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.

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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-

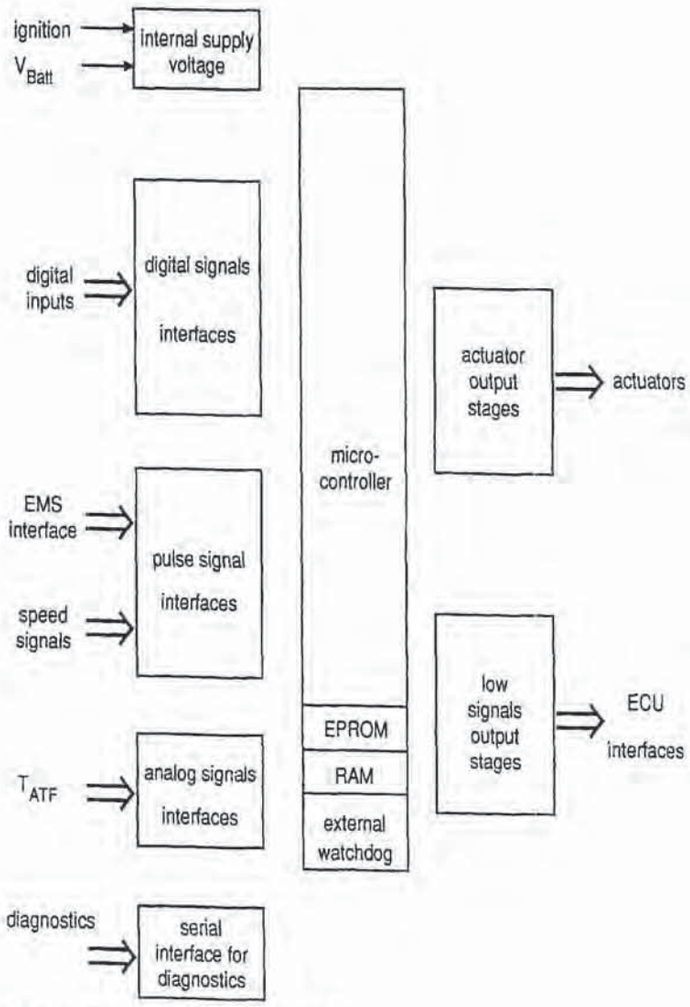


FIGURE 13.1 Overview of hardware parts.

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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.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

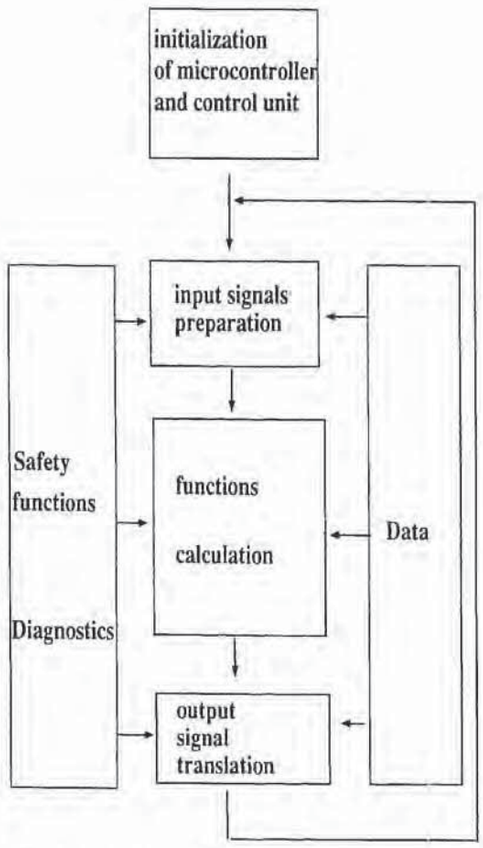


FIGURE 13.2 Software structure overview.

- 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.

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US PATENT NO. 6,557,540

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/s^2, 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 at 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

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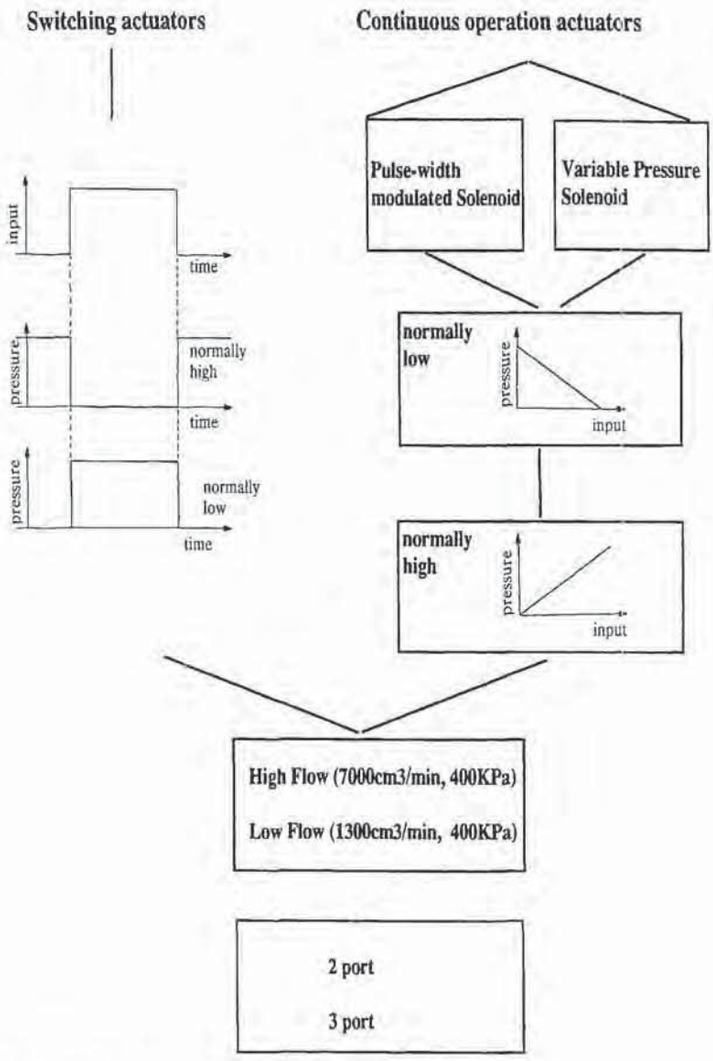


FIGURE 13.3 Electrohydraulic actuators for automatic transmissions.

