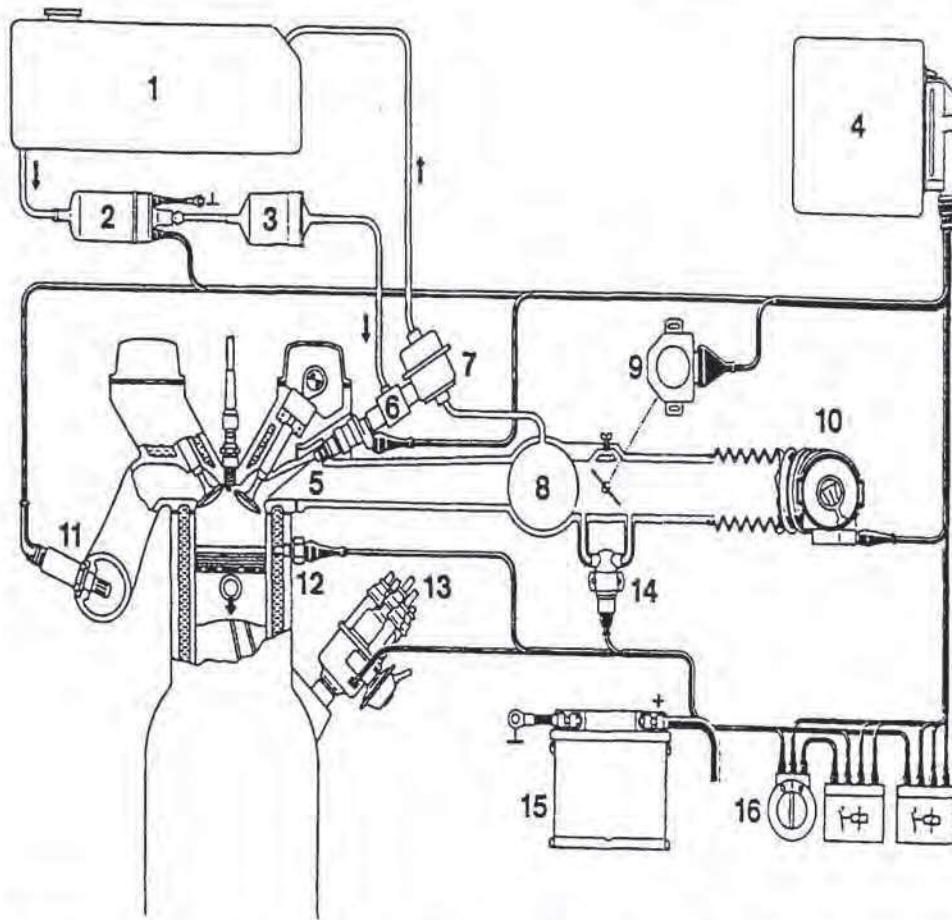


FIGURE 12.16 Schematic of a pulsed fuel injection system: fuel tank (1), electric fuel pump (2), fuel filter (3), ECU (4), injector (5), fuel distributor (6), fuel pressure regulator (7), intake plenum (8), throttle valve switch (9), hot-wire mass air flow sensor (10), lambda sensor (11), coolant temperature sensor (12), ignition distributor (13), idle speed actuator (14), battery (15), and ignition switch (16).

- **Fuel metering.** The fuel supply system includes an electric fuel pump, fuel filter, fuel rail, pressure regulator, and solenoid-operated injectors. The fuel pump provides more fuel than the maximum required by the engine. Fuel not used by the engine is returned to the fuel tank. The fuel rail supplies all injectors with an equal quantity of fuel and ensures the same fuel pressure at all injectors.

The pressure regulator keeps the pressure differential across the injectors constant. It contains a diaphragm that has intake manifold pressure on one side and fuel rail pressure on the other. Normally, it is mounted at the outlet end of the fuel rail. The diaphragm operates a valve which opens at a differential pressure between 2.0 and 3.5 bar and allows excess fuel to return to the fuel tank.

The fuel injectors are solenoid-operated valves that are opened and closed by means of electric pulses from the ECU. The injectors are mounted in the intake manifold and spray onto the back of the intake valves. In general, one injector is used for each cylinder.

In addition, some systems also use a separate cold-start injector mounted in the intake manifold just downstream of the throttle valve. This injector ensures good fuel vaporization during cold-start and supplies the additional enrichment needed to start the cold engine. Control of the cold-start valve is either by the ECU directly or in conjunction with a thermo-time switch.

- *Lambda closed-loop control.* The lambda sensor signal is processed by the ECU. The ECU determines the required injector pulse width to maintain the air/fuel ratio within the lambda control window (see Sec. 12.2.1 for further discussion on lambda closed-loop control).

Adaptation to operating conditions. For cranking, the fuel required is determined by a data table in the ECU with reference to engine temperature. The ECU then commands a pulse width for the fuel injectors. The air/fuel mixture is greatly enriched due to poor fuel vaporization and wall wetting, which reduces the amount of usable fuel. After start, the fuel mixture remains rich due to continuing poor air/fuel mixture formation. The amount of enrichment should be minimized to obtain good emission results. The target is to stay close to lambda (λ) = 1.

For acceleration enrichment, the throttle valve position sensor indicates that the throttle has moved rapidly. The ECU adds a correction factor to increase the pulse width so that a smooth transition occurs. For deceleration, the ECU uses input from the throttle position sensor and engine RPM to indicate that the throttle has closed and the engine speed is above the idle speed. Since no torque is required under this condition, the ECU provides no pulse width to the injectors and they are therefore closed. For full-load enrichment, when necessary, the ECU can provide an injector pulse width that would result in the engine achieving its maximum torque (roughly $\lambda = 0.9$). Fuel metering during idle is primarily controlled by lambda closed-loop control when the engine has reached operating temperature.

12.2.5 Ignition Systems

Overview. The purpose of the ignition system in the spark ignition engine is to initiate combustion of the air/fuel mixture by delivering a spark at precisely the right moment. The spark consists of an electrical arc generated across the electrodes of the spark plug. Two important factors for proper ignition are the energy of the spark and the point in the four-stroke cycle when the spark occurs (ignition timing).

Electrical Energy. The energy required for a spark to ignite an air/fuel mixture at stoichiometry depends on specific engine conditions. If there is insufficient energy available to ignite the air/fuel mixture, misfiring will occur. Misfiring will result in poor engine operation, high exhaust emissions, and possible catalytic converter damage. Therefore, the amount of ignition energy available must always exceed the amount necessary to ensure ignition even under adverse conditions.

Some of the conditions that affect ignitability of the air/fuel mixture are fuel atomization, access of the mixture to the spark, spark duration, and spark physical length. Fuel atomization is controlled by the fuel system and the engine design. Access to the spark depends on combustion chamber and spark plug design. Spark duration is a function of the ignition system. Spark physical length is determined by the spark plug dimensions (gap).

Ignition Timing. The ignition timing must be selected to meet the following objectives: maximize engine performance, limit fuel consumption, minimize engine knock, and minimize exhaust emissions. Unfortunately, all of these objectives cannot be achieved simultaneously under all operating conditions and compromises must be made.

It is desirable in the SI engine to have ignition of the combustible mixture occur prior to the piston reaching TDC on the compression stroke to achieve the best engine performance. The ignition spark must occur early enough to ensure that the peak cylinder combustion pressure occurs at the correct point after top dead center (ADC) under all operating conditions. Figure 12.17 is a graph of ignition angle vs. combustion pressure. The length of the combustion process from initial ignition to final combustion is approximately 2 ms. This combustion time remains relatively constant with respect to engine speed. Therefore, as the engine speed increases, the ignition spark must occur earlier in terms of crankshaft angle to ensure complete combustion.

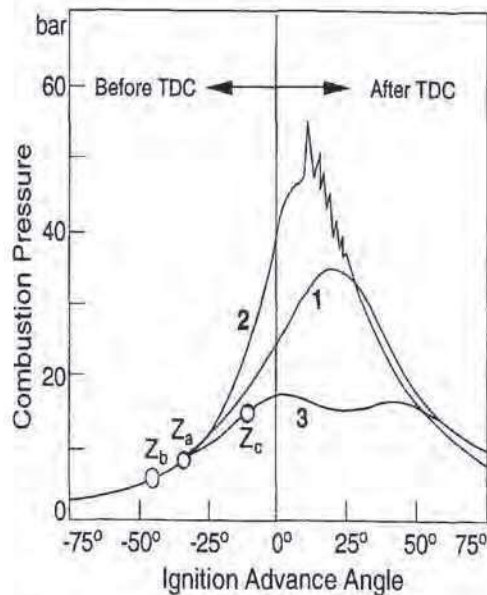


FIGURE 12.17 Combustion pressure curve for various ignition timing points: correct ignition advance Z_a (1), excessive ignition advance Z_b (2), and excessive ignition retard Z_c (3).

At low engine loads, the lower air charge and the residual gas content, due to valve overlap, serve to lengthen the time required for complete combustion. To compensate for this effect, the ignition timing is advanced at low loads to ensure that complete combustion occurs.

Ignition timing influences exhaust emissions and fuel consumption. With more advanced timing, the emission of unburned hydrocarbons (HC) and of oxides of nitrogen (NO_x) increases. Carbon monoxide (CO) emissions are not influenced greatly by ignition timing. To achieve improvements in fuel consumption, the air/fuel mixture must be lean. To ensure complete combustion for a lean mixture, the ignition timing must be advanced. However, as previously stated, advanced timing increases emissions of HC and NO_x .

Spark Ignition Systems. The general configuration of an ignition system consists of the following components: energy storage device, ignition timing mechanism, ignition triggering mechanism, spark distribution system, and spark plugs and high tension wires.

Inductive ignition systems use an ignition coil as the energy storage device. The coil also functions as a transformer, boosting the secondary ignition voltage. A typical turns-ratio of the primary to secondary winding is 1:100. Electrical energy is supplied to the coil's primary winding from the vehicle electrical system. Before the ignition point, the coil is charged during the dwell period to its interruption current. Open- or closed-loop dwell angle control ensures a sufficient interruption current even at high speeds. Sufficient ignition energy at the interruption current is ensured by an adequate coil design. At the ignition point, the primary current will be interrupted. The rapid change of the magnetic field induces the secondary voltage in the secondary winding. A distribution system assigns the high voltage to the corresponding spark plug. After exceeding the arcing over voltage at the spark plug, the coil will be discharged during the spark duration.

The ignition timing mechanism, ignition triggering mechanism, and the spark distribution system differ between ignition systems. Further discussion of these will occur within the discussion of each ignition system type.

The spark plugs provide the ignition energy via the high-tension wires to the air/fuel mixture in the cylinder to initiate combustion. The voltage required at the spark plug can be more

than 30 kV. Because the spark plug extends into the combustion chamber, it is exposed to extreme temperature and pressure conditions. Spark plug design and materials are chosen to ensure long-term operation under tough operating conditions.

A typical spark plug consists of a pair of electrodes, called a center and ground electrode, separated by a gap. The size of the gap is important and is specified for each plug type and engine. The center electrode is electrically connected to the top terminal of the plug which is attached to the high-tension wire. The electrical energy travels through the high-tension wire to the top terminal and down to the center electrode. The ground electrode is part of the threaded portion of the spark plug that is installed in the cylinder head. The ground electrode is at electrical ground potential because the negative terminal of the battery is also connected to the engine. The spark is produced when the high-voltage pulse travels to the center electrode and jumps the gap to the ground electrode.

Ignition System Types. Table 12.1 summarizes the various ignition systems used on SI engines.

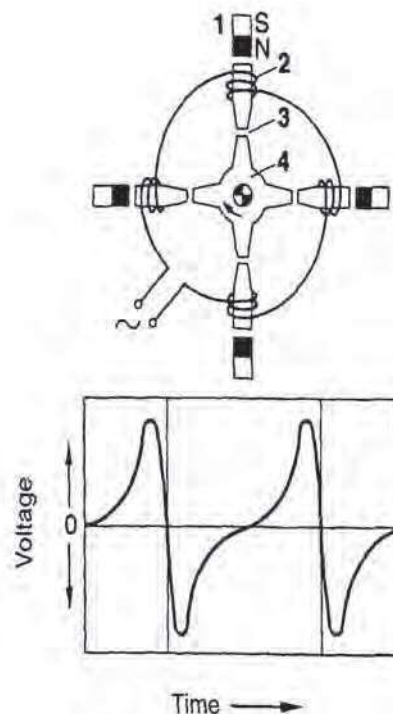


FIGURE 12.18 Induction-type pulse generator; permanent magnet (1), induction winding with core (2), variable air gap (3), trigger wheel (4).

Coil ignition. Breaker-triggered coil ignition systems have been replaced by breakerless transistorized ignition systems and are no longer installed as original equipment.

On breakerless transistorized ignition systems, the contact breaker's function is replaced by a magnetic pulse generator. The pulse generator is installed in the distributor and turns with the distributor shaft. There are commonly two types of pulse generators: induction-type and Hall-type. Induction-type pulse generators consist of a stator and a trigger wheel (Fig. 12.18). The stator consists of a permanent magnet, inductive winding, and core, and remains fixed. The trigger wheel teeth correspond to the number of cylinders, and the trigger wheel turns with the distributor shaft. The operating principle is that as the air gap changes between the stator and the rotor, the magnetic flux changes. The change in magnetic flux induces an ac voltage in the inductive winding. The frequency and magnitude of the alternating current increases with increasing engine speed. The electronic control unit or trigger box uses this information to trigger the ignition timing.

TABLE 12.1 Overview of Various Ignition Systems

Ignition function	Ignition designation				
	Coil system	Transistorized coil system	Capacitor discharge system	Electronic system with distributor	Electronic distributorless system
Ignition triggering	Mechanical	Electronic	Electronic	Electronic	Electronic
Ignition timing	Mechanical	Mechanical	Electronic	Electronic	Electronic
High-voltage generation	Inductive	Inductive	Capacitive	Inductive	Inductive
Spark distribution to appropriate cylinder	Mechanical	Mechanical	Mechanical	Mechanical	Electronic

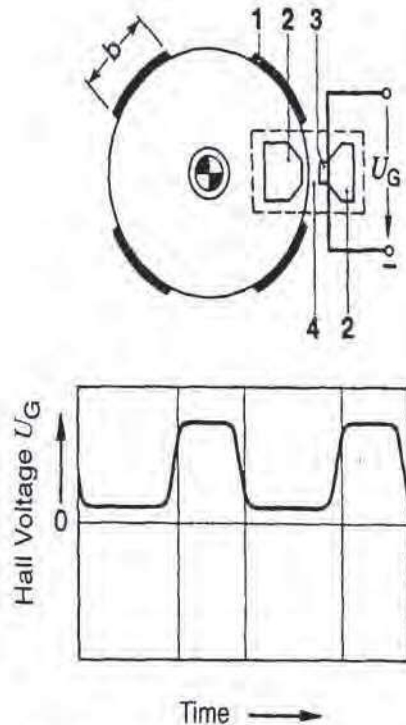


FIGURE 12.19 Hall-type pulse generator: vane (1), soft magnetic conductive elements (2), Hall IC (3), and air gap, U_G -Hall sensor voltage (4).

Hall-type pulse generators utilize the Hall effect (Fig. 12.19). As the distributor shaft turns, the vanes of the rotor move through the air gap of the magnetic barrier. When the vane is not in front of the Hall IC, the sensor is subjected to a magnetic field. The magnetic flux density is high and thus the voltage U_G is at a maximum. As soon as the rotor vane enters the air gap, the magnetic flux runs through the vane area and is largely prevented from reaching the Hall layer. The voltage U_G is at a minimum. The resulting pulses switch the primary current off and on.

The distributor discharges the ignition pulses to the spark plugs via the high-tension wires in a specific sequence. It also adjusts the ignition timing by means of spark advance mechanisms. The distributor rotor is turned by the engine at one-half the crankshaft speed. The electrical energy is fed to the center of the rotor. While the rotor turns, the rotor electrode aligns with the outer electrodes that are connected to the high-tension wires. One outer electrode and high-tension wire connection exists for each cylinder. When alignment occurs between the center and outer electrode, the spark is distributed to that particular cylinder.

The spark advance mechanisms advance the ignition timing by rotating the distributor plate relative to the distributor shaft. The centrifugal advance increases the spark advance with increasing engine speed. The vacuum advance, using intake manifold vacuum, increases the spark advance at low engine speeds.

Capacitor discharge ignition system. The capacitive discharge system differs from the coil-type ignition systems previously discussed. Ignition energy is stored in the electrical field of a capacitor. Capacitance and charge voltage of the capacitor determine the amount of energy that is stored. The ignition transformer converts the primary voltage discharged from the capacitor to the required high voltage.

Electronic ignition—with distributor. Electronic ignition calculates the ignition timing electronically (Fig. 12.20). This replaces the function of the centrifugal advance and vacuum advance in the distributor discussed on the previous coil ignition systems. Because the ignition timing is not limited by mechanical devices, the optimal timing can be chosen for each engine operating point. Figure 12.21 is a comparison of an ignition map from a mechanical advance system and a map of an electronically optimized system. Also, additional influences such as engine knock detection can be used to modify the ignition timing. The engine speed input and crankshaft position input can be obtained from a sensor mounted near the crankshaft. Precision is improved over using the distributor-mounted trigger. This input is provided to the electronic control unit (ECU) along with the engine temperature and engine load. The ECU references data tables to determine the optimal spark advance for each engine operating condition. Additional corrections to the spark timing, such as for EGR usage or knock sensor detection, are made in the ECU.

Electronic ignition—distributorless. On distributorless ignition systems, the high voltage distribution is accomplished by using either single or double spark ignition coils. Ignition timing is determined by the ECU, as discussed for electronic ignition with distributor. For the double spark ignition coils, one coil exists for two corresponding cylinders. Two high-tension

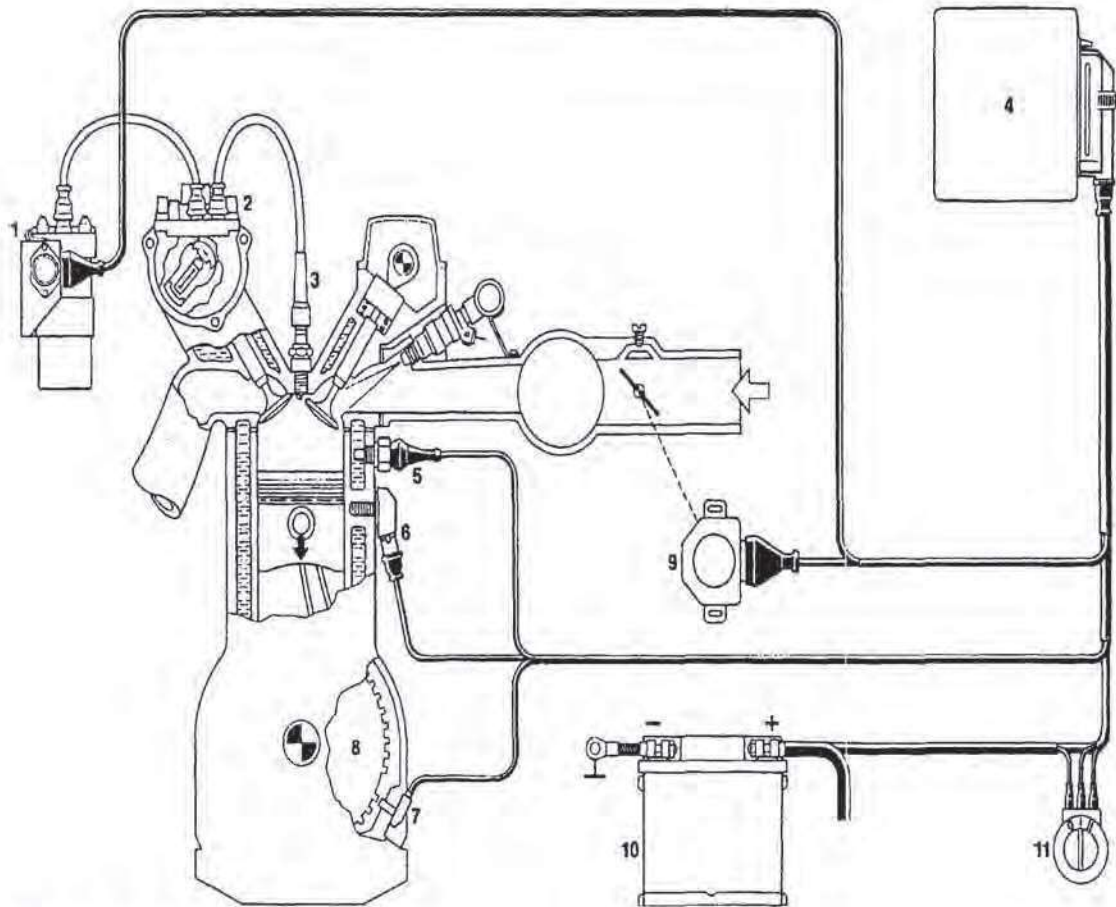


FIGURE 12.20 Schematic of an electronic ignition system with distributor: ignition coil (1), high-voltage distributor (2), spark plug (3), ECU (4), coolant temperature sensor (5), knock sensor (6), engine speed and crankshaft reference sensor (7), sensor wheel (8), throttle valve (9), battery (10), and ignition switch (11).

wires are routed from each coil to two cylinders, which are 360° out of phase. When the coil output stage is triggered via the ECU, a spark is delivered to both cylinders. One cylinder will be on the compression stroke, the other on the exhaust stroke. Because both cylinders are fired together, for a given crankshaft rotation, one will always be on the compression stroke and the other on the exhaust stroke. Therefore, there is no need to know which cylinder is compressing the ignitable mixture.

On single spark ignition coils, one coil exists for each cylinder. Each coil triggers only once during the four-stroke cycle. Because of this, it must be known which cylinder is on the compression stroke. Synchronization with the camshaft must occur. The information needed on camshaft position is supplied by a phase sensor mounted on the camshaft.

12.3 COMPRESSION IGNITION ENGINES

12.3.1 Engine Control Functions

Electronic engine controls are now being used on compression ignition (diesel) engines. These controls offer greater precision and control of fuel injection quantity and timing, engine speed, EGR, turbocharger boost pressure, and auxiliary starting devices. The following inputs

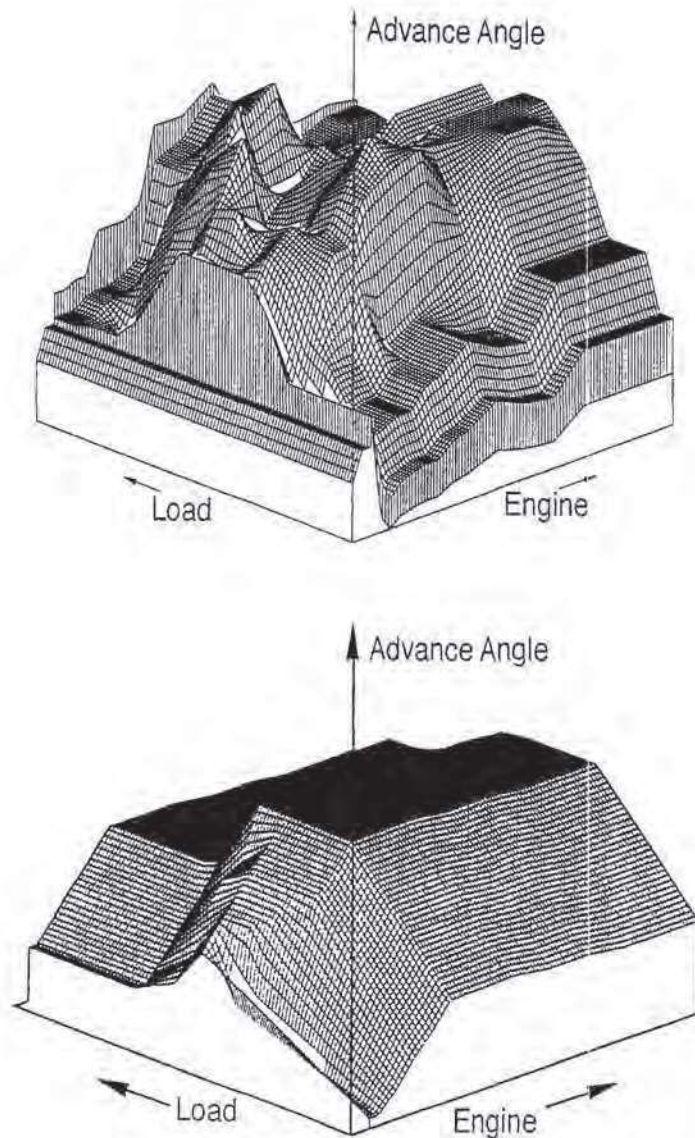


FIGURE 12.21 Ignition timing maps: electronically optimized (*above*) and mechanical advance system (*below*).

are used to provide the ECU with information on current engine operating conditions: engine speed; accelerator position; engine coolant, fuel, and inlet air temperatures; turbocharger boost pressure, vehicle speed, control rack, or control collar position (for control of fuel quantity); and atmospheric pressure. Figure 12.22 is a schematic of an electronic engine control system on an in-line diesel fuel injection pump application.

Fuel Quantity and Timing. The fuel quantity alone controls a compression ignition engine's speed and load. The intake air is not throttled as in a spark ignition engine. The quantity of fuel to be delivered is changed by increasing or decreasing the length of fuel delivery time per injection. On the injection pump, the delivery time is controlled by the position of the control rack on in-line pumps and the position of the control collar on distributor-type pumps. An ECU-controlled actuator is used to move the control rack or the collar to increase or decrease the fuel delivery time. The ECU determines the correct length of delivery time (expressed as a function of control rack or collar position) using performance maps based on engine speed and calculated fuel quantity. Corrections and/or limitations as functions of

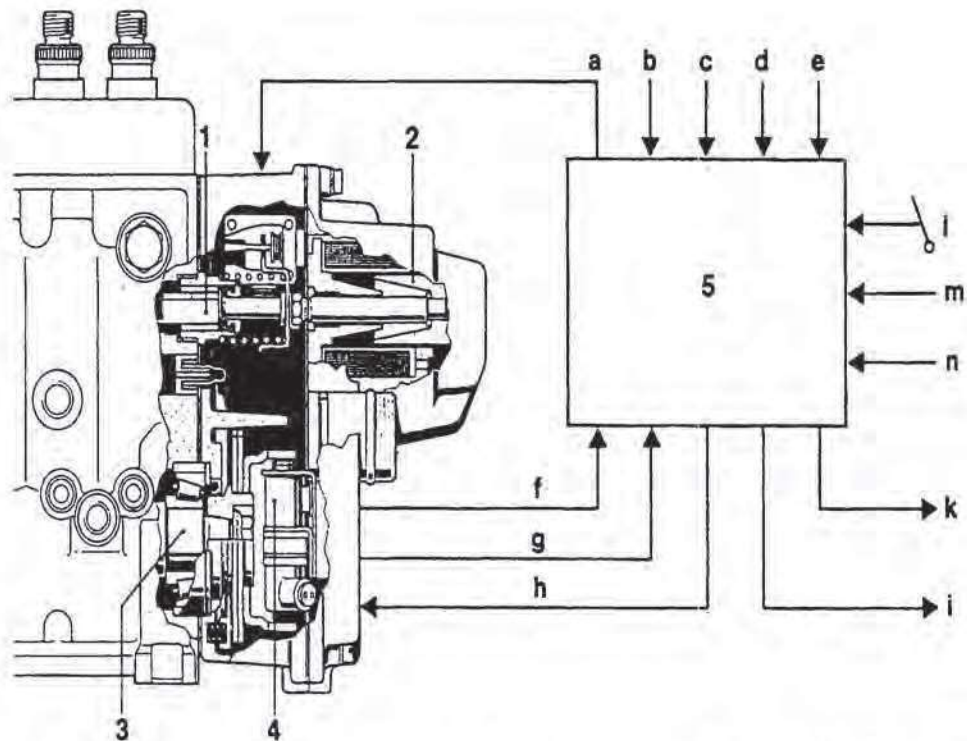


FIGURE 12.22 Electronic engine control system for an in-line injection pump: control rack (1), actuator (2), camshaft (3), engine speed sensor (4), ECU (5). Input/output: redundant fuel shutoff (a), boost pressure (b), vehicle speed (c), temperature—water, air, fuel (d), intervention in injection fuel quantity (e), speed (f), control rack position (g), solenoid position (h), fuel consumption and engine speed display (i), system diagnosis information (k), accelerator position (l), preset speed (m), and clutch, brakes, engine brake (n).

engine speed, temperature, and turbocharger boost pressure are used to modify the delivery time. In addition, the control rack or collar actuator contains a position sensor that provides feedback to the ECU on controller position. If the requested position differs from the commanded position, the ECU continues to move the controller via the actuator until the commanded and actual position are the same.

The start of injection time of the fuel at the cylinder is a function of the wave propagation speed (i.e., the speed of sound) of the fuel from the fuel injection pump to the injector. Because this time remains a constant, at increasing engine speed the delivery of fuel at the cylinder would be delayed with reference to crankshaft angle. Therefore, the timing at the injection pump must be advanced with increasing engine speed so that the start of injection occurs at the same crankshaft angle at higher engine speeds. Selection of injection timing has a large impact on exhaust emissions and engine noise. Delaying the start of injection reduces NO_x emissions, but excessive delay increases HCs in the exhaust. A 1° deviation in injection timing can increase NO_x emissions by 5 percent and HC emissions by as much as 15 percent. Therefore, precise control of the start of injection is essential.

Although many systems use mechanical devices to control injection timing, electronic control of injection timing is being used on some pump types. The advantage of electronic control is that a sophisticated timing data map can be used that provides the best injection timing for exhaust emissions under various operating conditions. On electronic control systems, the start of injection is monitored at the injector nozzle by a needle-motion sensor. The ECU uses this information to determine and control the injection timing. The timing is then modified by control of a pulse-width modulated solenoid valve. The valve varies the pressure exerted on the spring-loaded timing device plunger. The plunger rotates the pump's collar ring (for distributor type pumps) in the opposite direction of the pump's rotation which advances the timing.

Speed Control. As was mentioned previously, for a CI engine, fuel quantity alone controls the engine's speed and load. Therefore, presuming adequate injected fuel quantity, an unloaded CI engine can speed up out of control and destroy itself. Because of this, a governor is required to limit the engine's maximum speed. In addition, governors are also used for low idle and cruise control to maintain a constant engine or vehicle speed and meter the correct fuel for cold-starting. Fuel is also controlled as a function of speed and boost pressure to limit smoke levels, engine torque, and exhaust gas temperatures. On an electronically controlled CI engine, the governor's functions are controlled by the fuel delivery system described previously. Engine speed is provided by an RPM sensor that monitors the periods of angular segments between the reference marks on the engine's flywheel or in the in-line injection pump.

EGR Control. Rerouting of exhaust gases into the intake air stream is known as exhaust gas recirculation (EGR). EGR reduces the amount of oxygen in the fresh intake charge while increasing its specific heat. This lowers combustion temperatures and results in lower NO_x emissions. However, excessive amounts of EGR result in higher emissions of soot (particulates), CO, and HCs all due to insufficient air. Also, the introduction of EGR can have an adverse affect on driveability during cold-engine operation, full-load operation, and at idle. It is best, therefore, to control the EGR valve with the ECU. Both pneumatically controlled and solenoid-controlled EGR valves are in use. The ECU determines when and how much EGR will occur based on engine temperature and accelerator position.

Turbocharger Boost Pressure Control. Engines that have turbochargers benefit significantly from electronic boost pressure control. If only a pneumatic-mechanical wastegate is used, only one boost pressure point for the entire operating range is used to divert the exhaust gas away from the turbine side of the turbocharger. This creates a compromise for part-load conditions because all the exhaust gases must pass the turbine. The result is increased exhaust backpressure, more turbocharger work, more residual exhaust gas in the cylinders, and higher charge air temperatures.

By controlling the wastegate with a pulse-width-modulated solenoid valve, the wastegate can be opened at different pressures depending on the engine operating conditions. Therefore, only the level of air charge pressure required is developed. The electronic control unit uses information on engine speed and accelerator position to reference a data table and the proper boost pressure (actually, duty cycle of the control valve) is determined. On systems using intake manifold pressure sensors, a closed-loop control system can be developed to compare the specified value with the measured value.

Glow Plug Control. Electronic control of the glow plug duration can be handled by the ECU or a separate control unit. Input for determining glow time is from an engine coolant temperature sensor. At the end of the specified glow period, the controller turns out the start indicator light to signal the driver that the engine can be started. The glow plugs remain energized while the starter is engaged. An engine load monitor is used to switch off the glow process after start. To limit the loads on the battery and the glow plugs, a safety override is also used.

12.3.2 Fuel Delivery Systems

The diesel fuel delivery system comprises a low- and high-pressure side. On the low-pressure side is the fuel tank, fuel filter, fuel supply pump, overflow valve, and fuel supply lines. The high-pressure side is initiated in the plunger and barrel assembly and continues through the delivery valve, high-pressure injection lines, and injection nozzle.

The fuel injection pump must deliver fuel at a pressure between 350 and 1200 bar, depending on the engine's combustion configuration. The quantity and timing of injection must be precisely controlled to achieve good mixture quality and to minimize exhaust emissions.

Fuel Injection Process. An engine-driven camshaft (in-line pump) or cam plate (distributor pump) drives the injection pump's plunger in the supply direction, creating pressure in the high-pressure gallery. The delivery valve responds to the increase in pressure by opening. This sends a pressure wave to the injection nozzle at the speed of sound. The needle valve in the nozzle overcomes the spring force of the injection nozzle spring and lifts from its seat when the opening pressure is reached. Fuel is then injected from the spray orifices into the engine's combustion chamber. The injection process ends with the opening of the spill port in the plunger and barrel assembly. This causes the pressure in the pump chamber to collapse, which then causes the delivery valve to close. Due to the action of the delivery valve relief collar, the pressure in the injection line is reduced to the "stand-by pressure." The stand-by pressure is determined to ensure that the injector nozzle closes quickly to eliminate fuel dribble, and the residual pressure waves in the lines prevent the nozzles from reopening.

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CHAPTER 13

TRANSMISSION CONTROL

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13.1 INTRODUCTION

In North America and Japan, 80 to 90 percent of all passenger cars sold have automatic transmissions (ATs), but in Europe only 10 to 15 percent of passenger cars sold have ATs. There are two main reasons for the difference. In Europe, drivers tend to view ATs, compared to manual transmissions, as detrimental to driveability and responsible for a somewhat higher fuel consumption. But implementation of electronic control concepts has invalidated both of those arguments.

Since the introduction of electronic transmission controls units (TCUs) in the early 1980s by Renault and BMW (together with a four-speed transmission from Zahnradfabrik Friedrichshafen, or ZF), the acceptance of the AT rose steeply, even in Europe. For this reason, all new ATs are designed with electronic control. The market for ATs is divided into stepped and continuously variable transmissions (CVTs). For both types the driver gets many advantages. In stepped transmissions, the smooth shifts can be optimized by the reduction of engine torque during gear shift, combined with the correctly matched oil pressure for the friction elements (clutches, brake bands). The reduction of shift shocks to a very low or even to an unnoticeable level has allowed the design of five-speed ATs where a slightly higher number of gear shifts occur. In today's standard systems, the driver can choose between sport and economic drive programs by operating a selector switch. In highly sophisticated newer systems, the selection can be replaced by the self-adaptation of shift strategies. This leads not only to better driveability but also to a significant reduction in fuel consumption. Additionally, a well-matched electronic control of the torque converter lockup helps to improve the yield of the overall system. Both automotive and transmission manufacturers benefit from the reduced expense resulting from the application of different car/engine combinations. Different shift characteristics are easy to implement in software, and much adaptation can be achieved by data change, leaving the transmission hardware and TCU unchanged. The reduction of power losses in friction elements increases the life expectancy and enables the optimization of transmission hardware design.

With the CVT, one of the biggest obstacles to the potential reduction in fuel consumption by operating the engine at its optimal working point is the power loss from the transmission's oil pump. Only with electronic control is it possible to achieve the required yield by matching the oil mass-stream and oil pressure for the pulleys to the actual working conditions.

To guarantee the overall economic solution for an electronically controlled transmission, either stepped or CVT, the availability of precision electrohydraulic actuators is imperative.

13.2 SYSTEM COMPONENTS

The components of an electronic transmission control system are a transmission which is adapted to the electronic control requirements and an electronic control unit with corresponding inputs and outputs and attached sensor elements.

13.2.1 Transmission

The greatest share of electronically controlled transmissions currently on the market consists of four- or five-speed units with a torque converter lockup clutch, commanded by the control unit. Market share for five-speed transmissions is continuously increasing. With electronically controlled transmissions there are numerous possibilities to substitute mechanical and hydraulic components with electromechanical or electrohydraulic components. One basic method is to substitute only the shift point control. In a conventional pure hydraulic AT, the gear shifts are carried out by mechanical and hydraulic components. These are controlled by a centrifugal governor that detects the vehicle speed, and a wire cable connected to the throttle plate lever. With an electronic shift point control, on the other hand, an electronic control unit detects and controls the relevant components. In the transmission's hydraulic control unit, mechanical and hydraulic components are replaced by electrohydraulic controlling elements, usually in the form of electrohydraulic on/off solenoids. This way the number of solenoids, as well as the control logic, can be varied over a wide range. For example, for each gear, one specific solenoid can operate the relevant clutch for this gear shift. Alternatively, there can be one solenoid for each gear change, which is switched corresponding to the shift command. In this way, only three solenoids are required in a four-speed transmission. In some current designs, the gears are controlled by a logical combination of solenoid states. This design needs only two gear-controlling solenoids for a four-speed transmission. For five-speed applications, accordingly, three solenoids are required (Table 13.1)

TABLE 13.1 Example of a Gear-Solenoid Combination for a Five-Speed Transmission Application

	Solenoid 1	Solenoid 2	Solenoid 3
1st gear	on	on	on
2nd gear	on	on	off
3rd gear	on	off	off
4th gear	off	off	off
5th gear	off	on	off

The hydraulic pressure is controlled in this basic application by a hydraulic proportional valve which is, in turn, controlled by a wire cable connected to the throttle plate lever. With this design, the shift points can be determined by the electronic TCU, resulting in a wide range of freely selectable driving behaviors regarding the shift points. It is also possible to use different shift maps according to switch or sensor signals. The influence on driving comfort during gear shifting in this electronic transmission control application has important restrictions. The only possible way to control shift smoothness is with an interface to the electronic engine management. This way, the engine output torque is influenced during gear shifting. A systematic wide-range control of the hydraulic pressure during and after the gear shift necessitates the replacement of the hydraulic pressure governor with an electronically controlled hydraulic solenoid. This design allows the use of either a pulse-width-modulated (PWM) solenoid or a pressure regulator. The choice of which type of pressure control solenoid to use results from the requirements concerning shift comfort under all driving conditions. For

present-day designs with high requirements for shift comfort during the entire life of the transmission, at all temperatures, and with varying oil quality, the analog pressure control solenoid is superior to the usual PWM solenoid, providing there is no pressure sensor in operation as a guideline for pressure regulation. This application usually uses one central controlling element in the transmission for the pressure regulation to control the shift quality.

In other transmission developments, the shift quality is further increased using electronically controllable brake elements (brake bands) for some specific gear changes. In this case, the flywheel effect of the revolving elements is limited by an electronic control of a brake band according to an algorithm or special timing conditions.

The most sophisticated transmission application to date is so designed that overrunning clutches are eliminated and gear changes are exclusively controlled by the electronic control unit with pressure regulator solenoids.¹ This application is characterized by extremely high demands on the electronic TCU concerning real-time behavior and data handling. The relationship between weight, transmission outline, and transferrable torque has reached a high level. Compared to transmissions with overrun clutches, the necessary fitting dimensions are reduced.

Present electronically controlled ATs usually have an electronically commanded torque converter clutch, which can lock up the torque converter between the engine output and the transmission input. The torque converter clutch is activated under certain driving conditions by a solenoid controlled by the electronic TCU. The solenoid design, depending on the requirements of TCC functions and shift comfort, can either be an on/off solenoid, a PWM solenoid, or a pressure regulator. Locking up the torque converter eliminates the slip of the converter, and the efficiency of the transmission system is increased. This results in an even lower fuel consumption for cars equipped with AT.

13.2.2 Electronic Control Unit

Another important component in electronic transmission control is the electronic control unit, which is designed according to the requirements of the transmission and the car environments. The electronic control unit can be divided into two main parts: the hardware and the corresponding software.

Hardware. The hardware of the electronic control unit consists of the housing, the plug, the carrier for the electronic devices, and the devices themselves. The housing, according to the requirements, is available as an unsealed design for applications inside the passenger compartment or within the luggage compartment. It is also possible to have sealed variants for mounting conditions inside the engine compartment or at the bulkhead. The materials for the housing can be either various plastics or metals. There are many different nonstandardized housings on the market. The various outlines and plug configurations differ, depending upon the manufacturer of the electronic unit. The plug configuration, i.e., the number of pins and the shape, depends on the functions and the requirements of the automotive manufacturer. The number of pins is usually less than 100. Some control unit manufacturers try to standardize their plugs and housings throughout all their electronic control units, such as engine management, ABS, traction control, and others. This is important to simplify and to standardize the unit production and the tests during manufacturing.

The carrier for the electronic devices is usually a conventional printed circuit board (PCB). The number of layers on the PCB depends on the application. For units with a complex device structure and high demands for electromagnetic compatibility, multilayer applications are in use. In special cases, it is possible to use ceramics as a carrier. There are usually some parts of the electronic circuit, resistors for example, designed as a thick-film circuit on the hybrid. In this case the electronic unit is manufactured as a solder hybrid or as a bond hybrid with direct-bonded integrated circuit devices. Some single applications exist with a flex-foil as a carrier for the electronic devices. These applications are limited to very special requirements.

The transmission control area requires some specially designed electronic devices, in particular, the output stages for the actuators of pressure regulation and torque converter clutch control. These actuators for pressure control have extremely high demands regarding accuracy of the actuator current over the whole temperature range and under all conditions independent of battery voltage and over the entire lifetime. There are some known applications of customer-specific integrated circuits or devices. Here, special attention paid to quality and reliability over the entire lifetime is necessary to meet the continuously increasing quality requirements of the automotive market. Currently, there is an increasing spread of surface-mounted devices in transmission control applications. This is why the unit size is continuously decreasing despite an increasing number of functions.

On the functional side, the hardware configuration can be divided into power supply, input signal transfer circuits, output stages, and microcontroller, including peripheral components and monitoring and safety circuits (Fig. 13.1). The power supply converts the vehicle battery voltage into a constant voltage required by the electronic devices inside the control unit. Accordingly, special attention must be paid to the protection of the internal devices against destruction by transients from the vehicle electrical system such as load dump, reverse battery polarity, and voltage peaks. Particular attention is also necessary in the design of the elec-

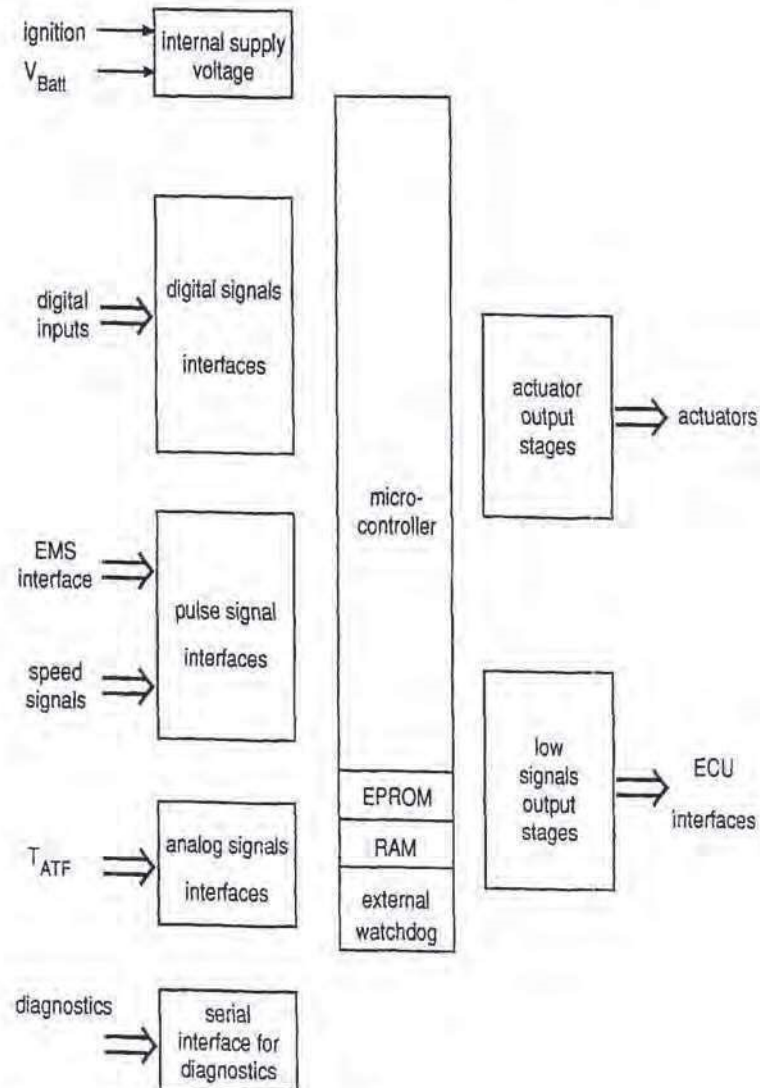


FIGURE 13.1 Overview of hardware parts.

tronic ground concept for the control unit, especially where the electromagnetic compatibility and RF interference is concerned. This is very important to prevent undesired gear shifting that may be troublesome for drivers. One part of the input circuit is the preparation of the digital signals, such as position switch, program selector, and kickdown switch. A second part is the transfer of the analog signals like ATF temperature and voltages according to potentiometer states. The third part is the interface to other electronic control units, especially to the engine management system. Here the single signal lines between the control units will be increasingly substituted by bus systems like CAN. The fourth part is the preparation of the transmission-specific signals from the speed sensors inside the transmission.

The calculators inside the control units are usually microcontrollers. The real-time requirements and the directly addressable program storage size of the selected microcontrollers are determined by the functions of the transmission control and the car environment. In present applications, either 8-bit or 16-bit microcontrollers are in use. There are systems with 32-bit microcontrollers in development for new, highly sophisticated control systems with increasing functional and extreme real-time requirements originating from the transmission concept. The memory devices for program and data are usually EPROMs. Their storage capacity is, in present applications, up to 64 Kbytes. Future applications will necessitate storage sizes up to 128 Kbytes. The failure storages for diagnostics and the storage for adaptive data are in conventional applications, battery voltage-supplied RAMs. These are increasingly being replaced by EEPROMs.

There are usually watchdog circuits in various configurations in use regarding safety and monitoring. These can be either a second, low-performance microcontroller, a customer-specific circuit, or a circuit with common available devices. The output stages can be divided into high-power stages for the transmission actuator control and low-power stages like lamp drivers or interfaces to other electronic control units. The low-power output stages are mostly conventional output drivers either in single or in multiple applications, which are mainly protected against short circuits and voltage overloads.

For the transmission solenoid control, special output stages are necessary, and they are specialized for operation with inductive actuators. The pressure regulation during shifting in some applications requires high accuracy and current-regulated output stages are needed. These are mainly designed as customer-specific devices. The type and number of solenoid output stages depend on the control philosophy of the transmission: they are generally of a special design for specific transmission applications. During the preparation of the speed sensor signals, attention must be paid to the electromagnetic compatibility and radio frequency interference conditions.

Software. The software within the electronic transmission control system is gaining increasing importance due to the increasing number of functions which, in turn, requires increasing software volume. The software for the control unit can be divided into two parts: the program and the data. The program structure is defined by the functions. The data are specific for the relevant program parts and have to be fixed during the calibration stage. The most difficult software requirements result from the real-time conditions coming from the transmission design. This is also the main criterion for the selection of the microcontroller (Fig. 13.2).

The program is generally made up in several parts:

- Software according to the special microcontroller hardware; e.g., I/O preparation and filter, driver functions, initialization of the microcontroller and the control unit, internal services for the controller peripheral devices, and internal software services like operating systems.
- Software coming from the defined functions, originating from specific transmission and car functions.
- Parts concerning safety functions like output switch-off, substitute values for the input signals, and safety states of the microcontroller environment in case of failures. Depending on the requirements, there can be a software watchdog or a hardware-configured watchdog circuit in use. The watchdog instruction is also part of the security software.

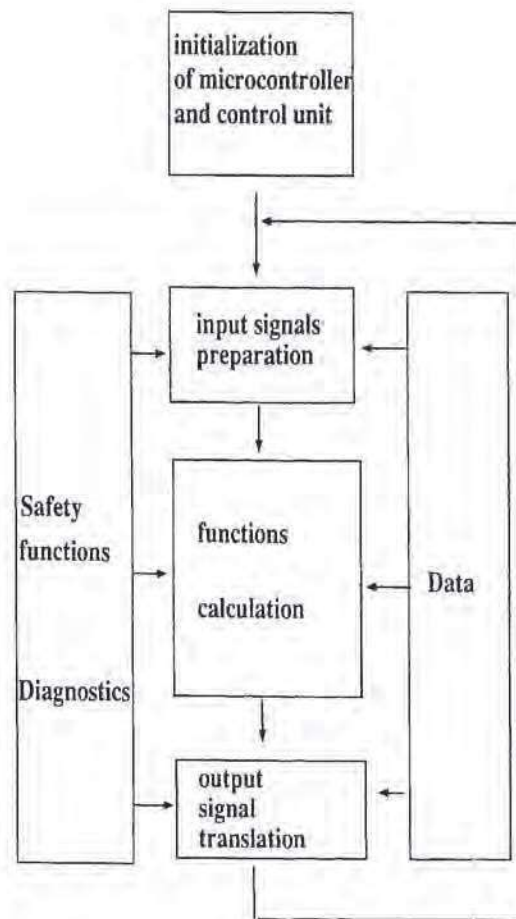


FIGURE 13.2 Software structure overview.

- Diagnostic and communication software for the self-test of the control unit and also the test of the control unit environment.

These functions are related to the defined functions of the electronic control system. Parts of the software component are usually the output stages monitoring, the input monitoring, and the diagnosis of the microcontroller environment. Failure handling and storage is gaining importance as system complexity increases. These diagnostic functions are also very useful for the service station to determine the reason for eventual problems. Part of these functions is reserved for the communication software needed for the test equipment to read the failures stored during car service. Current protocols are standardized communication protocols like ISO 9141. There is an increasing share of bus systems for communication with other electronic control units, using standardized protocols like CAN, VAN, or J1850. These bus systems allow an increasing unit function by changing software when other control units are added to the bus.

Most software models are written directly in an assembler to meet the real-time requirements and because there is a limited memory size in common mass production units. The number of powerful, cost-effective microcontrollers is continuously increasing. The availability of memory components with larger storage sizes suitable for automotive use is also rising, making it possible to use a higher programming language in future developments. This allows an ingenious structure of software models and an application of operating systems. This can be followed by an effective distribution of functions during and outside gear shifting with related time requirements and event management. This type of program structure improves the function of the electronic TCU because of the accelerated handling of time-critical functions during gear shifting.

The second software part, data, can be divided into fixed data, which is related to fixed attributes of the system; e.g., the number of actuators, and calibration data for system tuning. The calibration data can be adapted to changing parameters of the system such as the engine, vehicle, and transmission characteristics. The fixing of calibration data takes place during the tuning stage of the vehicle and has to be redetermined for each type of vehicle and transmission. With some applications, the calibration data are added to a basic program during the vehicle production according to different types of cars by the so-called end-of-line programming. This means that the units can be programmed with the calibration data with closed housings by a special interface. The share of software development in relation to the total development time is increasing continuously. The requirements for real-time behavior and memory size are rising in accordance with the considerably increasing demands for shift comfort and self-learning functions. This requires an ingenious structure of the software and an event-related distribution of software models, especially during gear shifting. The rising software complexity with simultaneously increasing quality requirements causes higher demands for software quality control.

13.2.3 Actuators

Electrohydraulic actuators are important components of the electronic transmission control systems.² Continuously operating actuators are used to modulate pressure, while switching actuators function as supply and discharge valves for shift-point control. Figure 13.3 provides a basic overview of these types of solenoids.

Important qualities for the use of actuators in ATs are low hydraulic resistance to achieve high flow rates, operation temperature range from -40 to $+150$ °C, small power loss, minimized heat dissipation in the ECU's output stages, small size and low weight, highest reliability in heavily contaminated oils, maximum accuracy and repeatability over lifetime, short reaction times, pressure range up to 2000 kPa, maximum vibration acceleration of 300 m/s², and high number of switch operations.

A very important aspect is that the hardware and software of the ECU be developed, taking into account the electrical specifications of the solenoid to obtain an optimized complete system concerning performance and cost.^{6,7} For further details in design and application, refer to Sec. 10.3.5.

It should be noted that these characteristics can be varied over a wide range and that many other types of solenoids exist or are in development for the special requirements of new applications.

13.3 SYSTEM FUNCTIONS

Functions can be designated as systems functions if the individual components of the total electronic transmission control system cooperate efficiently to provide a desired behavior of the transmission and the vehicle. There are different stages of functionality which have different effects on driving behavior and shift characteristics (Fig. 13.4). In general, there is an increasing complexity of the system relating to all components to improve the translation of driver behavior into transmission action. That means that the expense of actuators, sensors, and links to other control units is increasing, as is the expense of the TCU software and hardware in the case of high-level requirements regarding driveability and shift comfort. Figure 13.4 shows three main areas. These will be discussed in detail in the following material.

13.3.1 Basic Functions

The basic functions of the transmission control are the shift point control, the lockup control, engine torque control during shifting, related safety functions, and diagnostic functions for

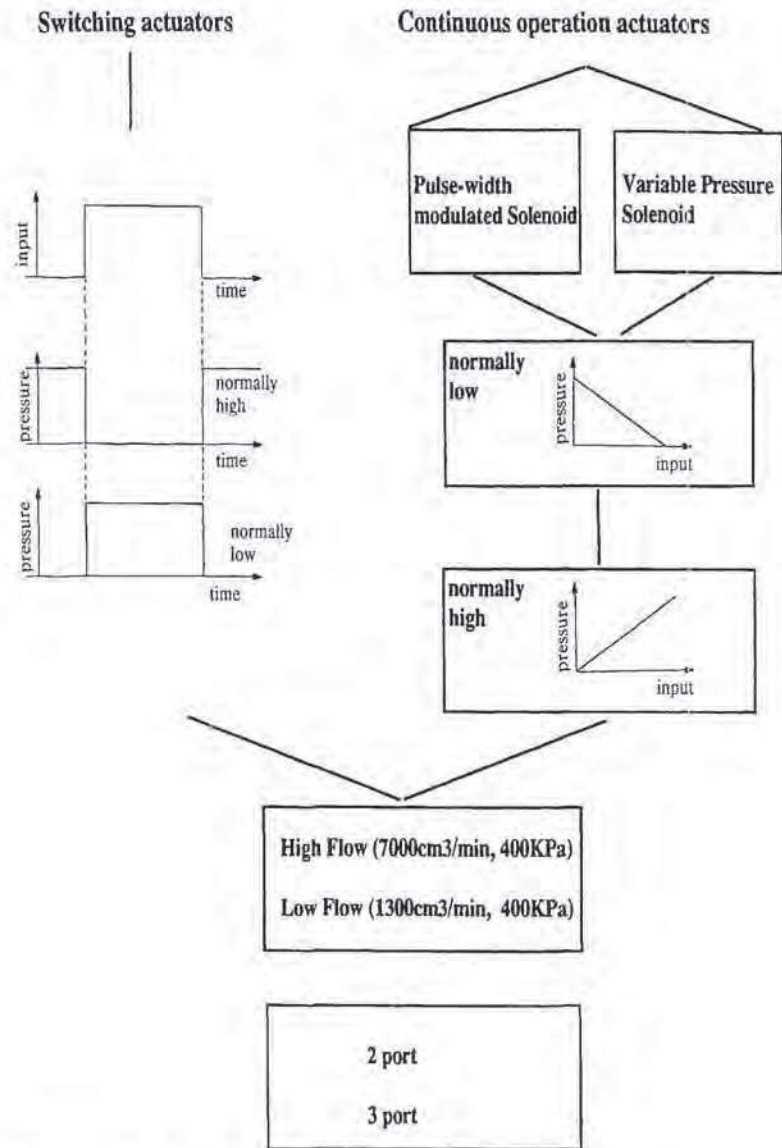


FIGURE 13.3 Electrohydraulic actuators for automatic transmissions.

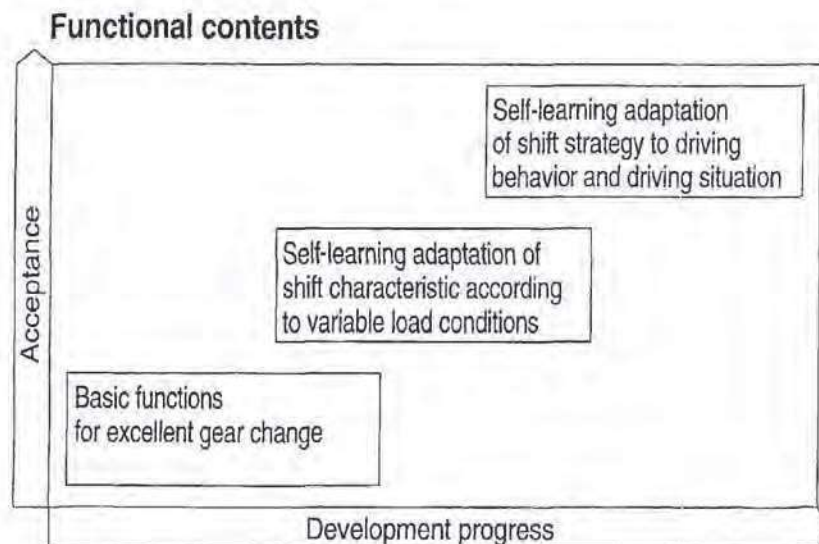


FIGURE 13.4 Relationship between driving characteristic and function complexity.

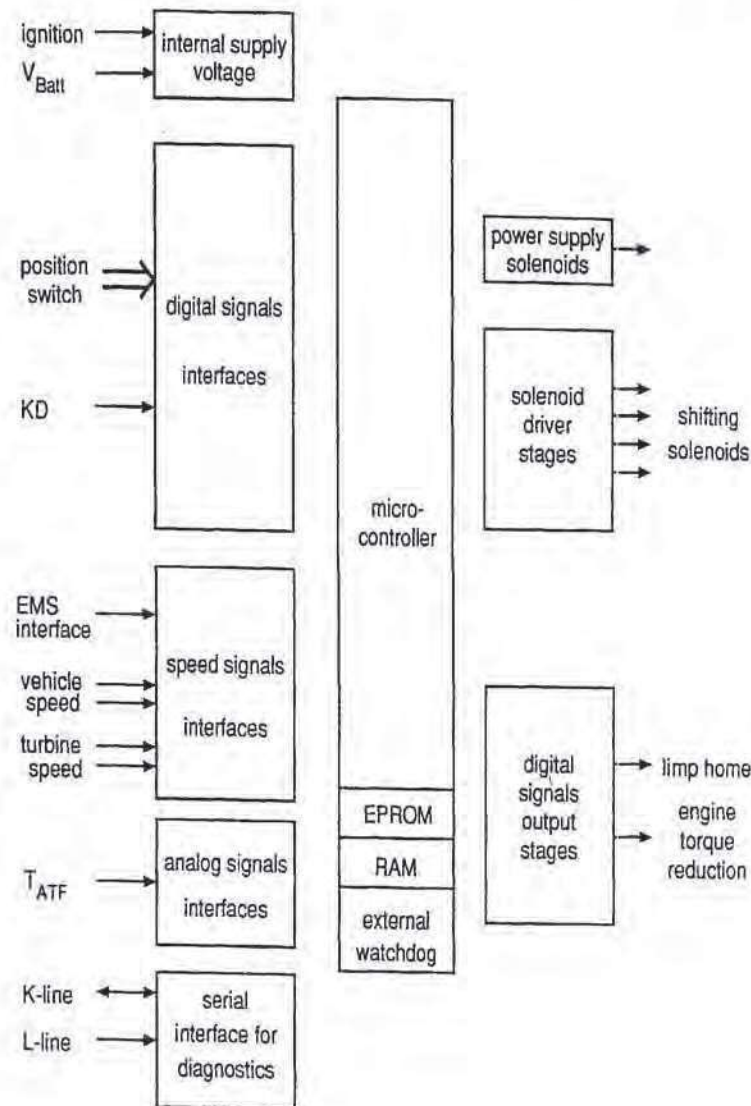


FIGURE 13.5 Structure of a basic transmission electronic control unit.

vehicle service. The pressure control in transmission systems with electrical operating possibilities for the pressure during and outside shifting can also be considered as a basic function. Figure 13.5 shows the necessary inputs and outputs as well as the block diagram of an electronic TCU suitable for the basic functions.

Shift Point Control. The basic shift point control uses shift maps, which are defined in data in the unit memory. These shift maps are selectable over a wide range. The shift point limitations are made, on the one hand, by the highest admissible engine speed for each application and, on the other hand, by the lowest engine speed that is practical for driving comfort and noise emission. The inputs of the shift point determination are the throttle position, the accelerator pedal position, and the vehicle speed (determined by the transmission output speed). Figure 13.6 shows a typical shift map application of a four-speed transmission.

To prevent overly frequent shifting between two gears, a hysteresis between the upshift and the downshift characteristic is incorporated. The hysteresis is determined by the desired shifting habit of the transmission and, alternatively, the car behavior. In the event that the particular shift characteristic is crossed by one of either of the two input valves, the electronic ECU releases the shift by activating the related actuators. This can be a direct shift into the

E-Program (Economy)

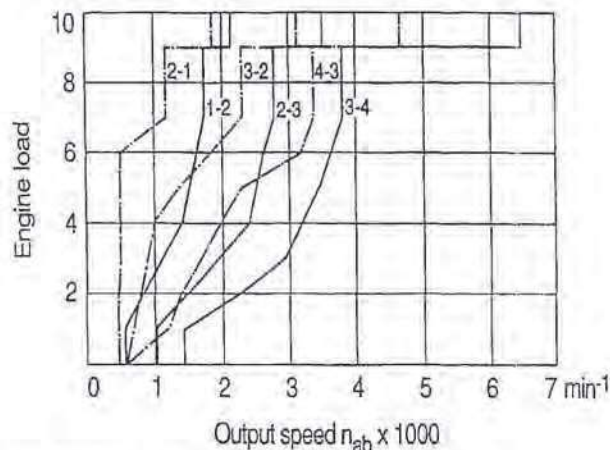


FIGURE 13.6 Shift characteristics of a four-speed application.

target gear or by a serial activation of specific actuators in a fixed sequence to the target gear, depending on the transmission hardware design.

Lockup Control/Torque Converter Clutch.⁸ The torque converter clutch connects both functional components of the hydraulic converter, the pump and the turbine. The lockup of the clutch reduces the power losses coming from the torque converter slip. This is a permanent slip because it is necessary in principle to have a slip between the pumpwheel and the turbine to translate torque from the engine output to the transmission input. To increase the efficiency of the lockup, it is necessary to close the clutch as often as possible. On the other hand, the torque converter is an important component to prevent vibrations of the powertrain. The activation of the lockup is, therefore, a compromise between low fuel consumption and high driving comfort. The shift points of the lockup are determined in the same way as the determination of the shift point in the gear shift point control. Usually there is one separate characteristic curve for the lockup for each gear. To prevent powertrain vibrations, it is advisable to open the lockup during coasting to use the damping effect of the torque converter. In the case of a high positive gradient of the accelerator pedal with low engine speed, the converter clutch has to open to use the torque gain of the converter for better acceleration of the car. In some applications, the lockup is opened during shifting for improved shift comfort. After shifting, the lockup can be closed again. When driving in first gear, the lockup is usually open, because the time spent in first gear is usually very low and, therefore, the frequency of lockup shifting versus gear shifting becomes very high. This may result in decreased driving comfort. A second reason is the improved acceleration of the car in first gear when using the converter gain for wheel torque.

Engine Torque Control During Shifting.⁸ The engine torque control requires an interface to an electronic engine management system. The target of the engine torque control, torque reduction during shifting, is to support the synchronization of the transmission and to prevent shift shocks.

In conventional applications, the engine torque reduction originates from an ignition angle control. The timing and absolute value of the ignition control depends on the operating conditions concerning actual engine torque and shifting type.

Upshift. The upshift occurs without an interruption of the tractive power. The engine torque reduction may be activated if the clutch of the target gear stays with the translation of torque. The beginning of the engine torque reduction is determined by the course of engine or transmission input speed. There it is important to detect a decreasing speed. The start of the

torque reduction is characterized by a specific speed difference. The end of the torque reduction is activated at an applicable speed lead before reaching the synchronous speed of the new gear.

The power losses, which have to be picked up by the clutches, are dependent on the engine torque and the slipping time

$$Q = f \times (M_{\text{eng}} \times t_s + Q_{\text{kin}}) \quad (13.1)$$

where Q = power losses

M_{eng} = engine torque

t_s = slipping time

Q_{kin} = kinetic energy of revolving elements

It is possible to reduce the temperature stress to the clutches by reducing the engine torque and, consequently, by increasing the slipping time at a fixed possible maximum power loss Q [Eq. (13.1)]. Figure 13.7 shows a typical upshift characteristic.

Shift Quality Comparison

Upshift

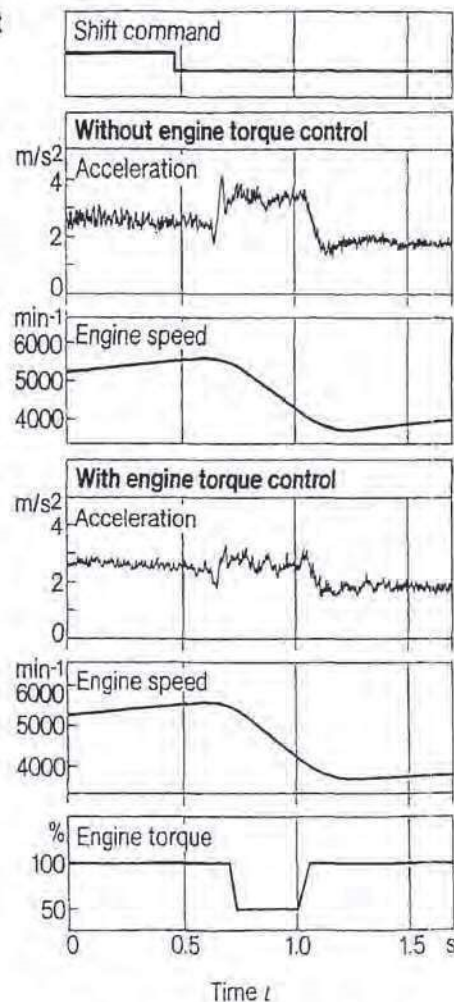


FIGURE 13.7 Timing of engine torque reduction during upshift.

Downshift. Downshift under driving conditions results in a short interruption of the tractive power. At the synchronous point, the tractive power is in operation. The higher revolving energy, on the other hand, results in undesired vibrations of the powertrain. To prevent such

vibrations, it is necessary to reduce the engine output torque before reaching the synchronous point of the new gear. When the transmission input speed reaches the synchronous speed of the new gear, the engine torque has to increase to the nominal value. The increase is usually applied as a torque ramp. Figure 13.8 shows a typical characteristic of a downshift. The values and timing of the engine torque reduction are generally part of the special calibration data for each combination of vehicle, engine, and transmission.

Shift Quality Comparison

Downshift

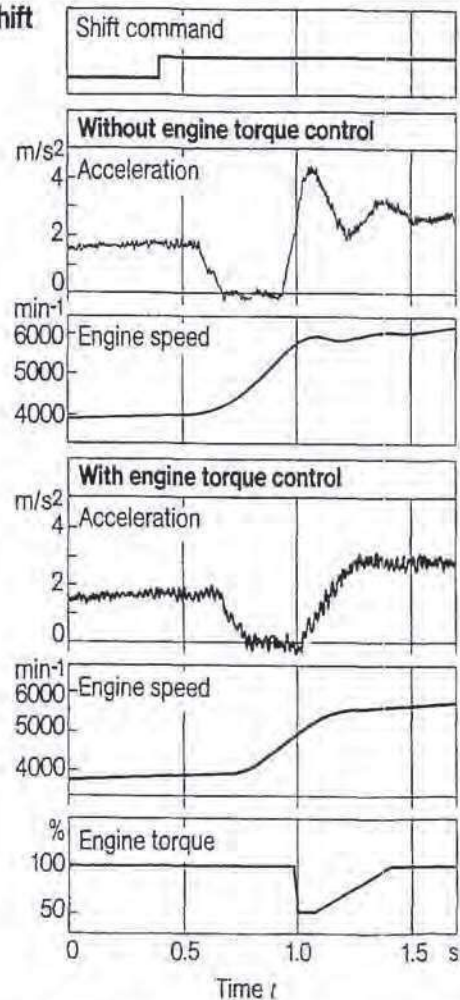


FIGURE 13.8 Timing of engine torque reduction during downshift.

Pressure Control.⁸ The timing and absolute values of the pressure, which is responsible for the torque translation of the friction elements, is, aside from the engine torque reduction, the most important influence to shift comfort. The electronic TCU offers additional possibilities for better function than a conventional hydraulic system.

The pressure values during and outside shifting can be calculated by different algorithms or can be determined by characteristic maps. The inputs for a pressure calculation are engine torque, transmission input speed, turbine torque, throttle position, and so on. The inputs depend on the special signal availability in different systems as well as the requirement concerning shift comfort. The variable pressure components are usually added to a constant pressure value according to the different transmission designs. Equation (13.2) gives a typical algorithm for a pressure calculation.

$$P_{\text{mod}} = P_{\text{const}} + k_n \times P_n + k_{\text{tor}} \times P_{\text{tor}} + k_s \times P_s \quad (13.2)$$

where P_{mod} = pressure

P_{const} = constant pressure value

k_n = adaptation factor for input speed

P_n = pressure component dependent on the revolution signal

k_{tor} = adaptation factor for engine torque

P_{tor} = pressure component dependent on torque

k_s = adaptation factor for vehicle speed

P_s = pressure component dependent on vehicle speed

During applications, the factors must be defined in the calibration phase. In general, the determination of these factors requires many vehicle tests, because the dynamic characteristic of the total system has an important influence on shift comfort. Another possibility for the pressure determination is to use characteristic maps which have to be defined during the calibration phase. This kind of pressure determination allows an improved selection of the optimum pressure at various extreme points independent of an algorithm.

Safety and Diagnostic Functions. The functions, which are usually known as diagnostic functions of the electronic TCU, can be divided into real safety functions to prevent critical driving conditions and diagnostic functions which affect an increasing availability of the car and a better failure detection for servicing. The boundary between safety and diagnostic functions depends on the philosophy of the automotive manufacturer. In the category of real safety functions belong all security functions that prevent uncontrollable shifting, especially unintended downshifting. One section is the monitoring of the microcontroller and its related peripheral devices. The monitoring of the transmission, like gear ratio detection, is also a part of this functional block, as are the actuator and speed sensor monitoring. The microcontroller monitor is usually a watchdog circuit. One possibility is to use the controller internal watchdog. In common applications, it is necessary to use an external watchdog circuit for safety reasons. This can be done with a second, low-performance microcontroller or by a separate hardware watchdog designed as an ASIC or as a conventional circuit device. Usually there is a safety logic circuit connected to the watchdog, which, in the case of a microcontroller breakdown, activates the failure signal and switches the outputs for the transmission actuators to a safety condition.

For the detection of the watchdog, it is necessary to test the watchdog function after each power-on during the electronic initialization. The monitoring of the controller peripheral components, in general EPROM, RAM, and chip-select circuits, works continuously with specific algorithms; e.g., by writing fixed data values to the storage cells and following comparison with the read value or by checksum comparison with fixed sum values. The actuator monitoring includes detection of short circuit to supply voltage and ground, as well as open-load conditions.

In case of actuator malfunction, the limp home mode is selected. This means that the transmission runs in a fixed, safe gear, depending on the driving conditions. The safe state of the actuators is the noncurrent condition, which is secured by the electronic control unit. The control unit can put the output stages into the noncurrent stage separate for each output or by a common supply switch, usually a relay or a transistor. There are some applications that use a combination of both the watchdog and safety circuits.

The monitoring of the transmission-specific sensors, such as input speed, output speed, and oil temperature, works as a plausibility check. For example the transmission input speed can be calculated as a combination of the transmission output speed and the gear ratio. In case of a detected speed sensor malfunction, the limp home mode is generally required. With a temperature sensor failure, the TCU usually works with a substitute value.

The diagnostic functions, which facilitate the finding of failures in the service station, contain the failure storage and the communication to the service tester, which allow the stored

failures of the electronic TCU to be read. The communication between the control unit and the service tester is mainly car manufacturer-specific and must be defined by the car manufacturer before going into series production. The communication runs on a bidirectional, separate communication link.

The failure storage takes place in a nonvolatile memory device; e.g., in a permanent supplied RAM or in an EEPROM. It is also possible to store sporadic failures to detect such problems during the next service. The failure codes, number of stored failures, the handling of the failure storage, as well as the reaction of the TCU in case of a particular failure, is manufacturer-specific and is part of the unit specification. The real safety functions are part of the basic functions of an electronic TCU. The diagnostic functions concerning service tester and communication protocols are, over a wide range, manufacturer-specific. These range from a simple blink code up to a real self-test of the electronic unit, including all peripheral components.

13.3.2 Improvement of Shift Control

In a second development stage, the basic functions can be revised by a modification of the software functions and by adding new parts to the basic functions. This action results in a significant enhancement of the driving and shifting comfort. By a revision of the basic safety and diagnostic functions with so-called substitute functions, it is possible to increase the availability of the vehicle with AT as well as the driveability in case of a malfunction.

Shift Point Control. The basic function can be improved significantly by adding a software function, the so-called adaptive shift point control.⁸ This function requires only signals which are available in an electronic TCU with basic functions. The adaptive shift point control is able to prevent an often-criticized attribute, the tendency for shift hunting especially when hill climbing and under heavy load conditions.

The adaptive function calculates the vehicle acceleration from the transmission output speed over time. The value of the actual acceleration in relation to a set value of the acceleration is the input for the shift point correction. The set value is given by the traction resistance characteristic. For a certain difference between set and actual value, the adaptation of the shift point occurs. The dimension of the shift point correction can be determined by calibration data and depends in general also on the actual vehicle speed and the engine load.

The shift point correction leads to higher hysteresis between upshift and downshift characteristics. With a high difference between set and actual values, it is also possible to forbid certain gears. The return to the basic shift point control is organized by software and can be fixed by calibration data. Usually, in the case of power-on, the adaptive shift point control is reset (Figs. 13.9 and 13.10).

In addition to these functions, different shift maps can be implemented into the data field of the TCU. For example, it is possible to have one shift map for low fuel consumption, which has shift points in the range of the best efficiency of the engine, and additionally to have another map for power operation, where the shift points are placed at points of highest engine output power. The character and number of different shift maps can be selected over a wide range. The choice of the different shift maps can be done by a selector push button or switch commanded by the driver. In further applications, the changing of the different shift programs is possible by self-learning strategies. It is also possible to implement a manual program in which fixed gears are specific to predetermined positions of the selector lever.

Lockup Control. There are some additional functions which can improve considerably the shift comfort of the lockup. In a first step, it is possible to replace the on/off control of the lockup actuator by a pulse control during opening and closing. This can be achieved using conventional hardware only by a software modification. In a further step, the on/off solenoid is replaced by a pressure regulator or a PWM solenoid.

By coordinating intelligent control strategies and corresponding output stages within the electronic TCU, a considerable improvement of the shift behavior of the lockup results. Here

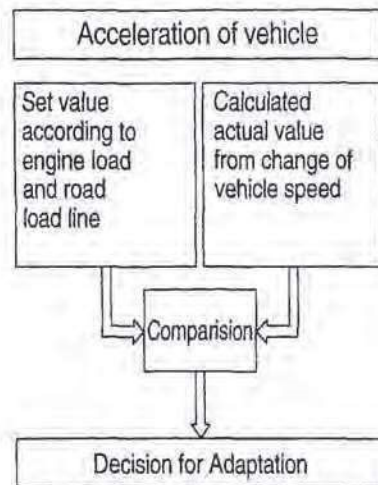


FIGURE 13.9 Basic principle of adaptive shift point control.

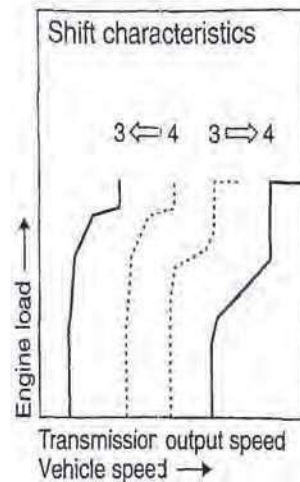


FIGURE 13.10 Shift characteristics before (---) and after (—) adaptation.

it is possible to close the lockup at low engine speed and low engine load with good shift comfort, resulting in decreased fuel consumption.

Engine Torque Reduction During Gear Shifting. By an improved interface design to the engine management system, it is possible to extend the engine torque reduction function. It is necessary to use a PWM signal with related fixed values or a bus interface. The engine torque reduction is controlled directly by the TCU. The advantage of such an interface is an independent calibration of the TCU data over a wide range without changing the engine management data. A further advantage is the improved possibility for the coordination of the engine torque reduction and the pressure control within the TCU. The improvement of this interface can be extended up to a real torque interface, especially when using a bus communication link.

Pressure Control.⁸ The pressure control can be improved in a similar way as the shift point control with an adaptive software strategy. The required inputs for the adaptive pressure control are calculated from available signals in the transmission control. The main reasons for the implementation of the adaptive pressure control are the variations of the attributes of the transmission components like clutch surfaces and oil quality as well as the changing engine output torque over the lifetime of the car.

The principle of adaptive pressure control is a comparison of a set value for the shift time with an actual value, measured by the transmission input speed course. At a specific difference of the set value to the actual value, the pressure value is corrected by a certain increment in the positive or negative direction. The original adaptation time and the pressure value increment were fixed during the calibration phase. For safety reasons, the total deviation of the pressure value from a given value is limited, depending on the particular application. Usually the correction values are stored in the nonvolatile memory to have the correct values available after power-on of the electronic TCU.

Safety and Diagnostic Functions. The safety functions extend over better monitoring of the selector lever and functions concerning misuse by the driver. With a corresponding transmission hardware design, the implementation of a reverse gear inhibit function is possible; i.e., above a certain vehicle speed, the position R is blocked hydraulically by a single solenoid or by a particular solenoid combination commanded by the electronic TCU. This function pre-

vents the destruction of the transmission in the event of an unintentional shift to the reverse gear. Downshift prevention is part of the safety function, especially during manual shifting by the driver. Here the synchronous speed of the new gear is calculated and compared with the admissible maximum engine speed. In the case of a calculated synchronous speed above the maximum engine speed, the downshift is prohibited by the TCU. This function can be supported by an overrun safeguard which releases the limp home mode in case of exceeding the admissible maximum engine speed.

All of those functions can be extended and have to be defined during the development stage by the automobile transmission and electronic TCU manufacturers. To increase availability of the AT system, even with the failure of certain signals, it is possible to provide a substitute operation with better drivability than in the limp home mode. This can be done by substitute functions. The electronic TCU falls back on substitute values or signals in the case of a breakdown of certain interfaces. There is, for example, the possibility to run with a programmable fixed throttle value with a breakdown of the throttle position signal. This results in a reduction of the shift characteristics to shift points. Shifting into all gears is possible, however, with reduced shift comfort. A further method is to use secondary signals in case the original signals break down. For example, the calculation of vehicle speed can be from wheel speed during breakdown of the transmission output speed signal. This technique usually requires a connection between ABS and transmission control. The third variant is the canceling of certain functions if the necessary input signals are missed. For example, in the case of a kickdown switch failure, the kickdown function is canceled. This results in no downshift after operation of the kickdown. Downshifts are nevertheless still possible via the full-throttle opening point according to the full-load shift characteristic.

The availability and driveability of automobiles equipped with electronic TCU in case of system failures can be improved significantly with the implementation of substitute functions. This results in a considerable increase in acceptance by the drivers of automobiles with electronic transmission control.

13.3.3 Adaptation to Driver's Behavior and Traffic Situations

In certain driving conditions, some disadvantages of the conventional AT can be prevented by using self-learning strategies.⁹ This is especially valid when improving the compromise in the shift characteristics regarding gear selection under particular driving conditions and under difficult environmental conditions. The intention of the self-learning functions is to provide the appropriate shift characteristic suitable to the driver under all driving conditions. Additionally, the behavior of the car under special conditions can be improved by suitable functions. Available input signals of the car, provided by the related electronic TCUs from interfaces and communication links, are processed by the TCU with specific algorithms. The self-learning functions can be divided into a long respectively medium term adaptation for driver's style detection and into a short-term adaptation which reacts to the present driving situation, such as hills or curves.

The core of the adaptive strategies is the driver's style detection. The driver's style can be detected by monitoring of the accelerator pedal movements. The inputs are operation speed, operation frequency, and the rating position of the accelerator pedal. These inputs are processed depending on priorities with special algorithms related to the desired driving behavior of the car. The calculated driver style is related to certain shift maps. There is a large choice of shift maps available. With the currently known applications, there are mostly four different shift maps ranging from fuel economic to extremely sporty vehicle behavior. The calculated driver's style can also depend on the actual vehicle speed and the share of constant driving conditions during a certain driving cycle. These self-learning functions can be calibrated by the car manufacturer, depending on his philosophy and target market. In this way, the number of shift maps and the speed of the adaptation have the main influence. A further possibility to match the driver's style is by rating the accelerator pedal operation during vehicle start, for example, after a red light stop. In this way the operation speed and frequency of the accelera-

tor pedal below a certain vehicle speed can be interpreted and calculated as part of the driver's style rating. In the event of kickdown, the shift maps of the driver's style rating are shut down by a priority command. The driver has the usual behavior of the car during kickdown, generally a downshift, providing no other safety function is in operation.

To prevent shift hunting, the self-learning functions are carried out over a long respectively medium term adaptation with the adaptation timer ranging from several seconds up to one minute. The second part of the self-learning functions is the driving condition detection. There is a correlation between the input signals of the transmission control and the driving condition.

One of the main disadvantages of a conventional electronic transmission control is the upshifting at constant vehicle speed by crossing the upshift characteristic with a reduction of the accelerator pedal angle. This results in an unintended gear shift, especially when cornering and when approaching a crossing or an obstacle. To prevent these gear shifts it is possible to use so-called upshift prevention. Cornering can be detected by the acceleration of the car along the driving direction related to the vehicle speed. The vehicle speed is calculated from the transmission output speed. The acceleration can be detected by an acceleration sensor or by the difference between the nondriven wheel speeds. In this way it is possible to prevent the upshift when cornering, resulting in a considerable improvement in vehicle stability.

The detection of a crossing or obstacle approach is possible by the detection of a fast off condition of the accelerator pedal. At a certain gradient of the pedal position, the upshift is prevented. This is a considerable advantage especially when overtaking low-speed vehicles. With this strategy the correct gear is available without a shift delay.

Another part of the driving situation detection is the recognition of uphill driving and full-load conditions. This is possible by adding special functions to the adaptive gear shift control. When driving downhill, it makes sense to support the engine braking effect for a better deceleration of the car. Downhill driving can be detected by a comparison of throttle position and vehicle speed gradient. An upshift is prevented and, in some special cases, a downshift is activated by the electronic TCU.

A further section of the self-learning functions is the environmental monitoring with related shift strategies. A special application can be a self-learning winter program. The wheel slip of the driven wheels is compared with a set value of a combination of given wheel torque and vehicle speed. When exceeding a set limit of wheel slip, a special shift strategy is chosen. For example, the vehicle starts off in second gear or an upshift takes place at lower engine speeds.

The development of adaptive shift strategies started a few years ago and is currently one of the main areas in electronic transmission development. The efficiency of the self-learning functions has led to a wide acceptance of AT-equipped vehicles. The future development concerning new adaptive functions and an improvement of the already known functions is an important area in control development. This can be supported by an increasing share of electronic units and interfaces for the communication between units. With multiple use of sensors providing the necessary input signals, the total system gains increased functionality, especially with bus systems.

At present, an increasing share of manual programs with an AT can be registered. The driver instructs the AT to shift via a switch or a push button. In this manner, the driver can operate the AT like a manual gearbox independently of other shift maps, with only the safety functions in operation. This has led to a broad acceptance, especially in the sports car market. These functions can all be calibrated and applied by the car manufacturer with data relating to his philosophy and to the target market. The result is the prevention of the known disadvantages of the conventional AT control without canceling the advantages in driving comfort and safety.

13.4 COMMUNICATIONS WITH OTHER ELECTRONIC CONTROL UNITS

With the existence of electronic control units for various applications in vehicles, many opportunities exist to link these ECUs and to establish communications between them. The main partner of the TCU is the engine management system. Due to the coupling of engine and

transmission within the vehicle powertrain, it is necessary to have an interface between these ECUs for a functional coupling and an interchange of signals. It is essential for the pressure control inside the transmission control to sensor the engine load, the engine speed, and the throttle position. The engine torque reduction during shifting is also important to establish a good shift comfort and a satisfactory lifetime for the clutches. By handing over certain signals like position lever state, lockup condition, or shift commands to the engine management, the driving comfort of the vehicle can be improved significantly. An interface to ABS and traction control is useful for some self-learning functions in the transmission control when using the wheel speeds.

It is possible to implement certain shift strategies in the transmission control as an active support for ABS and traction control. A link to the electronic throttle control or cruise control makes it possible to optimize certain functions for the total vehicle. By interfaces between the ECUs, a reduction of the sensor expense results by a multiple use via communications. Suitable links include, especially, PWM or bus configurations for trouble-free communication. Bus systems in particular have the advantage of the link-up of additional ECUs without changing their existing hardware. Additional coupling requires only a software change. The interchange of required supplementary signals for new functions is possible without any problems. An example of powertrain management by coupling the powertrain ECUs to achieve lower fuel consumption, simultaneously improving the driveability, is described as follows.

13.5 OPTIMIZATION OF THE DRIVETRAIN

The newest generation of transmission controllers has overcome the former disadvantage regarding fuel efficiency. Adaptive functions in cooperation with carefully designed torque converter clutch control,⁸ which allows the clutch to be closed even at low gears, have improved fuel consumption significantly. Based on the driver's behavior, together with an adaptive shift strategy as previously described, part of the TCU's adaptive program software may select an economy or even super-economy shift strategy whenever possible. There is, however, still more potential for fuel economy by optimization of the drivetrain.

The concept called Mastershift¹⁰ is shown in Fig. 13.11. The basic idea is to interpret the accelerator pedal position as an acceleration request. That acceleration request, or a request for wheel torque, has to be converted by operating the engine at high torque, i.e., open throttle and low rpm values. In order to realize this, it is necessary to use an electronic throttle control system. The communication between the electronic throttle, the engine, and transmission is shown in Fig. 13.12.

Mastershift, concept for drivetrain optimization

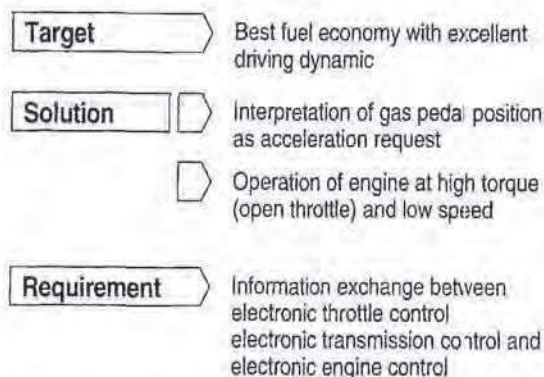


FIGURE 13.11 Drivetrain operation.

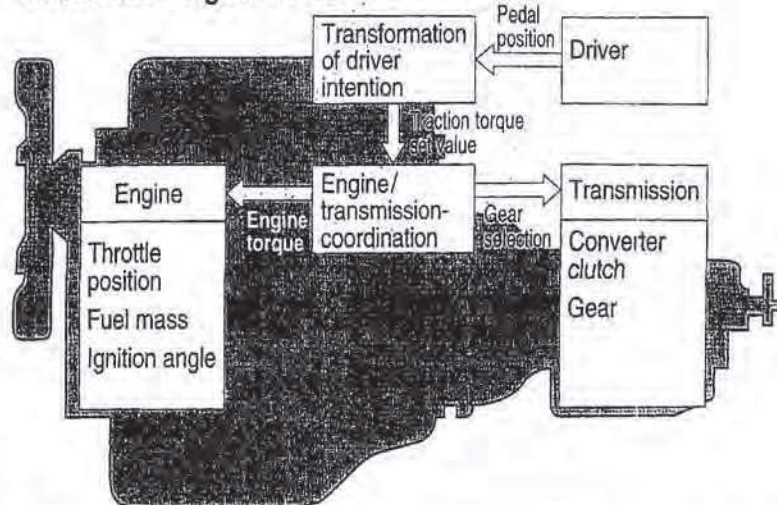
Mastershift – logical structure

FIGURE 13.12 Mastershift: logical structure and communication between different control systems.

In such a system, a well-defined coordination between the engine torque, mainly given by throttle position (air mass), fuel mass, and ignition angle on one side and selection of the appropriate gear including torque converter clutch on the other side, is imperative. Depending on the type of engine, fuel consumption can be reduced further by 5 to 10 percent with this optimized Mastershift concept. Because the average engine operation is at higher torque levels compared to standard systems, a greater number of gear shifts may occur. This is important to guarantee optimal shift comfort. Figure 13.13 shows how that can be accomplished by using the additional degree of freedom given by the electronic throttle control. It is possible to operate the throttle angle during the gear shift in such a way as to achieve constant wheel torque before and after downshifts.

13.6 FUTURE DEVELOPMENTS

In future years, development work will be concentrated on redesign of hardware components for cost reduction, improvement of yield to reduce fuel consumption, and improvement of driveability. A good approach to meet cost targets on the electronic hardware side would be to integrate two or more individual control modules into a common housing. Regarding the electronic components, one could continue using two separate microcontrollers. This would have the advantage that the software development and application could be done individually for two different systems, for example engine and transmission controllers. Another approach could be to mount the TCU on the transmission housing itself. This could lead to a significant reduction in the expense for the wiring harness. Here, however, the problem of hostile ambient temperatures on electronic components has to be solved. Today's stand-alone actuators could be integrated into a common housing similar to the solution shown by Chrysler Corp. in its A 604 transmission.

The improvement of the yield is a main topic for designers of ATs. Oil pumps and torque converters are a major source of energy losses. A significant improvement of yield will be possible as soon as torque converter clutches are available with the capability for continuous slip operation. The torque converter clutch can then be operated in low gears and at low engine speeds without facing problems from drivetrain oscillations and/or noise emission.

The driveability is the most important feature for the drivers' acceptance of ATs. In addition to the self-adaptive functions described, the implementation of shift strategies benefiting from control algorithms using fuzzy theory may further improve driveability.

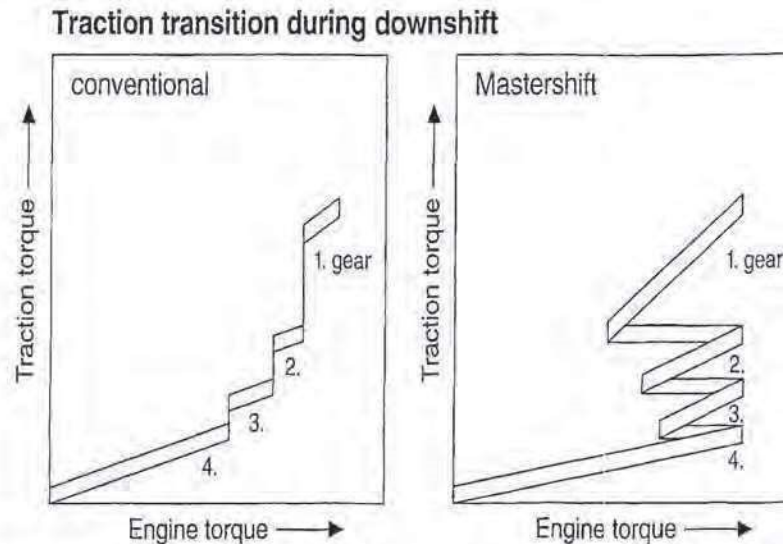


FIGURE 13.13 Constant traction torque by operation of throttle opening during gear shift.

GLOSSARY

- ASIC** Application-specific integrated circuit.
- AT** Automatic transmission.
- ATF** Automatic transmission fluid.
- CAN** Controller area network.
- CVT** Continuously variable transmission.
- EEPROM** Electrically erasable and programmable read-only memory.
- EMC** Electromagnetic compatibility.
- EPROM** Erasable programmable read-only memory.
- PWM** Pulse-width modulation.
- RAM** Random access memory.
- RFI** Radio frequency interference.
- TCC** Torque converter clutch.
- TCU** Transmission control unit.

REFERENCES

1. F. Kucukay and Lorenz, K., "Das neue Fünfgang-Automatikgetriebe für V8-Motoren in der 7er Bau-reihe von BMW," *ATZ Automobiltechnische Zeitschrift* 94, Heft 7/8, 1992.
2. K. Neuffer, "Recent development of AT-control: adaptive functions and actuators," Symposium No. 9313, *Advanced Technologies in Automotive Propulsion Systems*, Society of Automotive Engineers of Japan Inc., 1993, pp. 42-49.

3. J. G. Eleftherakis and Khalil, A., "Development of a laboratory test contaminant for transmissions," SAE Paper 90 0561, Society of Automotive Engineers, Warrendale, Pa.
4. B. Aldefeld, "Numerical calculation of electromagnetic actuators," *Archiv für Elektrotechnik*, Bd. 61, 1979, pp. 347-352.
5. K. Hasuuaka, Takagi, K., and Sinji, W., "A study on electro-hydraulic control for automatic transmissions," SAE Paper 89 2000, Society of Automotive Engineers, Warrendale, Pa.
6. P. C. Sen, "Principles of electric machines and power electronics," J. Wiley, New York, 1989.
7. "Method and Apparatus to Convert an Electrical Value into a Mechanical Position by Using an Electromagnetic Element Subject to Hysteresis," U. S. Patent 4,577,143 March 18, 1986.
8. K. Neuffer, "Elektronische Getriebesteuerung von Bosch," *ATZ Automobiltechnische Zeitschrift* 94, Heft 9, 1992, pp. 442-449.
9. A. Welter, et al., "Die Adaptive Getriebesteuerung für Automatikgetriebe der BMW-Fahrzeuge mit Zwölfzylindermotor," *ATZ Automobiltechnische Zeitschrift* 94, 1992, pp. 428-436.
10. H. M. Streib and R. Leonhard, "Hierarchical control strategy for powertrain function," *XXIV Fisita Congress*, London, 1992.

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CHAPTER 14

CRUISE CONTROL

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14.1 CRUISE CONTROL SYSTEM

A vehicle speed control system can range from a simple throttle latching device to a sophisticated digital controller that constantly maintains a set speed under varying driving conditions. The next generation of electronic speed control systems will probably still use a separate module (black box), the same as present-day systems, but will share data from the engine, ABS, and transmission control systems. Futuristic cruise control systems that include radar sensors to measure the rate of closure to other vehicles and adjust the speed to maintain a constant distance are possible but need significant cost reductions for widespread private vehicle usage.

The objective of an automatic vehicle cruise control is to sustain a steady speed under varying road conditions, thus allowing the vehicle operator to relax from constant foot throttle manipulation. In some cases, the cruise control system may actually improve the vehicle's fuel efficiency value by limiting throttle excursions to small steps. By using the power and speed of a microcontroller device and fuzzy logic software design, an excellent cruise control system can be designed.

14.1.1 Functional Elements

The cruise control system is a closed-loop speed control as shown in Fig. 14.1. The key input signals are the driver's speed setpoint and the vehicle's actual speed. Other important inputs are the faster-accel/slower-coast driver adjustments, resume, on/off, brake switch, and engine control messages. The key output signals are the throttle control servo actuator values. Additional output signals include cruise ON and service indicators, plus messages to the engine and/or transmission control system and possibly data for diagnostics.

14.1.2 Performance Expectations

The ideal cruise system features would include the following specifications:

- *Speed performance:* ± 0.5 m/h control at less than 5 percent grade, and ± 1 m/h control or vehicle limit over 5 percent grade.
- *Reliability:* Circuit designed to withstand overvoltage transients, reverse voltages, and power dissipation of components kept to minimum.

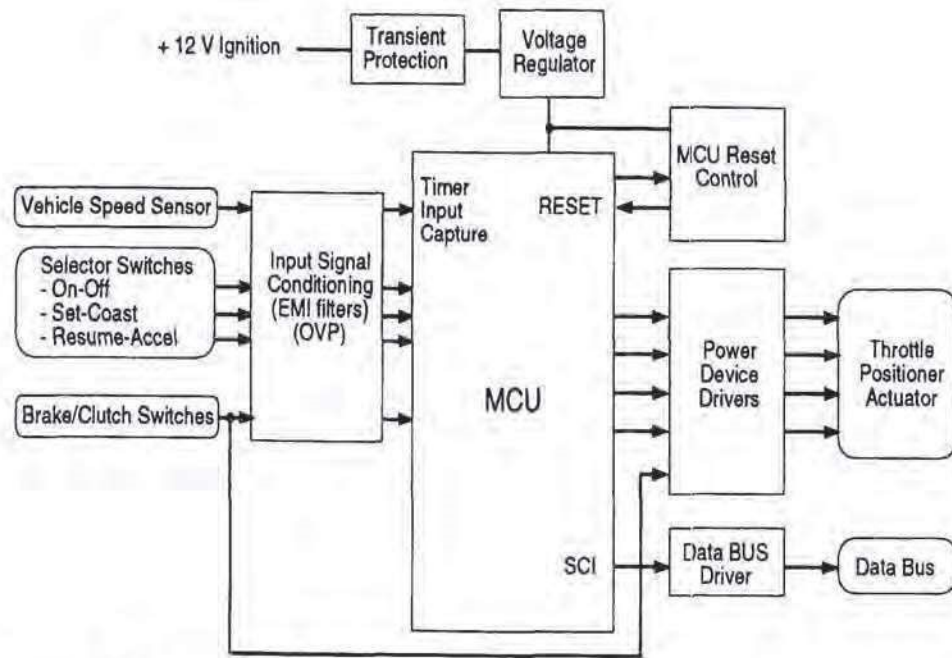


FIGURE 14.1 Cruise control system.

- *Application options:* By changing EEPROM via a simple serial data interface or over the MUX network, the cruise software can be upgraded and optimized for specific vehicle types. These provisions allow for various sensors, servos, and speed ranges.
- *Driver adaptability:* The response time of the cruise control can be adjusted to match the driver's preferences within the constraints of the vehicle's performance.
- *Favorable price-to-performance ratio:* The use of integrated actuator drivers and a high-functionality MCU reduce component counts, increase reliability, and decrease the cruise control module's footprint.

14.1.3 Safety Considerations (Failsafe)

Several safety factors need to be considered for a vehicle speed control design. The most basic is a method designed into the throttle control circuit to insure a failsafe mode of operation in the event that the microcontroller or actuator drivers should fail. This electronic failsafe circuit shuts off the control servos so that the throttle linkage will be released when the brake switch or cruise off switch is activated, no matter the condition of the MCU or servo actuator control transistors. (This assumes the actuators are mechanically in good shape and will release.)

Other safety-related items include program code to detect abnormal operating conditions and preserving into memory the data points associated with the abnormal condition for later diagnostics. Abnormal conditions, for example, could be an intermittent vehicle speed sensor, or erratic driver switch signals. A test could also be made during the initial ignition "key on time" plus any time the cruise is activated to verify the integrity of the cruise system, with any faults resulting in a warning indicator to the driver. Obviously, the most serious fault to avoid is runaway acceleration. Continuous monitoring of the MCU and key control elements will help minimize the potential for this type of fault.

14.2 MICROCONTROLLER REQUIREMENTS FOR CRUISE CONTROL

The MCU for cruise control applications requires high functionality. The MCU would include the following:

- a precise internal timebase for the speed measurement calculations
- A/D inputs
- PWM outputs
- timer input capture
- timer output compares
- serial data port (MUX port)
- internal watchdog
- EEPROM
- low-power CMOS technology

14.2.1 Input Signals

The speed sensor is one of the most critical parts in the system, because the microcontroller calculates the vehicle speed from the speed sensor's signal to within $\frac{1}{2}$ m/h. Any speedometer cable whip or oscillation can cause errors to be introduced into the speed calculation. An averaging routine in the speed calculations can minimize this effect. The speedometer sensor drives the microcontroller's timer input capture line or the external interrupt line. The MCU then calculates the vehicle's speed from the frequency of the sensor signals and the MCU internal timebase. The vehicle's speed value is continually updated and stored into RAM for use by the basic speed control program. Speed sensors traditionally have been a simple ac generator located in the transmission or speedometer cable. The ac generator produces an ac voltage waveform with its frequency proportional to the sensor's rpm and vehicle speed. Optical sensors in the speedometer head can also be incorporated. Usually the speed sensor produces a number of pulses or cycles per km or mile. With the increasing ABS system usage, a backup speed sensor value could be obtained from the ABS wheel speed sensors. The ABS speed data could be obtained by way of a MUX network.

The user command switch signals could either be single MCU input lines to each switch contact or a more complex analog resistor divider type to an A/D input line. Other input signals of interest to the cruise system program would be throttle position, transmission or clutch status, A/C status, actuator diagnostics, engine status, etc., which could be obtained over the MUX data network.

14.2.2 Program Flow

The microcontroller is programmed to measure the rate of vehicle speed and note how much, and in which direction, the vehicle speed is drifting. The standard PI (proportional-integral) method produces one output signal p that is proportional to the difference between the set-speed and actual vehicle speed (the error value) by a proportional gain block K_p . Another signal i is generated that ramps up or down at a rate set by the error signal magnitude. The gains of both K_i and K_p are chosen to provide a quick response, but with little instability. In effect, the PI system adds up the error rate over time, and, therefore, if an underspeed condition occurs as in a long uphill grade, the error signal will begin to greatly increase to try to compensate. Under level driving conditions, the integral control block K_i will tend toward zero

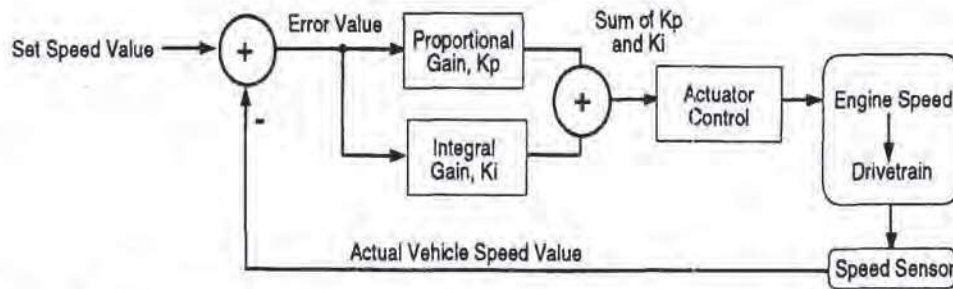


FIGURE 14.2 PI speed error control.

because there is less error over time. The vehicle's weight, engine performance, and rolling resistance all factor in to determine the PI gain constants. In summary, the PI method allows fast response to abrupt grades or mountains and stable operation under light grades or hills. Figure 14.2 shows the traditional PI cruise control diagram.

14.2.3 Output Controls

When the error signal has been computed, an output signal to the servo actuators is generated to increase, hold, or decrease the throttle position. The servo is updated at a rate that is within the servo's mechanical operating specifications, which could be several milliseconds. The error signal can be computed at a much faster rate and, therefore, gives extra time for some averaging of the vehicle speed sensor signal.

Throttle positioning is traditionally either a vacuum type servo or motor. The vacuum supply to the vacuum servo/actuator is discharged as a failsafe measure whenever the brake system is engaged in addition to the normal turn-off of the actuator driver coils. Electric servo type motors require more complex drive electronics and some type of mechanical failsafe linked backed to the brake system.

14.3 CRUISE CONTROL SOFTWARE

The cruise error calculation algorithm can be designed around traditional math models such as PI or fuzzy logic.

14.3.1 Fuzzy Logic Examples

Fuzzy logic allows somewhat easier implementation of the speed error calculation because its design syntax uses simple linguistics. For example: IF speed difference negative and small, THEN increase throttle slightly.

The output is then adjusted to slightly increase the throttle. The throttle position update rate is determined by another fuzzy program which looks for the driver's cruise performance request (slow, medium, or fast reaction), the application type (small, medium, or large engine size), and other cruise system factory preset parameters. Figure 14.3 shows one part of a fuzzy logic design for computing normal throttle position. Other parts would compute the effects of other inputs, such as resume, driver habits, engine type, and the like.

Other program design requirements include verification that the input signals fall within expected boundaries. For example, a broken or intermittent speed sensor could be detected.

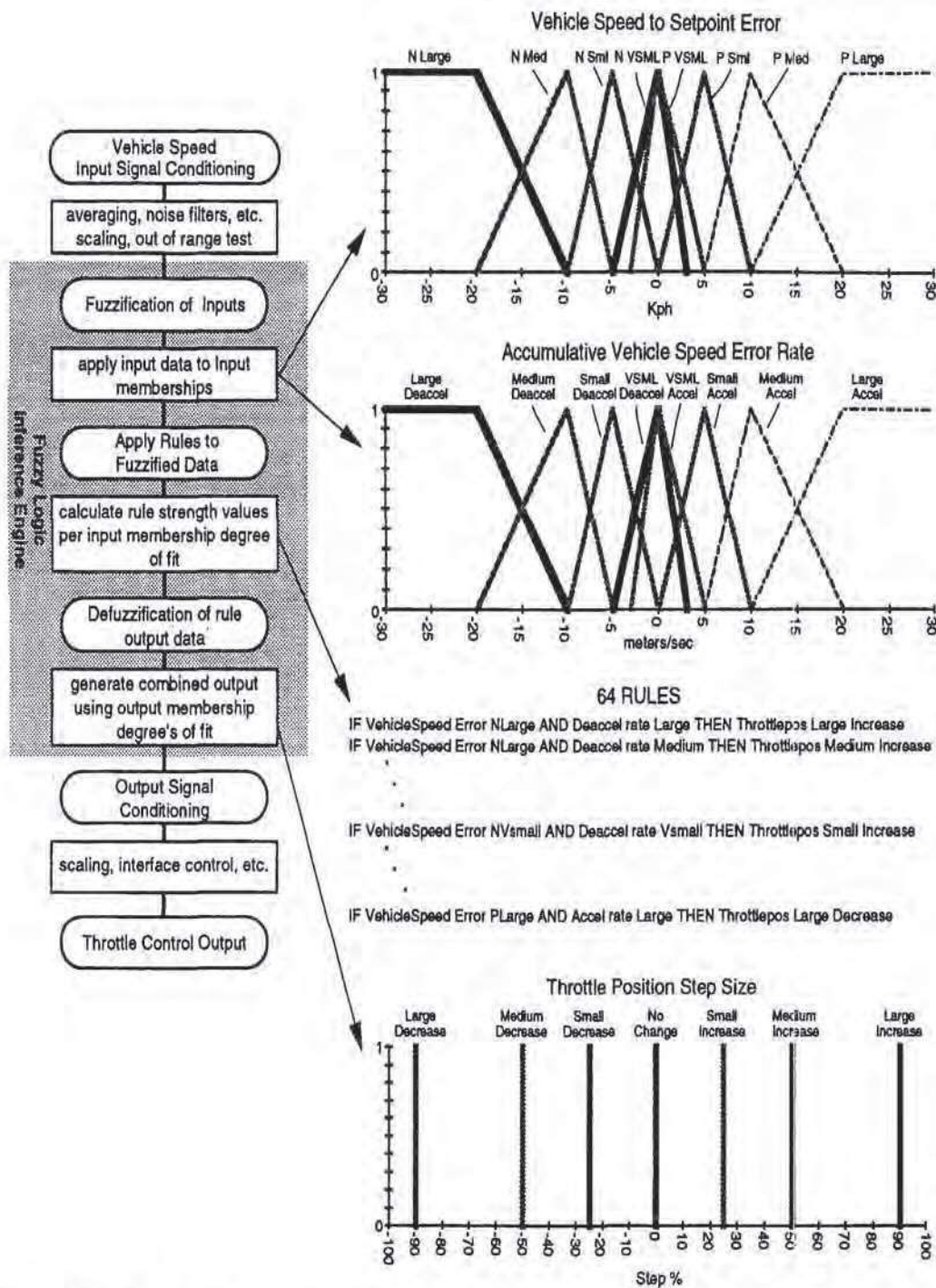


FIGURE 14.3 Fuzzy speed error program flow.

A heavily loaded vehicle with a small engine may not be able to maintain a high setpoint speed up a steep grade, and the cruise control needs to be disengaged to protect the engine from sustained full-throttle operation under a heavy load. This could be preset to occur 20 percent below the setpoint speed. Another program can test the vehicle speed to resume setpoint speed and prevent unsafe acceleration under certain conditions. For example, if a high-performance vehicle (>200-kW or 268-hp engine) has a setpoint speed of 125 km/h (78 mi/h), and drives from the freeway into heavy city traffic doing 48 km/h (30 mi/h) and the vehicle's

driver fortuitously hits the cruise resume switch at this low speed, the cruise control invokes a near full-throttle action, and an accident is likely. A fuzzy design can limit the acceleration upon resume using simple rules such as IF resume and big speed error, THEN increase throttle slightly.

14.3.2 Adaptive Programming

The response time and gain of the cruise system can be adjusted to match individual drivers. For example, some drivers may prefer to allow the vehicle to slow down somewhat when climbing a grade and then respond quickly to maintain a set speed; other drivers may prefer a constant speed at all times, while still other drivers may prefer a very slow responding cruise system to maximize fuel efficiency. The cruise system can be adapted either by a user selection switch (slow, medium, fast) or by analyzing the driver's acceleration/deacceleration habits during noncruise operation. Once these habits are analyzed, they can be grouped into the three previously mentioned categories. One drawback of a totally automatic adaptive cruise system is when various drivers with vastly different driving preferences operate the vehicle on the same trip. The cruise system would have to be "retrained" for each driver.

14.4 CRUISE CONTROL DESIGN

Many of the required elements of a cruise control can be integrated into one single-chip MCU device. For example, the actuator drivers can be designed in the MCU if their power requirements are on the low side.

14.4.1 Automatic Cruise System

Figure 14.4 shows an experimental system design for a cruise control based upon a semicustom 8 or 16-bit single-chip MCU that incorporates special high-power output driver elements and a built-in voltage regulator.

14.4.2 Safety Backup Examples

The design of a cruise control system should include many safeguards:

- A test to determine vehicle speed conditions or command inputs that do not fall within the normal conditions for operation of the cruise control function.
- A test to determine if the vehicle speed has decreased below what the cruise routine can compensate for.
- Speed setpoint minimums and maximums (30 km/h min to 125 km/h max, for example) are checked and, if exceeded, will cause the cruise function to turn off.
- Speedometer cable failure is detected by checking for speed sensor electrical output pulses over a 100-ms time period and, if these pulses are absent, the system is disengaged.
- Software program traps should also be scattered throughout the program and, if memory permits, at the end of each program loop. These will catch an out-of-control program and initiate a vector restart.

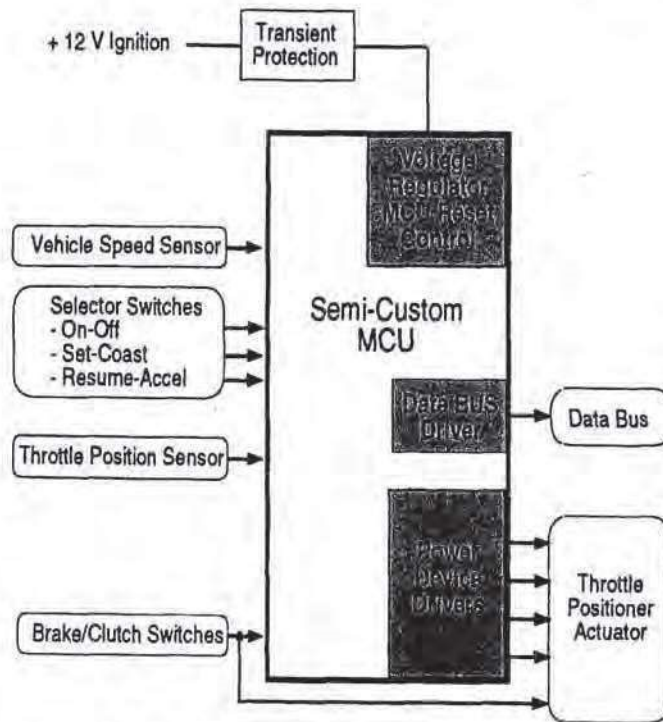


FIGURE 14.4 Automatic cruise control.

14.4.3 EMI and RFI Noise Problems

As with any electronic design, consideration must be given to suppressing RFI (radio frequency interference) from the circuit, besides minimizing effects of external EMI (electromagnetic interference) and RFI to the circuit's normal operation. It is not uncommon that the circuit must operate in RF fields up to 200 V/m intensity. This requires careful layout of the module's PCB (printed circuit board) and RF filters on all lines going in or out of the module. The module case may even have to contain some type of RF shielding. Minimizing generated RFI from the cruise circuit can be accomplished by operating the MCU's crystal oscillator at a minimal power level (this is controlled mostly by the MCU internal design), careful PCB trace layout of the MCU oscillator area, metal shielding over the MCU, ground planes on the PCB under the MCU, and setting the actuator switching edge transition times to over 10 ms. (See Chaps. 27 and 28.)

14.5 FUTURE CRUISE CONCEPTS

Several research projects are underway to develop a crash avoidance system that could be interconnected with a cruise system. The development of a low-cost distance sensor that can measure up to a few hundred meters away with a tight focal point in all weather conditions is proving to be a challenge. When a practical vehicular distance sensor is available, the cruise control can be programmed to maintain either constant speed or constant distance to another vehicle. Other methods of cruise control could include receiving a roadside signal that gives an optimum speed value for the vehicle when travelling within certain traffic control areas.

14.5.1 Road Conditions Integration with IVHS

The IVHS (Intelligent Vehicle-Highway System) network may be a more practical approach to setting optimum cruise speed values for groups of vehicles. The IVHS can monitor road conditions, local weather, etc., and broadcast optimal speed data values for vehicles in its zone. (See Chap. 29.)

GLOSSARY

Analog input Sensors usually generate electrical signals that are directly proportional to the mechanism being sensed. The signal is, therefore, analog or can vary from a minimum limit to a maximum limit. Normally, an 8-bit MCU A/D input using a 5-V reference, the analog input resolution is 1 bit, which is $1/256$ of 5 V or 0.0193 V.

Defuzzification The process of translating output grades to analog output values.

Fuzzification The process of translating analog input values to input memberships or labels.

Fuzzy logic Software design based upon a reasoning model rather than fixed mathematical algorithms. A fuzzy logic design allows the system engineer to participate in the software design because the fuzzy language is linguistic and built upon easy-to-comprehend fundamentals.

Inference engine The internal software program that produces output values through fuzzy rules for given input values. The inference process involves three steps: fuzzification, rule evaluation, and defuzzification.

Input memberships The input signal or sensor range is divided into degrees of membership, i.e., low, medium, high or cold, cool, comfortable, warm, hot. Each of these membership labels is assigned numerical values or grades.

Output memberships The output signal is divided into grades such as off, slow, medium, fast, and full-on. Numerical values are assigned to each grade. Grades can be either singleton (one value) or Mandani (a range of values per grade).

Rule evaluation Output values are computed per the input memberships and their relationship to the output memberships. The number of rules is usually set by the total number of input memberships and the total number of output memberships. The rules consist of IF inputvar A is x, AND inputvar B is y, THEN outvar is z.

Semicustom MCU An MCU (microcontroller unit) that incorporates normal MCU elements plus user-specified peripheral devices such as higher-power port outputs, special timer units, etc. Mixed semiconductor technologies, such as high-density CMOS (HCMOS) and bipolar analog, are available in a semicustom MCU. Generally, HCMOS is limited to 10 V, whereas bipolar-analog is usable to 60 V.

BIBLIOGRAPHY

- Bannatyne, R., "Fuzzy logic—A new approach to embedded control solutions," Motorola Semiconductor Design Concept, DC410, 1992.
- Catherwood, M., "Designing for electromagnetic compatibility (EMC) with HCMOS microcontrollers," Motorola Semiconductor Application Note, AN1050, 1989.

- Chaudhuri, et al., "Speed control integrated into the powertrain computer," *New Trends in Electronic Management and Driveline Controls*, SAE SP-653, 1986, pp. 65-72.
- Hosaka, T., et al., "Vehicle control system and method therefore," U.S. Patent 4809175, May 29, 1990.
- Hosaka, T., et al., "Vehicle control system," U.S. Patent 4930084, Feb. 28, 1989.
- Mamdani, E. H., "Application of fuzzy logic to approximate reasoning using linguistic synthesis," *IEEE Transactions on Computers*, C-26-12, 1977, pp. 1182-1191.
- Ribbens, W., "Vehicle Motion Control," *Understanding Automotive Electronics*, 4th ed., 1992, pp. 247-257.
- Takahashi, Hiochi, "Automatic speed control device using self-tuning fuzzy logic," *IEEE Workshop on Automotive Applications of Electronics*, 88THO321, 1988, pp. 65-71.
- Self, Kevin, "Designing with fuzzy logic," *IEEE Spectrum*, Nov. 1990, pp. 42-44, 105.
- Sibigtroth, J., "Implementing fuzzy expert rules in hardware," *AI Expert*, April 1992.
- Stefanides, E. J., "Cruise control components packaged as one unit," *Design News*, Oct. 1, 1990, pp. 162-163.
- Zadeh, L. A., "Fuzzy sets, information and control," vol. 8, 1965, pp. 338-353.

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CHAPTER 15

BRAKING CONTROL

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15.1 INTRODUCTION

This chapter describes braking by first examining vehicle braking fundamentals, including the tire-to-road interface, vehicle dynamics, and conventional brake system components, and progressing to antilock systems objectives, components, safety considerations, control logic, and testing. The chapter concludes with a discussion of future vehicle braking systems.

For simplicity and because of applicability to the majority of automotive vehicles on the road, hydraulic brake systems as used on two-axle, nonarticulated vehicles will be discussed exclusively; this type of brake system is used on passenger cars, light trucks, and, in North America, on medium trucks.

15.2 VEHICLE BRAKING FUNDAMENTALS

Essential to the understanding of the technology associated with modern automotive vehicle braking is knowledge of the tire-to-road interface, vehicle dynamics during braking, and the components of a brake system. This section discusses these subjects to a system level of understanding.

15.2.1 Tire-to-Road Interface

The braking force generated at each wheel of a vehicle during a braking maneuver is a function of the normal force on the wheel and the coefficient of friction between the tire and the road. The simplified relationship between the weight on a wheel and the resulting frictional (braking) force is shown in Eq. (15.1).

$$F_x = \mu W_{wh} \quad (15.1)$$

where F_x = friction force $\times$ direction
 μ = coefficient of friction, tire-to-road
 W_{wh} = static and dynamic weight on the wheel

The tire-to-road coefficient of friction is not a constant but is a function of factors, most prominent being type of road surface and the relative longitudinal slip between the tire and the road. General curves relating coefficient of friction to wheel slip on various surfaces are shown in Fig. 15.1. From this figure and Eq. (15.1), the following observations are evident:

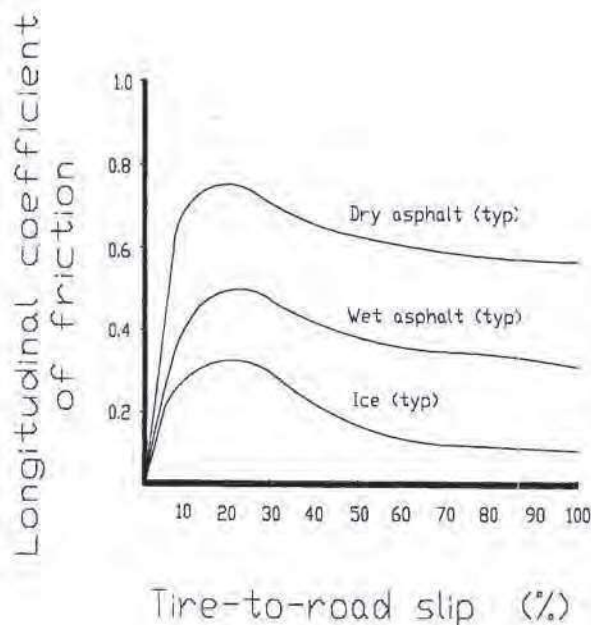


FIGURE 15.1 Longitudinal coefficient of friction as a function of wheel slip.

- The generation of frictional forces depends on wheel slip. If the tire is rolling at the same tangential velocity as the road surface, there is no longitudinal (braking) force. This relationship is fundamental in understanding braking and is not easily observed: wheel slip other than near 100 percent (no rotational wheel speed) is difficult to discern without instrumentation.
- The peak frictional (braking) force occurs under conditions of relatively little slip. This indicates that a hard apply of the brakes which causes a 100 percent slip typically does not produce the most braking force and an evenly modulated, controlled brake pressure applied by a skilled driver or through antilock control tends to produce shorter stops on most surfaces.
- The frictional (braking) force generated varies widely with road surface. The result of this relationship is obvious to both drivers and passengers in terms of stopping distance and deceleration if dry asphalt braking is compared with braking on ice.
- Typically, beyond the peak coefficient of friction attainable on a given road surface, the slope of the curve becomes negative. This phenomenon (essentially indicating that, beyond the slip resulting in peak frictional force, more pedal force results in less braking) explains why a skilled driver can attain significantly shorter stopping distances than can a less experienced driver and why electronic vehicle braking control is as complicated as it is. Also, the amount of "peak" in the coefficient of friction curves varies widely with road surface. More braking force benefit can be gained through slip control on surfaces such as ice than on dry asphalt, for example.

Another characteristic of automotive tires important in braking is lateral force versus slip. Lateral force is the force keeping a tire from sliding in a direction normal to the direction of the vehicle. The equation for lateral force is as follows:

$$F_y = \mu_{\text{lateral}} W_{wh}$$

where F_y = friction force, by direction
 μ_{lateral} = lateral coefficient of friction, tire-to-road

The lateral coefficient of friction drops off quickly once a wheel begins to slip longitudinally, as can happen during braking. Excessive wheel slip at the rear wheels of a vehicle and the resulting loss of lateral frictional force will contribute to instability as the rear of the vehicle tends to slide sideways with relatively small lateral forces on the vehicle. Excessive wheel slip and the resulting loss of lateral friction force on the front wheels of a vehicle will contribute to loss of steerability; this loss of steering phenomenon is common during panic stops on low coefficient surfaces such as ice, as a hard apply of the brakes puts the tires in a 100 percent slip situation.

15.2.2 Vehicle Dynamics During Braking

An equation for braking performance can be obtained from Newton's second law: the sum of the external forces acting on a body in a given direction is equal to the product of its mass and the acceleration in that direction. Relating this law to straight-line vehicle braking, the significant factors are shown in Eq. (15.2) and the sum of the forces acting on the vehicle is shown in Fig. 15.2.¹

$$\Sigma F = Ma_x = \frac{+W}{g} D_x = +F_{xf} + F_{xr} + D_A + W \sin \Theta + f_r W \cos \Theta \quad (15.2)$$

where M = mass of the vehicle
 a_x = linear acceleration in the x direction
 W = weight of the vehicle
 g = acceleration due to gravity
 $D_x = -a_x$ = linear deceleration
 F_{xf} = front axle braking force
 F_{xr} = rear axle braking force
 D_A = aerodynamic drag (considered to be acting at a point)
 Θ = angle of roadway
 f_r = rolling resistance coefficient = $(R_{xf} + R_{xr}) / W \cos \Theta$

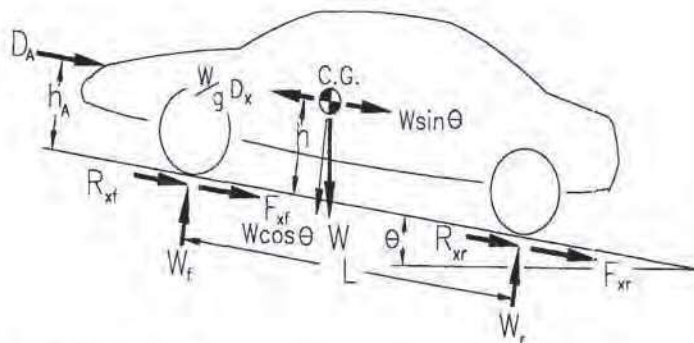


FIGURE 15.2 Significant forces' action on a vehicle during braking.

If braking forces are held constant and the vehicle velocity effects on aerodynamic drag and rolling resistance are neglected, the time for a vehicle velocity change, Eq. (15.3), and the distance traveled during a velocity change, Eq. (15.4), can also be derived from Newton's second law.²

$$t = \frac{M}{F_{xt}} (V_o - V_f) \quad (15.3)$$

where F_{xt} = total of all longitudinal deceleration forces on the vehicle
 t = time
 V_o = initial velocity
 and V_f = final velocity

$$x = \frac{M}{F_{xt}} \left(\frac{V_o^2}{2} - \frac{V_f^2}{2} \right) \quad (15.4)$$

where x = distance in forward direction

These approximations indicate that the time to stop is proportional to vehicle velocity and the stopping distance is proportional to the square of the vehicle velocity.

During braking, the dynamic load transfer that occurs is a function of the height of the center of gravity, the weight of the vehicle, the wheelbase, and the deceleration rate. Equation 15.5 describes this dynamic load shift.

$$W_d = \left(\frac{h}{L} \right) \left(\frac{W}{g} \right) D_x - \frac{h_A}{L} D_A \quad (15.5)$$

where W_d = dynamic weight
 h = center of gravity height
 L = wheelbase
 W = static vehicle weight
 g = acceleration due to gravity
 D_x = deceleration in the forward direction
 h_A = height of the aerodynamic drag

Considering two-axle vehicles, this load transfer is additive to the front wheels and subtractive to the rear wheels during braking, as shown in Eq. (15.6) and (15.7), respectively.

$$F_{xmf} = \mu_p \left(W_{fs} + \frac{hWD_x}{Lg} - \frac{h_A}{L} D_A \right) \quad (15.6)$$

where F_{xmf} = maximum friction force in the longitudinal direction on the front wheels
 μ_p = peak coefficient of friction
 W_{fs} = static weight on the front wheels

$$F_{xmr} = \mu \left(W_{rs} - \frac{hWD_x}{Lg} + \frac{h_A}{L} D_A \right) \quad (15.7)$$

where F_{xmr} = maximum friction force in the longitudinal direction on the rear wheels
 W_{rs} = static weight on the rear wheels

Simplifying Eq. (15.2) for the case of $\Theta = 0^\circ$ and negligible aerodynamic drag and rolling resistance yields the following:

$$\sum F = \frac{W}{g} D_x = +F_{xf} + F_{xr}$$

Solving for D_x and substituting in simplified Eqs. (15.6) and (15.7) yields Eq. (15.8) and (15.9), respectively:

$$F_{xmf} = \mu \frac{\left(W_{fr} + \frac{hF_{xmr}}{L} \right)}{1 - \mu_p \frac{h}{L}} \quad (15.8)$$

$$F_{xmr} = \mu \frac{\left(W_{rf} + \frac{hF_{xmf}}{L} \right)}{1 - \mu_p \frac{h}{L}} \quad (15.9)$$

These relationships indicate that the maximum braking force on the front wheels is dependent on the braking force on the rear wheels through the deceleration and the associated forward load transfer and, in a similar fashion, the braking force on the rear wheels is dependent on the braking force on the front wheels.

Through application of the preceding equations, brake systems designers can determine the total braking force required to achieve the desired deceleration, and the brake system components can be sized appropriately. Safety and legal requirements dictate that system designers consider deceleration under vehicle loaded and unloaded conditions as well as under partially failed brake system conditions (either half-system failures or loss of brake boost to the entire system). Because of these considerations and numerous others, such as desired customer pedal stroke and pedal force/deceleration expectations, vehicle brake system sizing is a complicated engineering effort usually accomplished with the aid of a vehicle simulator computer program.

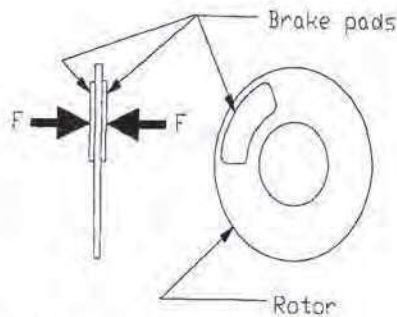


FIGURE 15.3 Disk brake schematic.

15.2.3 Brake System Components

Disk Brakes. Figure 15.3 shows a schematic diagram of a disc brake. In this type of brake, force is applied equally to both sides of a rotor and braking action is achieved through the frictional action of inboard and outboard brake pads against the rotor. The pads are contained within a caliper (not shown), as is the wheel cylinder. Although not a high-gain type of brake, disk brakes have the advantage of providing relatively linear braking with lower susceptibility to fading than drum brakes.

Force applied to the rotor by the pads is a function of hydraulic pressure in the brake system and the area of the wheel cylinder (or cylinders, as the design dictates). Static brake torque can be calculated using the following equation:

$$T = PAER \quad (15.10)$$

where T = brake torque

P = application pressure

A = wheel cylinder area

E = effectiveness factor: ratio of the disk rubbing surface to the input force on the shoes

R = brake radius

The static brake force can be calculated with the following relationship:

$$F_b = \frac{T}{r}$$

where F_b = brake force
 r = tire rolling radius

Drum Brakes. Figure 15.4 depicts a schematic diagram of a drum brake. In drum brakes, force is applied to a pair of brake shoes in a variety of configurations, including leading/trailing shoe (simplex), duo-duplex, and duo-servo. Drum brakes feature high gains relative to disk brakes, but some configurations tend to be more nonlinear and sensitive to fading and other brake lining coefficient-of-friction changes.

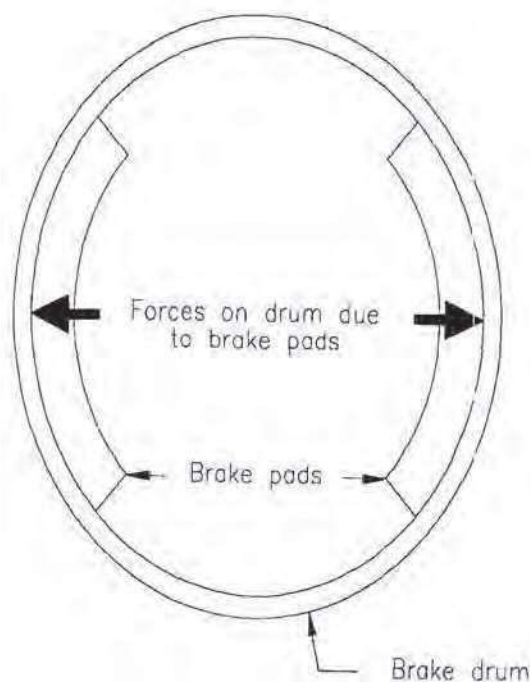


FIGURE 15.4 Drum brake schematic.

The static brake torque equation previously presented for disk brakes, Eq. (15.10), is equally applicable to drum brakes with design-specific changes for drum brake radius and effectiveness factor. By design, the brake radius for a drum brake is one-half the drum diameter. The effectiveness factor represents the major functional difference between drum and disk brakes; the geometry of drum brakes may allow a moment to be produced by the friction force on the shoe in such a manner as to rotate it against the drum and increase the friction force developed. This action can yield a mechanical advantage that significantly increases the gain of the brake and the effectiveness factor as compared with disk brakes. The dynamic brake force calculation for drum and disk brakes is more complex since the brake lining coefficient of friction is a function of temperature; as the lining heats during a braking maneuver, the effective coefficient of friction increases and less pressure is needed to maintain a constant brake torque.

Booster and Master Cylinder. Figure 15.5 is a schematic of a brake pedal, a vacuum booster, and a master cylinder. In actual practice, in passenger cars and light trucks the mechanical force gain due to the brake pedal geometry is usually 3 to 4 and the gain through a vacuum

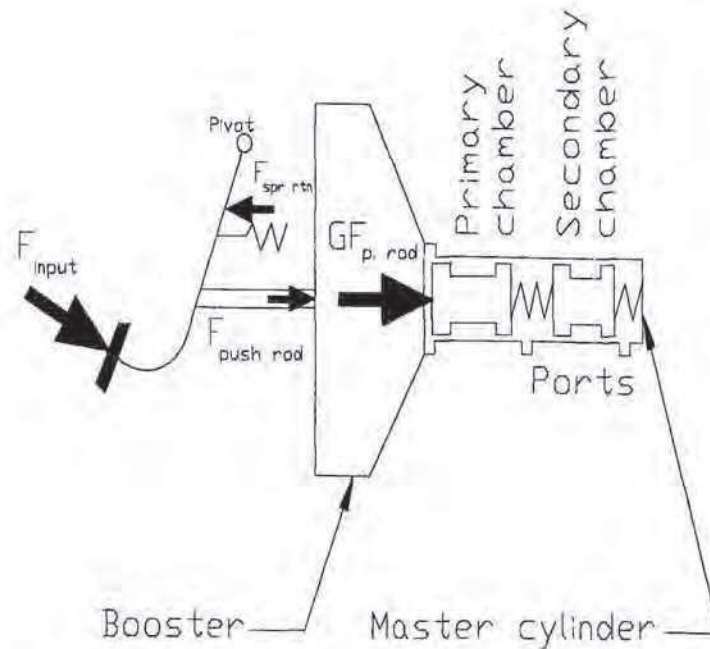


FIGURE 15.5 Brake pedal, vacuum booster, and master cylinder schematics.

booster is typically 5 to 9 after the booster reaches its crack point and before runout occurs. Therefore, force applied by the operator typically will be multiplied by a factor of 12 to 36 at the master cylinder in order to achieve the pressure necessary for braking. The resulting pressure in the master cylinder is as follows:

$$P_{MC} = \eta \frac{(F_{op} G_{mech} G_{boost} - F_s)}{A_{piston}}$$

where η = mechanical efficiency

P_{MC} = master cylinder pressure

F_{op} = operator force on the brake pedal

G_{mech} = mechanical gain primarily related to the brake pedal assembly geometry and the instantaneous return spring force

G_{boost} = brake booster gain, a function with the nonlinearities of a minimum crack force being necessary to initiate boost and a runout phenomenon resulting in a decreased force gain after a given input force is applied

F_s = return spring force

A_{piston} = area in the master cylinder on which the force is acting (chamber piston area)

Master cylinders are separated into primary and secondary chambers to improve safety by avoiding total brake system loss in case of a failure in one portion of the system. The most common configuration is shown in Fig. 15.5 with two chambers in a single bore.

Proportioning Valve. Due to the dynamic weight shift, as shown in Eq. (15.5), brake pressures that are appropriate for high-deceleration braking on front wheels usually are too high for the rear wheels; the result is that the rear wheels will tend to lock during braking. This problem can be decreased significantly through the use of proportioning valves. Standard proportioning valves allow equal front and rear brake pressures during low input pressures (corresponding to low deceleration rates and little dynamic load shift) but decrease the gain through the valve to less than one when a fixed input pressure (crack pressure) is reached. More sophisticated load-sensing valves are used in some applications when necessary, such as

when dynamic load shifts and vehicle loading changes are wide enough to make a fixed proportioning valve insufficient for proper braking in all conditions. Load-sensing valves feature a means to measure the weight on the rear wheels and adjust the gain through the valve accordingly.

Figure 15.6 shows the two most common passenger car and light truck system schematics including proportioning valves. The vertically split brake system typically is used on rear-wheel-drive vehicles and the diagonally split system is typically used on front-wheel-drive vehicles.

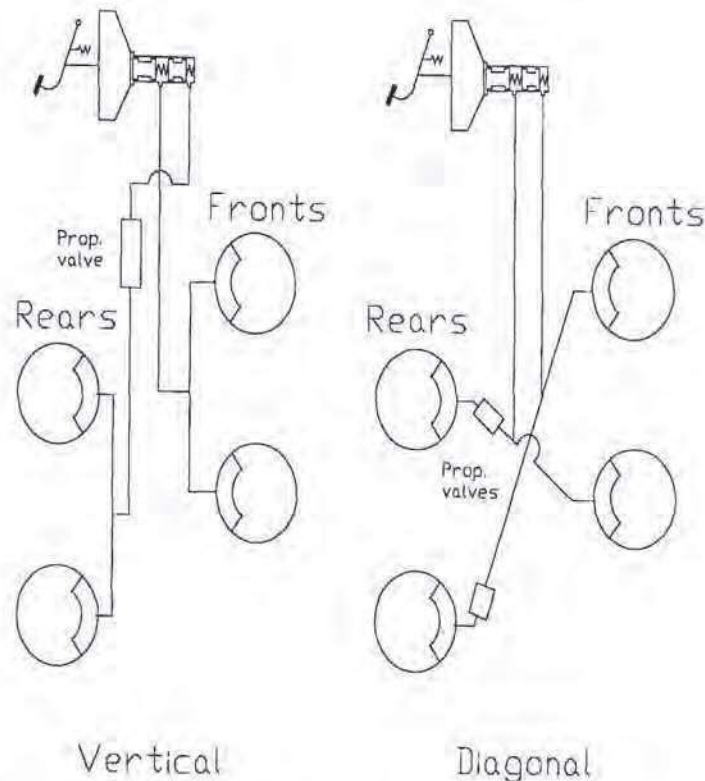


FIGURE 15.6 Vertical and diagonal split brake systems schematics.

Widespread use of diagonally split systems has been a direct result of the popularity of front-wheel-drive vehicles. Current law requires a half-system (hydraulic) failure stopping rate that is difficult to meet if the half system is the rear brakes (on a vertically split system) and the vehicle weight is significantly shifted towards the front as it is in front-wheel-drive vehicles. Diagonally split systems afford the use of one front brake regardless of the half-system failure, and front-wheel-drive vehicles can be made to pass the legal requirements despite the typically large difference between the weight on the front and on the rear wheels. However, diagonally split systems require two proportioning valves and tend to require more complex hydraulic plumbing than do vertically split systems.

15.3 ANTILOCK SYSTEMS

Although antilock concepts have been known for decades, widespread use of antilock (also called antiskid and ABS) began in the 1980s with systems developed with digital microprocessors/microcontrollers replacing the earlier analog units. An antilock system consists of a

hydraulic modulator and hydraulic power source that may or may not be integrated with the system master cylinder and booster, wheel speed sensors, and an electronic control unit. The fundamental function of an antilock system is to prohibit wheel lock by sensing impending wheel lock and taking action through the hydraulic modulator to reduce the brake pressure in the wheel sufficiently to bring the wheel speed back to the slip level range necessary for near-optimum braking performance.

15.3.1 Objectives

The objectives of antilock systems are threefold: to reduce stopping distances, to improve stability, and to improve steerability during braking.

Stopping Distance. As shown in Eq. (15.4), the distance to stop ($V_f = 0$) is a function of the initial velocity, the mass of the vehicle, and the braking force. From this equation it can be seen that by maximizing the braking force the stopping distance will be minimized, all other factors remaining constant. From Fig. 15.1 it is evident that on all types of surfaces, to a greater or lesser extent, there exists a peak frictional force. It follows that by keeping all of the wheels of a vehicle near the peak, an antilock system can attain maximum frictional force and, therefore, minimum stopping distance. This is an objective of antilock systems; however, it is tempered by the need for vehicle stability and steerability.

Stability. Although decelerating and stopping vehicles constitutes a fundamental purpose of braking systems, maximum friction force may not be desirable in all cases. For example, if a vehicle is on a split-coefficient surface, (asphalt and ice, for example), such that significantly more braking force is obtainable on one side of the vehicle than on the other side, applying maximum braking force on both sides will result in a yaw moment that will tend to pull the vehicle to the high-coefficient side and contribute to vehicle instability. Typically, on short-wheelbase vehicles a control strategy is employed to control the pressure in the rear wheels together to improve stability; similarly, it is common for a front-wheel strategy to be employed to limit the initial side-to-side pressure difference so as to not induce excessive moment changes in the steering wheel and force the operator to make excessive steering corrections to counteract the yaw moment.

If an antilock system can keep the vehicle wheels near the peak frictional force range, then lateral force is reasonably high, though not maximized. This contributes to stability and is an objective of antilock systems.

Steerability. Steerability depends on high lateral force. Good peak frictional force control is necessary in order to achieve satisfactory lateral force and, therefore, satisfactory steerability. Steerability while braking is important not only for minor course corrections but also for the possibility of steering around an obstacle. Antilock systems provide this feature through control to the peak frictional force range.

15.3.2 Antilock Components

The components of an antilock system are the wheel speed sensors, the hydraulic modulator, the hydraulic power source (usually an electric motor/pump), and the electronic control unit.

Wheel Speed Sensors. Due to simplicity and proven reliability, variable reluctance wheel speed sensors typically are used in antilock systems. Used in conjunction with exciter rings, this type of sensor produces a sinusoidal output that is directly proportional in frequency and amplitude to the angular velocity of the sensed wheel.

Depending on the design of the sensor and exciter ring and the gap between them, the sensor output amplitude may be as low as 100 mV at very low vehicle speeds and over 100 V at high vehicle speeds.

Both single-pole and dual-pole variable reluctance sensors are used, depending on the application: single-pole sensors tend to have higher outputs and dual-pole sensors tend to have better immunity to some types of noise. A limitation of this technology is that the very low speed output tends to be too low to be sensed reliably by the electronic control unit, given the electrically noisy environment typical of vehicles. This can result in errors below 1 to 3 m/h and cumulative inaccuracies if this sensor is used in conjunction with an odometer function; normally, antilock function is inhibited at very low speeds. Both single-ended and balanced inputs are used in electronic control units to receive wheel speed signals. A variety of active sensor technologies, including Hall effect and magnetoresistive, can be used in applications requiring very low speed sensing and in applications in which an appropriate signal level cannot be achieved with conventional variable reluctance sensors.

Hydraulic Modulators. Hydraulic modulators typically take two forms in production antilock systems: solenoid valves and electric motors. A simplified solenoid valve system schematic is shown in Fig. 15.7. In this system, if the solenoid valves are de-energized, hydraulic fluid is free to flow between the master cylinder and the brakes. If too much pressure is presented to the brakes and wheel lock is imminent, the antilock system will actuate a solenoid valve and energize the hydraulic pump. Actuation of the solenoid valve allows pressure to decrease from the brake through the valve to a low-pressure accumulator/sump. The fluid is temporarily stored in the sump prior to being pumped back into the system by the hydraulic pump. Through repetitive energization/de-energization cycles, average pressure to a given wheel can be regulated to the level necessary to achieve the desired braking force. Typical brake pressure and resulting wheel speed cycling is shown in Fig. 15.8.

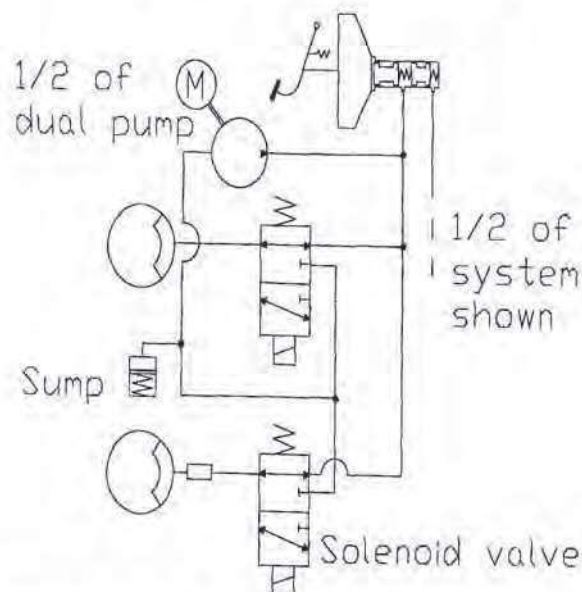


FIGURE 15.7 Simplified solenoid valve antilock system schematic.

Electric Motor/Pump. Although some antilock systems use multiple electric motors driving pistons to provide multiple-channel pressure reduction and rebuild, usually an electric motor-driven pump is used in conjunction with solenoid valves to achieve individual brake or brake channel pressure reduction and rebuild. A dual pump is often used to maintain a complete hydraulic separation of the two channels of the brake system. This is done to ensure that failure in one channel of the brake system will not affect operation of the other channel.

Electronic Control Unit. Control of the hydraulic modulator and electric motor/pump is performed by the electronic control unit. Modern customer expectations coupled with

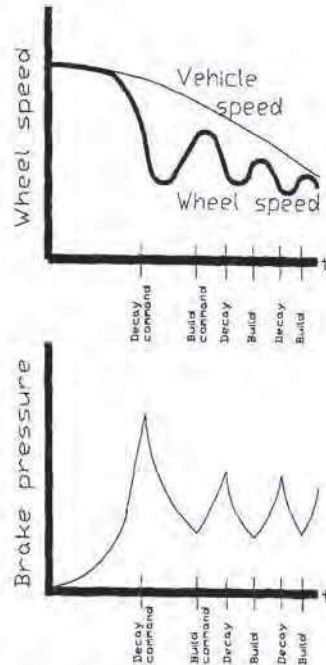


FIGURE 15.8 Typical antilock brake cycling.

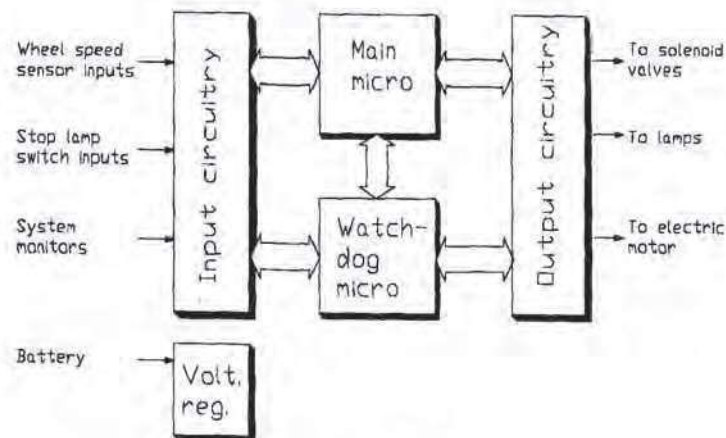


FIGURE 15.9 Electronic control unit block diagram.

decreasing microcontroller costs have made microcontroller-based electronic control units the norm rather than the exception. Although the control units can be either engine compartment-mounted or passenger compartment-mounted, reduced wiring costs favors the former. Also, for enhanced reliability, electronic control units may be either attached to or integrated with the hydraulic modulator.

15.3.3 Safety Considerations

Standard automotive brake systems have been developed and refined over the years to be highly reliable and safe. Because of its ability to decrease pressure in brakes, an antilock system must be designed using a disciplined methodology and must be rigorously tested prior to release for production.

Failure Mode and Effects Analyses/Fault Tree Analyses. Failure mode and effects analyses (or fault tree analyses) are essential to the proper design of antilock systems. Both system-level and subsystem-level analyses need to be performed and fault effects and detection assumptions must be tested. No single failure can result in an unsafe condition and, if a fault is undetectable in the field, that fault in conjunction with any other fault must not result in an unsafe condition. Because of the complexity of the electronic control units, simulation techniques are used to test those fault effects in which bench or field testing is impractical.

Common Design Techniques to Improve Safety. One of the most common techniques used to improve safety in antilock systems is to include extensive built-in-test within the electronic control unit. Typically, all inputs to the electronic control unit and outputs to the other components of the antilock system are tested for proper signals and loads, respectively, and all functions internal to the electronic control unit are extensively tested.

In addition, redundant processing is commonly used to insure the proper internal working of a microcontroller. This may take the form either of identical microcontrollers or of a main and a watchdog microcontroller that can inhibit operation.

In order to ensure inhibition of faulty antilock operation, antilock systems employ a relay function to remove actuation power from the output actuators; this function may take the form of a discrete relay or it may be a transistorized circuit. This relay function is a key element of the design since it affords a secondary method in which to inhibit energization of valves or the motor/pump and, therefore, a second level of safety relative to improper antilock operation.

Figure 15.9 is a typical electronic control unit block diagram.³ Inputs are filtered and buffered prior to being presented to the microcontrollers for processing. Likewise, the microcontroller outputs are buffered/amplified and filtered prior to exiting the electronic control unit. In the diagram shown, the main microcontroller is responsible for the majority of processing and control of the outputs; the watchdog microcontroller, as its name implies, is responsible for monitoring for proper operation and inhibiting antilock if faults are indicated. A characteristic of modern antilock electronic control units is bidirectional communication between functional blocks; this is a result of the high level of built-in-test designed into the control units. For example, the output circuitry may be commanded to test the solenoid valves for proper current draw and convey the test results to the microcontrollers; similarly, the input circuitry may be commanded to perform tests on the sensors and other antilock components external to the electronic control unit, and convey the test results to the microcontrollers.

15.3.4 Antilock Control Logic Fundamentals

Due to the complexity of antilock braking and the requirements of stability and steerability as well as good stopping distance, the brake control algorithm is more easily represented as a state-space diagram than as a classical proportional-integrative-derivative control scheme.

A simplified state diagram for a single-channel antilock system is shown in Fig. 15.10. In this diagram, a vehicle not braking or decelerating would be in the NORMAL BRAKING state. If antilock action is warranted, it is because the brake pressure on a given channel has caused the wheel to begin to lock; the first action would be to decrease the brake pressure (DECAY state) in an effort to permit the locking wheel to reaccelerate. Fine control of the brake pressure is indicated by the states labeled HOLD OR BUILD/DECAY and SLOW BUILD and course control is indicated by the FAST BUILD state. (The course control is typically used during rapidly changing road surface conditions such as ice-to-asphalt transitions.) During the antilock cycle the state will change, as needed, to attain the type of brake pressure and resulting wheel speed activity as shown in Fig. 15.8. Once the need for antilock action has ended, the END ANTILOCK state is entered, the pump motor is de-energized, the valves are de-energized, and the system can return to the original NORMAL BRAKING state.⁴

How this state-space approach is integrated into a typical microcontroller flowchart is shown in Fig. 15.11. After RESET and INITIALIZATION, a microcontroller enters into a

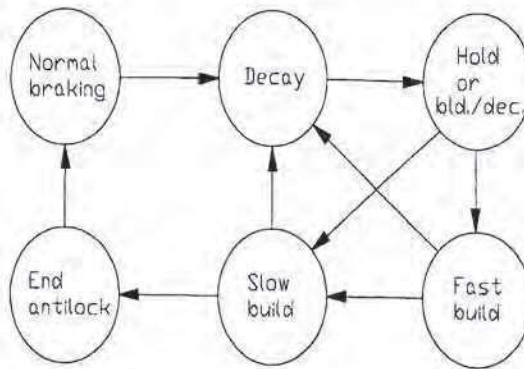


FIGURE 15.10 Simplified single-channel state diagram.

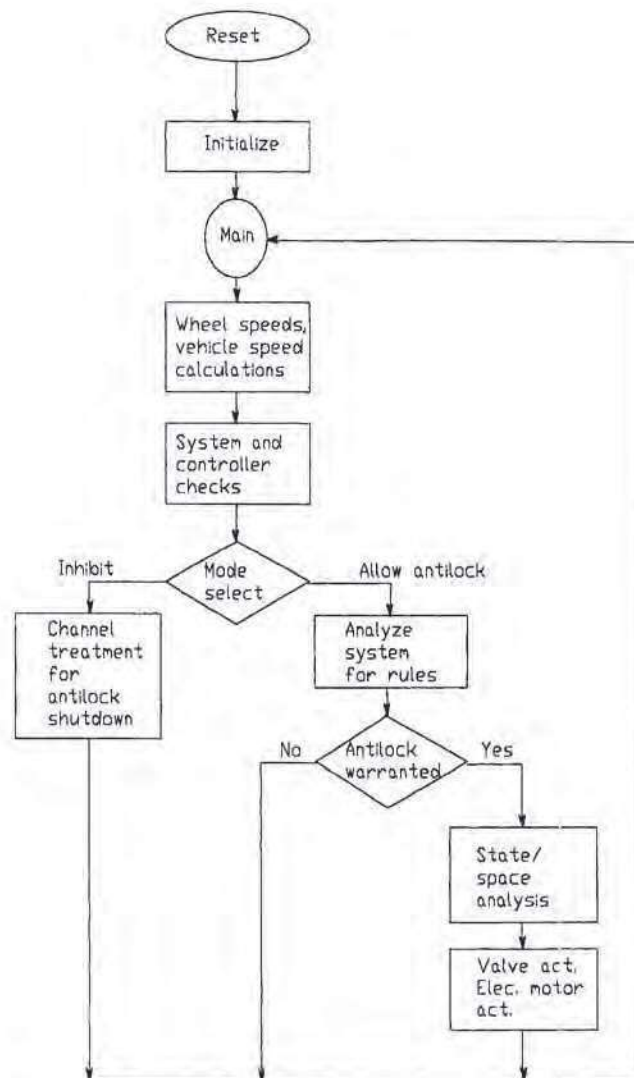


FIGURE 15.11 Simplified antilock flowchart.

MAIN loop that includes extensive system and electronic control unit checks as well as calculations of wheel speeds, prediction of vehicle speed, analysis of conditions warranting antilock action/state-space control law, and valve and motor/pump actuations.

Calculation of wheel speeds consists of scaling the wheel speed sensor inputs to a more usable form and possibly filtering noise due to axle deflection, brake squeal, other electrical systems, etc. A consideration is that the bandwidth of wheel acceleration and deceleration is large—50 g's may be attainable.

The vehicle velocity prediction is critical to many control schemes because wheel velocity relative to vehicle velocity, as well as wheel slip, may be used as a factor in determining appropriate valve action. Vehicle velocity prediction becomes difficult once the wheels begin to lock because the sensors will no longer be reliable indicators of vehicle speed. The methods used to predict vehicle velocity once the wheels have begun to lock consist of a set of rules that have been developed by antilock manufacturers through years of experience to ensure a prediction that has a high degree of accuracy to true vehicle speed.

The antilock system checks typically consist of sensor and valve/motor continuity tests and system voltage range tests. In addition, the checks normally include tests internal to the electronic control unit, such as inter-microcontroller communication.

Once it is determined that conditions are such that antilock action can be safely invoked if warranted, the wheel speed conditions are analyzed to establish the appropriate state for that channel. Primary indicators for most antilock control schemes are wheel slip and wheel deceleration. Another factor considered is the effect on vehicle stability if a particular state is commanded.

Actuation of the valves or electric motor actuators is a direct result of the decisions made in the analysis/state-space logic. Other than actuators requiring pulse-width modulation drives, the actuators normally will remain in the commanded state until the microcontroller loops back through the code (usually a few milliseconds).

15.3.5 Antilock System Testing

Antilock vehicle testing has evolved over the years to include the following most common tests:

- Straight-line stopping
- Braking in a turn
- Split coefficient stopping with associated stability criteria
- Transitional road surface testing including checkerboard and low/high and high/low coefficient surfaces
- Lane change maneuver

All of these tests may be performed on a variety of surfaces, at a variety of speeds, and with lightly loaded and heavily loaded vehicles.

15.4 FUTURE VEHICLE BRAKING SYSTEMS

A trend that will impact braking systems is the industry's desire to reduce vehicle wiring through the use of multiplexing techniques. As increasing numbers of vehicles are outfitted with antilock, this trend is expected to result in an increased number of antilock systems communicating with other vehicle systems through a multiplex link. In addition to the wheel speed/vehicle velocity information available from the antilock system, the antilock electronic control unit could benefit from this technology by being able to receive engine, transmission, steering angle, and other subsystem information.

Another trend in advanced electronically controlled braking systems is vehicle dynamics control during nonbraking maneuvers, as well as during braking. This is accomplished through use of the traction control actuators normally integrated in antilock hydraulic modulators, the addition of sensors to more accurately determine the dynamic state of the vehicle, and communication links with the drivetrain electronic controllers. Vehicle dynamic control holds the promise of safer vehicle operation through improved stability in all maneuvers.

The vehicle brake systems engineering community also is investigating the addition of radar to individual vehicles. This addition could lead to semiautomatic or automatic braking in emergency situations as the brake system anticipates the potential problem and aids the operator in safely applying the vehicle brakes in time to avoid a collision. This concept also lends itself to automatic braking in nonemergency situations to maintain safe distances between vehicles at high speeds.

Continuing interest in electric vehicles and the need for regenerative braking in these vehicles likely will significantly impact future braking systems. It is expected that the regenerative braking function will not be sufficient to provide adequate braking deceleration under all conditions and to provide operators the comfort and safety obtainable with conventional friction brake systems augmented by antilock systems. It is expected that a more complex electronic control system will be used in conjunction with electric vehicles to afford optimum power regeneration without sacrificing braking stopping distance, stability, or steerability.

These trends point to a continued use of friction brake systems through the end of the century and a significant expansion of the role of electronics in these systems.

GLOSSARY

Antilock (or ABS or Antiskid) A system designed to prevent wheel lock during overbraking.

Antilock hydraulic modulator A hydraulic brake pressure modulation actuator used in antilock systems.

Booster A brake pedal force amplifier, typically vacuum or hydraulically powered.

Booster crack point The brake pedal/push rod travel point initiating booster force amplification.

Booster runout A condition in which the brake booster can no longer provide the gain required due to high input forces and the input force/output force slope becomes less positive.

Brake caliper A part of a disc brake that contains the brake pads and the brake cylinder.

Braking force A force tending to stop a moving vehicle. Usually applied to the force resulting from brake torque being applied to a wheel of a moving vehicle.

Braking maneuver Any vehicle braking action intended to decelerate a moving vehicle, including partial as well as full stops.

Diagonal split brake system A brake system configuration in which a front brake and its opposing rear brake are included in the same brake channel. This technique is used to allow braking on one front wheel in the case of catastrophic failure of the other brake channel.

Disk brake A type of brake characterized by force being applied to both sides of a rotor, thereby creating braking torque.

Drum brake A type of brake characterized by brake force being applied to the inner surface of a drum, thereby creating braking torque.

Dynamic load transfer The characteristic of weight shift during deceleration that places more weight on the front wheels and reduces weight on the rear wheels.

Electric motor/pump The typical hydraulic power source used in antilock systems; an electric motor driving a hydraulic pump.

Lateral force Force perpendicular to the direction of travel.

Longitudinal slip Relative slip between the wheels and the road surface in the direction of travel.

Master cylinder A two-chambered hydraulic cylinder operated by the driver through actuation of the brake pedal.

Proportioning valve A hydraulic valve designed to reduce pressure to the rear brakes relative to the front brakes once a crack point is reached. The valve may be fixed or load-sensing.

Regenerative braking A type of braking used in electric vehicles in which the drive motor is used as a generator during braking, and it serves as the load to brake the vehicle. This technique is used to reclaim a portion of the energy expended during vehicle motion.

Vertical split brake system A brake system configuration in which both front brakes are on one channel and both rear brakes are on the other channel.

Wheel slip The difference between tangential wheel speed and road speed. A rolling tire with no braking torque on it exhibits 0 percent slip; a nonrotating tire on a moving vehicle exhibits 100 percent slip.

REFERENCES

1. J. Y. Wong, *Theory of Ground Vehicles*, John Wiley & Sons, New York, 1978.
2. T. D. Gillespie, *Fundamentals of Vehicle Dynamics*, Society of Automotive Engineers, Inc., Warrendale, Pa., 1992.
3. J. L. Cage, "The Bendix ABS underhood electronic control unit," *Automotive Technology International '90*, Sterling Publications International Limited, London, 1990.
4. J. L. Cage, and J. T. Hargenrader, "Safe electronic modules for antilock applications," *Automotive Technology International '91*, Sterling Publications International Limited, London, 1991.

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CHAPTER 16

TRACTION CONTROL

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16.1 INTRODUCTION

Traction control systems designed to prevent the drive wheels from spinning in response to application of excess throttle have been on the market since 1987. Vehicles with powerful engines are particularly susceptible to drive-wheel slip under acceleration from standstill and/or on low-traction road surfaces. The results include attenuated steering response on front-wheel-drive (FWD) vehicles, diminished vehicle stability on rear-wheel-drive (RWD) cars, and loss of effective accelerative force.

Large mutual discrepancies in left- and right-side traction levels engender early drive-wheel slip on slick surfaces. Under these conditions, the effective accelerative forces at both drive wheels are limited to a level corresponding to the adhesion available at the low-traction side. The traction control system inhibits wheelspin, allowing the wheel on the high-traction surface to apply maximum accelerative force to the road.

16.1.1 Optimizing Stability (Steering Control)

The essential requirement for systems designed to optimize vehicle stability (with RWD) and steering control (with FWD) is to maintain adequate lateral traction. The most basic arrangements achieve this end by controlling engine torque alone. Both drive wheels transmit the same level of motive force, dosed in accordance with the adhesion available at the low-traction wheel and thereby providing particularly large lateral-traction reserves at the wheel with the greater adhesion. When the traction levels are roughly equal at both drive wheels, the system enhances vehicle stability (steering control) while providing a certain increase in available effective accelerative force beyond that available on an uncontrolled vehicle with slipping wheels.

16.1.2 Optimizing Traction

The optimization of traction becomes a top priority when the motive force must be transmitted to surfaces on which the adhesion varies substantially between sides.

The typical passenger car features a differential unit at the drive axle; this unit allows virtually loss-free differences in wheel speed (for instance, in corners) in combination with uniform torque distribution to the drive wheels. This layout generally provides favorable dynamic vehicle response, as the equal distribution of drive torque inhibits vehicle yaw. However, a difference in the force-transmission potentials at the drive wheels can combine with demands for maximum traction to expose basic liabilities in the design principle.

Figure 16.1 illustrates the system dynamics of drive shaft, differential, and drive wheels on road surfaces affording differing levels of traction with adhesion coefficients μ_H , μ_L (high wheel, low wheel). The torque emanating from the driveshaft is distributed equally between the drive wheels. The low wheel responds to inadequate adhesion potential by spinning during brief wheel acceleration. The accelerative force transmitted through the high wheel then corresponds to the sum of the accelerative force at the low wheel plus its inertia $\Theta_R [\dot{\Omega}]/R$. Once the low wheel reaches its terminal speed, the accelerative force available at both wheels is limited to the maximum at the low wheel.

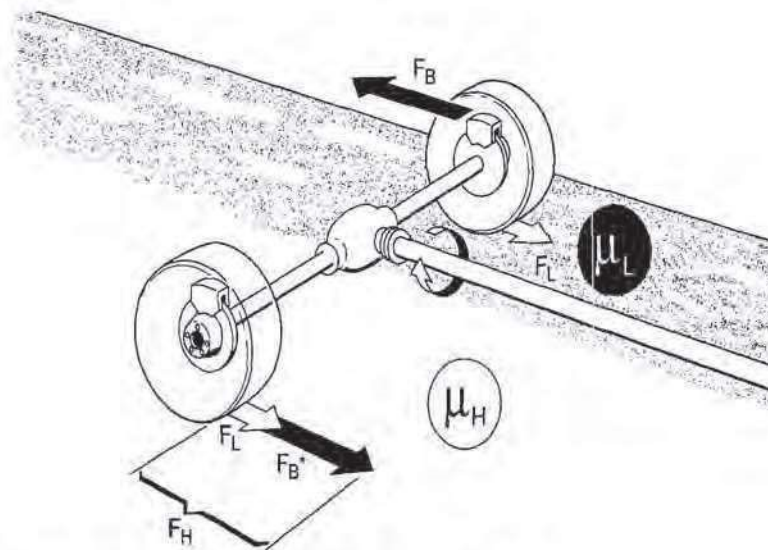


FIGURE 16.1 Braking intervention to limit differential slip.

The only way to increase the accelerative force at the high wheel is to prevent the low wheel from spinning. The first option, application of the wheel brake, is illustrated in Fig. 16.1. The application of braking force F_B at the low wheel prevents it from spinning. This makes the additional accelerative force F_B^* (the product of F_B multiplied by the ratio of effective braking radius to wheel radius) available at the high wheel.

A second option for maximum exploitation of traction potential is represented by the application of fixed, variable, or controlled differential-slip limitation mechanisms. These provide fixed coupling to ensure equal slippage rates at the drive wheels, thereby allowing them to develop maximum accelerative force.

During cornering at high rates of lateral acceleration, lateral variations in drive-wheel load occur, again producing a difference in acceleration potential. Brakes and limited-slip differential arrangements can also be applied to assist in ensuring maximum traction under these conditions.

16.1.3 Optimizing Stability and Traction

Traction-control systems incorporating engine-torque control and supplementary braking intervention (or controlled differentials) can be applied simultaneously to ensure consistent

vehicle stability (steering control) and optimal acceleration within the limits imposed by physical constraints. Engine-torque control is the preferred method on road surfaces affording uniform adhesion, while application of braking force (or differential control) provides optimal acceleration at both drive wheels for dealing with surfaces displaying lateral variations in traction.

16.2 FORCES AFFECTING WHEEL TRACTION: FUNDAMENTAL CONCEPTS

The dynamic forces that define the tires' braking response on straights and during cornering are already familiar from the technical literature. The transmission of accelerative force in straight-line operation and in curves is subject to the same qualitative principles that apply during braking. The slip ratio which applies for braking,

$$\lambda_B = \frac{V_F - \Theta_R R}{V_F}$$

is replaced by the ratio

$$\lambda_A = \frac{\Theta_R r - V_F}{V_F} \quad \text{with } \Theta_R r \geq V_F$$

Acceleration slip rates can range all the way from 0 to the very high numbers used to describe the conditions that can occur when the drive wheels spin freely during attempts to accelerate from rest.

Figures 16.2 to 16.4 show acceleration and side-force coefficients as a function of the acceleration slip. Figure 16.2 applies for acceleration during straight-line operation. The demand for reserves in lateral adhesion is fairly diminutive under these conditions (including, for instance, compensation for side winds); thus, traction remains the salient factor.

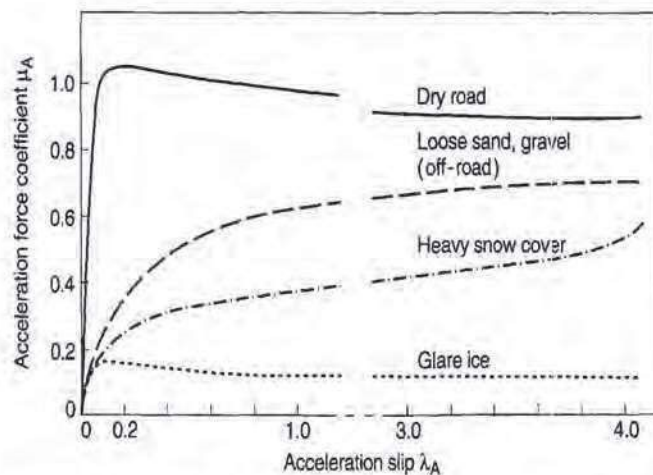


FIGURE 16.2 Adhesion coefficient for acceleration μ_A as a function of acceleration slip λ_A .

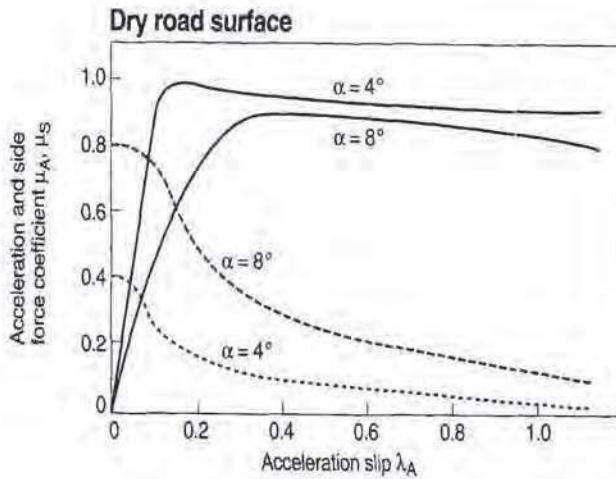


FIGURE 16.3 Acceleration and lateral traction coefficients at different slip angles α .

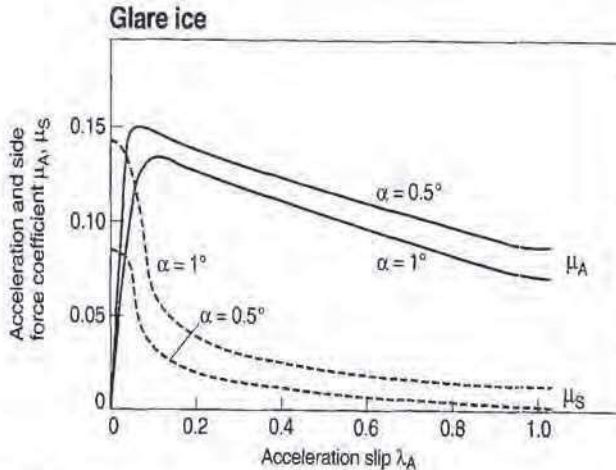


FIGURE 16.4 Acceleration and side-force coefficients at different slip angles α .

On dry road surfaces, maximum accelerative force is available at slip rates of 10 to 30 percent, with traction enhancements of 5 to 10 percent possible relative to spinning drive wheels.

On glare ice, maximum traction is achieved at extremely diminutive acceleration slip levels (2 to 5 percent). On loose sand and gravel and in deep snow (especially in combination with snow chains), the coefficient of acceleration force increases continually along with the slip rate, with the respective maxima only being reached somewhere beyond 60 percent. Thus, the slip rates of 2 to 20 percent found within the ASR's operating range will not provide adequate traction under all operating conditions.

For this reason, all known ASR systems incorporate slip-threshold switches or ASR deactivation switches, which allow the vehicle operator to either reset the ASR slip-control threshold to substantially higher levels, or to switch the system off entirely should the need arise.

Figures 16.3 and 16.4 apply to acceleration during cornering; under these conditions the drive wheels are subject to various degrees of lateral force as a function of the vehicle's rate of lateral acceleration. Increasing acceleration slip (and increasing accelerative forces) cause a drop in the lateral forces, which then respond to still higher slip rates by collapsing to small residual levels.

Figure 16.3 represents the response pattern on a dry road surface. The curve starts at a rate of acceleration slip of zero. Initially, the side-force coefficient displays a moderate downward trend. However, continuing increases in the coefficient of acceleration force induce a substantial fall in the side-force coefficient. The figure shows that the accelerative force must be limited to a fraction of its ultimate potential if sufficient lateral forces are to be maintained.

On glare ice (Fig. 16.4), the extremely limited friction potential means that vehicle stability under acceleration remains available only at relatively small slip angles (ca. $\leq 2^\circ$). Relatively diminutive slip angles ($\leq 0.05^\circ$) will be sufficient to induce a radical drop in the side-force coefficient. This makes it clear that an extremely precise and sensitive slip control is required on glare ice (and other low-friction road surfaces). The traction-control system must thus exhibit a high degree of monitoring accuracy, while signal processing and actuation of the final-control elements must be rapid and precise.

16.3 CONTROLLED VARIABLES

The four wheel speeds used for the ABS supply the following closed-loop control parameters for the ASR traction control system: the acceleration slip from the lateral variation in the rotation speeds of the driven and nondriven wheels, and the angular acceleration of the driven wheels.

The following secondary control parameters are also calculated: vehicle velocity and acceleration based on the speeds of the nondriven wheels, and curve recognition, derived from comparisons of the speeds of the nondriven wheels.

The target value for acceleration slip is defined as the mean rotational velocity of the nondriven wheels plus a specified speed difference known as the slip threshold setpoint. The main goal of regulating acceleration slip can thus be divided into two subsidiary objectives: closed-loop control of acceleration slip to maintain slip rates at the specified levels with maximum precision, and calculation of optimal slip setpoints for different operating conditions and their implementation as control objectives.

Depending on the final-control strategy being used, various control concepts can be employed to meet the first objective. With throttle-valve control, a setpoint calculated from a number of signals is adopted for regulation as soon as the closed-loop control enters operation. The subsequent control process basically corresponds to that of a PI controller. When the brakes are used, arrangements are necessary to compensate for the nonlinear pressure-volume curve which governs the response in the brake calipers. The first stage of the closed-loop control program thus employs a sensing pulse corresponding to a relatively large volume; this compensates for compliance in the brake caliper. In the next stage, the system responds to positive deviations from the setpoint with graduated pressure increases; the rate of increase corresponds to the degree of divergence. A subsequent drop below the control setpoint initiates a pressure-relief stage (sequence of defined pressure-relief and holding phases). This impulse series, in which the length of the pressure-relief phases increases continually, is followed by termination of braking intervention.

Ignition and fuel-injection intervention essentially conform to the D controller closed-loop control concept. The difficulty associated with determining satisfactory setpoint values results from the fact that optimal acceleration and lateral forces cannot be achieved simultaneously. The ASR control algorithm must therefore meet varying operator demands for linear traction and lateral adhesion by using priority-control strategies and adaptive response patterns.

High vehicle speeds are accompanied by lower operator requirements for traction, especially with low coefficients of adhesion. At the same time, reductions in vehicle stability and steering response are not acceptable. The control strategy is thus designed to provide progressively lower slip threshold setpoints as the vehicle speed increases, with priority being shifted from linear traction to lateral adhesion.

The vehicle's acceleration rate and the regulated level of engine output provide the basis for reliable conclusions regarding the coefficient of friction. Thus, another important strategy takes into account the coefficient of friction at the road surface. The slip threshold setpoint is raised in response to higher friction coefficients. This ensures that an ASR system designed for optimum performance on low-friction surfaces will not intervene prematurely on high-traction surfaces.

Yet another important control strategy is based on the cornering detection mentioned previously. This system employs the difference in the wheel speeds of the nondriven wheels as a basis for reductions in the slip setpoint to enhance stability in curves. This speed differential can be used to calculate the vehicle's rate of lateral acceleration. A large discrepancy indicates a high rate of lateral acceleration, meaning that a high coefficient of friction may also be assumed. In this case, the slip setpoint should not be reduced, but rather increased.

16.4 CONTROL MODES

16.4.1 Modulation of Engine Torque

Various control intervention procedures can be employed, either singularly or in combination, to regulate engine torque:

- Throttle-valve control with the assistance of the electronic performance control or an automatic throttle-valve actuator (ADS)
- Adjustment of the ignition-advance angle
- Selective ignition cutout, combined with suppression of fuel injection
- Fuel injection suppression alone

Slippage at the drive wheels generally occurs in response to an excess of torque relative to the coefficient of friction available at the road surface. Controlled reduction of engine torque is thus a logical step. It is always the most suitable method in cases where virtually identical adhesion is present at both drive wheels. At the same time, the response times for the individual engine controls must be considered if adequate vehicle stability is to be ensured (see Fig. 16.5).

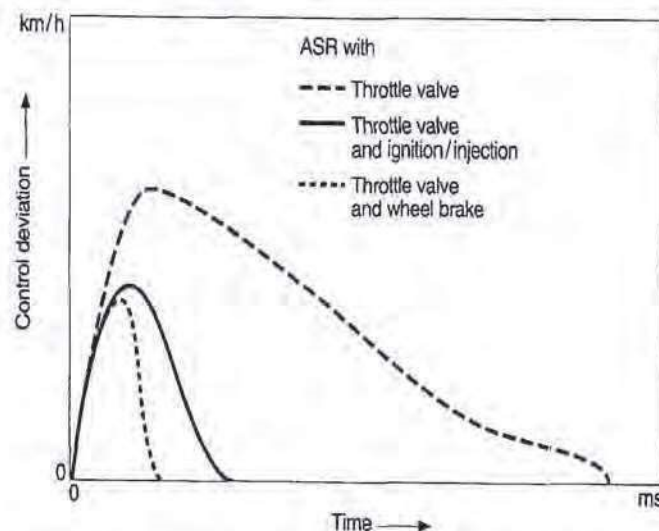


FIGURE 16.5 Deviation of the controlled variable during first control cycle with different actuators.

If control is restricted to the throttle-valve position alone, the throttle valve's response time, response delays within the intake tract, inertial forces in the engine, and drivetrain compliance will all result in palpable wheel slippage continuing for a relatively long period of time. Throttle regulation alone cannot ensure adequate vehicle stability on rear-wheel-drive vehicles. This qualification is particularly applicable to vehicles with a high power-to-weight ratio. On FWD and 4WD vehicles, throttle-valve control alone can be sufficiently effective if response delays are minimized.

Arrangements combining throttle-valve regulation with interruptions in the fuel injection produce substantial reductions in the amplitude and duration of wheel slip. Thus, this concept can be used to guarantee good vehicle stability regardless of which axles are driven.

In principle, it is also possible to design an ASR system based solely on regulation of the ignition and injection systems. This concept employs a system with sequential fuel injection. It alternately cuts out individual cylinders, while the ignition is also adjusted for the duration of the control process. Although this concept can be employed to ensure adequate vehicle stability regardless of drive configuration, certain sacrifices in comfort are unavoidable, especially during operation on ice and during the warm-up phase.

16.4.2 Brake Torque Control

The brakes at the drive wheels are capable of converting large amounts of kinetic energy into heat, at least for limited periods of time. In addition, the response times can be held extremely short, making it possible to limit slippage increases to very low levels.

ASR systems relying exclusively on braking intervention appear suitable for regulating spin at the drive wheels.

Traction enhancements during starts and under acceleration on road surfaces affording varying levels of adhesion at the left and right sides are especially significant with this system.

The ASR hydraulic unit used to generate the braking forces employs components which are already present for the ABS. Cost considerations make it important that ASR hydraulic systems require an absolute minimum in additional components beyond those already available for the ABS.

The hydraulic concepts can be classified in two categories, according to whether stored hydraulic energy is employed or not. A dual-strategy system including rapid braking intervention with stored hydraulic energy is always to be recommended where the engine torque control is based entirely on throttle valve adjustments with their relatively long response times.

Figures 16.6 and 16.7 illustrate two examples of ASR braking intervention using stored energy.

Brake Torque Control with Stored Energy. In this ASR system, designed for RWD vehicles in the upper price range, engine torque is regulated exclusively by the engine performance control (EPS) unit. Rapid braking intervention is required for enhanced regulation of vehicle stability; this also ensures optimal traction at both drive wheels, especially with lateral variations in adhesion potential. This system offers optimal vehicle stability and traction combined with a high level of comfort. Figure 16.6 illustrates the design of the integrated ABS/ASR hydraulic unit.

The primer pump draws brake fluid from the master-cylinder reservoir and supplies it to the ABS return pump under small positive pressure. To meet the requirements of the ABS system, the dual-circuit ABS return pump is expanded to include a third plunger, which assumes responsibility for the ASR storage charge. For "rapid" braking intervention, the pressure in the wheel cylinder must escalate from 0 to 50 bar in less than 200 ms. A hydropump, with its limited supply capacity, is not capable of providing this order of pressure build-up in the required time, necessitating the use of a high-pressure accumulator to supply the required brake fluid with adequate alacrity.

When the traction control system is activated, the switchover valve moves to the third position and brake fluid from the reservoir is supplied to the two ABS/ASR control valves. While

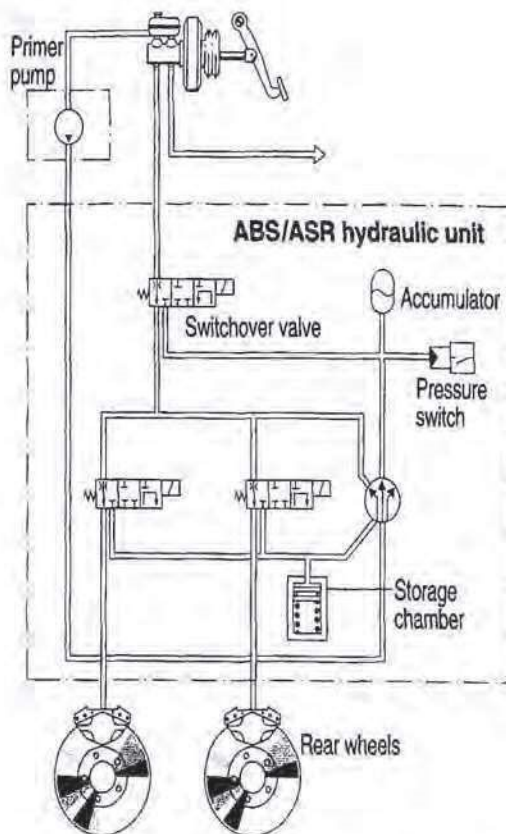


FIGURE 16.6 Diagram showing the ABS/ASR hydraulic circuit using braking force with stored energy (integrated system).

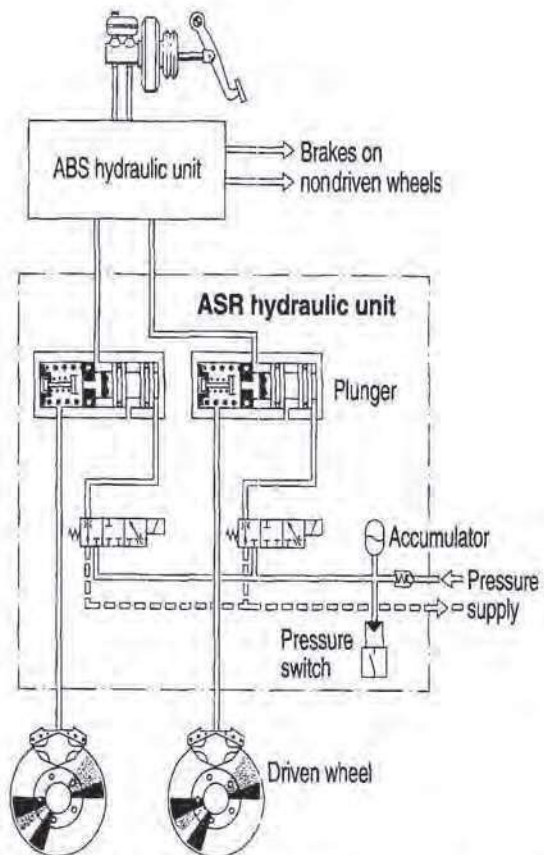


FIGURE 16.7 Diagram showing hydraulic unit for ASR system with braking intervention using stored energy via plunger.

an ABS solenoid valve can satisfy the requirements of the ABS hydraulic circuit, two solenoid valves are needed for the ABS/ASR hydraulic unit in order to allow individual braking control at the drive wheels.

These two solenoid valves regulate brake pressures at the driven wheels in accordance with the ASR control commands. Accumulator charging is controlled by the pressure switch and continues during normal vehicle operation. A special low-noise pump design is employed to meet stringent customer specifications.

ABS/ASR System with Brake Intervention Using Stored Energy (separate design). One version employs a high-pressure accumulator to provide hydraulic energy for ASR braking. An engine-driven pump (level control) supplies hydraulic fluid to the accumulator. The system includes a plunger for each driven-wheel brake. An additional 3/3 ABS valve is also installed to govern the supply of hydraulic fluid from the reservoir to the primary side of the plunger and to separate this medium from the brake fluid present on the secondary side. The solenoid valves respond to the position commands from the ABS/ASR control unit to control pressure accumulation, maintenance, and release operations to maintain the required pressure level on the plungers' primary sides. The plungers move to the left. Initially, a central valve closes to block the connection with the master cylinder. The plunger's displacement pressurizes the brake fluid on the secondary side to produce the desired pressure level in the wheel cylinder.

ASR designs relying on stored energy to activate plungers are always worthy of consideration in those cases where the vehicle is already equipped with a hydraulic energy supply for other purposes (e.g., power steering, level control).

Brake Torque Control without Stored Energy. Each of the ASR hydraulic units described requires a high-pressure accumulator to ensure that the braking energy can be provided quickly enough. This means additional design complication with attendant expenditure.

Another system differs from those already described by using the supply circuit of the ABS return pump exclusively to regulate the braking force at the drive wheels. The return pump forms part of a self-priming circuit, thus employing the (already installed) ABS return pump as an inexpensive source of energy for braking. The ASR braking function can thus be achieved with a minimum of additional design complication.

Figure 16.8 shows the hydraulic circuit for a passenger car with a K-pattern brake circuit and front-wheel-drive. When the intake valve opens, the self-priming pump extracts brake fluid from the reservoir and draws it into its circuit before supplying it directly to the ABS/ASR control valve at the drive wheel. The control valve regulates braking force by building up, maintaining, and releasing pressure in accordance with the respective position commands from the control unit. The fluid bled by the control valve in the pressure-release mode is returned to the pump's intake side. A switchover valve connects the high-pressure side of the self-priming return pump with the second brake circuit (governing the other front-wheel brake) and the pressure-relief valve responsible for regulating the ASR system pressure. With the exception of the suction line, the second brake circuit features a symmetrical layout. The passage to the master cylinder is also closed.

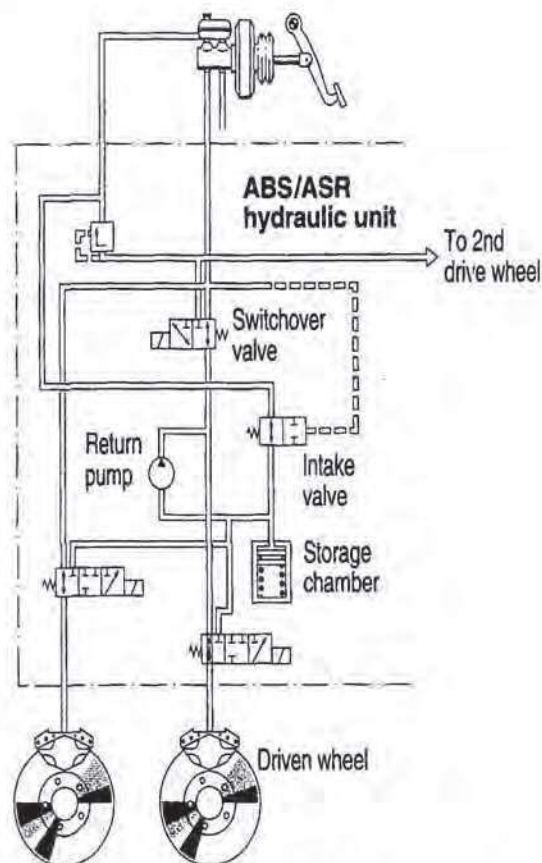


FIGURE 16.8 Diagram showing the ABS/ASR hydraulic unit employed for braking intervention without stored energy.

This ASR system can be installed in rear-wheel-drive vehicles with TT brake circuit configurations with even less expenditure: because both drive-wheel brakes are then in the same circuit, only a single switchover valve is required.

New Design for Brake Torque Control without Stored Energy. The principle of braking intervention without stored energy has been developed further for a new generation of ASR systems entering production in 1993. (See Fig. 16.9.)

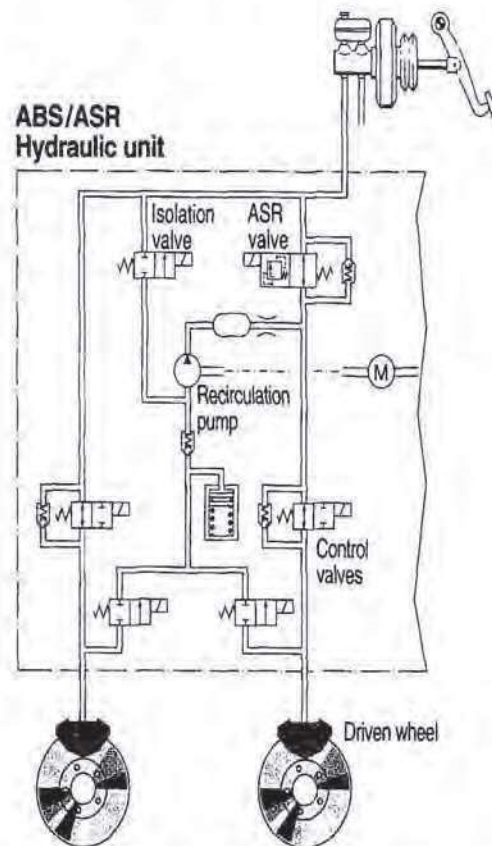


FIGURE 16.9 New design for brake torque control without stored energy.

The self-priming recirculation pump draws fluid from the master cylinder through an electrically controlled isolation valve, obviating the requirement for an additional intake line. Meanwhile, the pressure-relief valve can be integrated within the ASR valve. This configuration allows creation of a self-contained system combining simplified installation with enhanced safety. In addition, the reduction in the number of components improves reliability.

The 3/3 valves employed on earlier versions have been replaced by extremely small 2/2 solenoid valves.

16.4.3 Differential Slip Modulation

ABS/ASR System Regulating Differential Slip on Rear-Wheel-Drive Vehicles. This system controls the differential's lateral slip to improve traction for starting off and for simultaneous acceleration and cornering on road surfaces affording different levels of traction from left to right. The slip-limitation mode remains active until a specific vehicle speed is attained, and is deactivated completely at higher speeds.

When the vehicle starts off, the rotation speeds of the wheels on the outside of the curve are subjected to a mutual comparison. The lock is activated once a specific difference in the two speeds is exceeded. There then follows a comparison of the rotation speeds of the driven

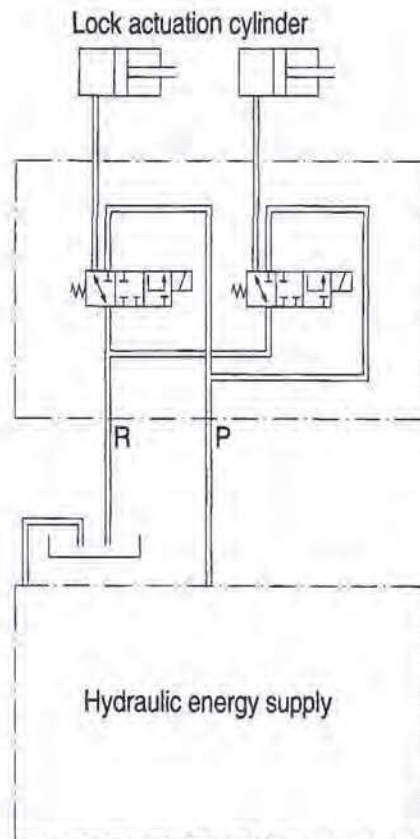


FIGURE 16.10 Hydraulic unit for control of rear-axle and interaxle differential locks.

wheels. The lock is deactivated as soon as the difference in rotation speeds drops below a specified level.

ABS/ASR System for Controlling Rear-Axle and Interaxle Differential Slip Rates on 4WD Passenger Cars. This version incorporates an additional interaxle control feature.

Four-wheel drive passenger cars employ a specific fixed front-to-rear distribution of engine output to provide optimum vehicle characteristics within the stable range (that is, with limited amounts of acceleration slip). One or both wheels at either axle can respond to throttle application on low-traction surfaces with immoderate wheelspin. This is where the interaxle slip limiter is activated to adapt the distribution of engine torque to the traction available at the respective axles, thereby improving traction while also enhancing vehicle stability and steering response.

Figure 16.10 illustrates the design of the hydraulic unit used to control the lateral and interaxle slip-limitation mechanisms. An electric pump supplies a high-pressure accumulator. The accumulator, in turn, provides pressure to a 3/3 solenoid valve for control of the lateral and interaxle locking mechanisms. The return volume from the pressure-relief phase is conducted to a separate reservoir.

16.5 TRACTION CONTROL COMPONENTS

The following is a selection of the components used in Bosch traction control systems.

16.5.1 Wheel-Speed Sensors

The system employs the same wheel-speed sensors that provide the information for the antilock braking system.

16.5.2 Electronic Control Unit

Figure 16.11 shows an ASR circuit diagram. An input amplifier IC receives the signals from the wheel-speed sensor; the signal frequency indicates the wheel speed. Two microcontrollers then process the signals to determine the wheel speed and acceleration rate. These data, in turn, provide the basis for calculations to determine the actual and desired values for slip control. Overall signal processing, the control algorithm, and the monitoring software are present in each of these microcontrollers to provide the system with backup capabilities.

Three output amplifiers control the solenoid valves, the ABS and ASR indicator lamps, the driver-information lamp, and the motor and valve relay. An additional IC is required to monitor the braking requirement and pump-motor voltage and for diagnosis.

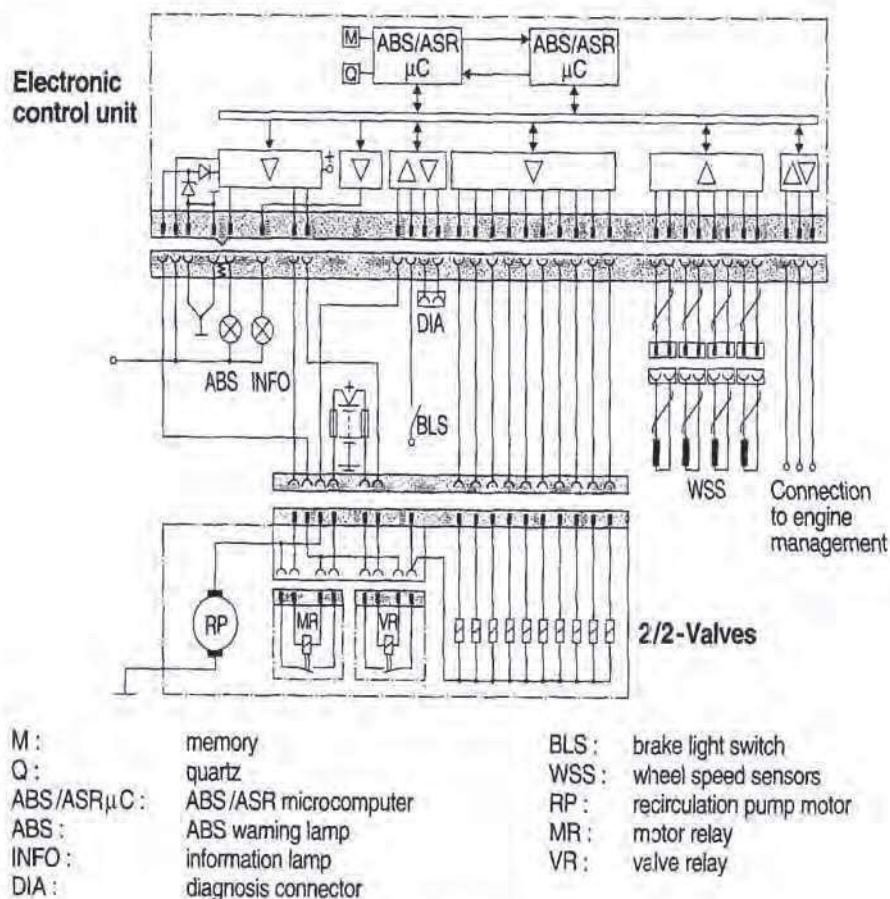


FIGURE 16.11 Operation diagram for the complete system.

16.5.3 Hydraulic Unit

Four examples of ASR hydraulic units were given in Sec. 16.4.2.

16.5.4 Electronic Throttle Control Actuator

See Sec. 10.3.1.

16.5.5 Simplified Throttle Control Actuator

See Sec. 10.3.1.

16.5.6 Fuel Injection and Ignition Control

This system reduces engine output by suppressing the fuel injection process.

Complete fuel injection suppression would lead to a total loss of engine output—a smooth, graduated response would be impossible with this kind of arrangement. In contrast, selective suppression of the injection process at individual cylinders can be employed to achieve a good compromise between quick response and a graduated reduction of engine power. This is the design principle behind the new concept.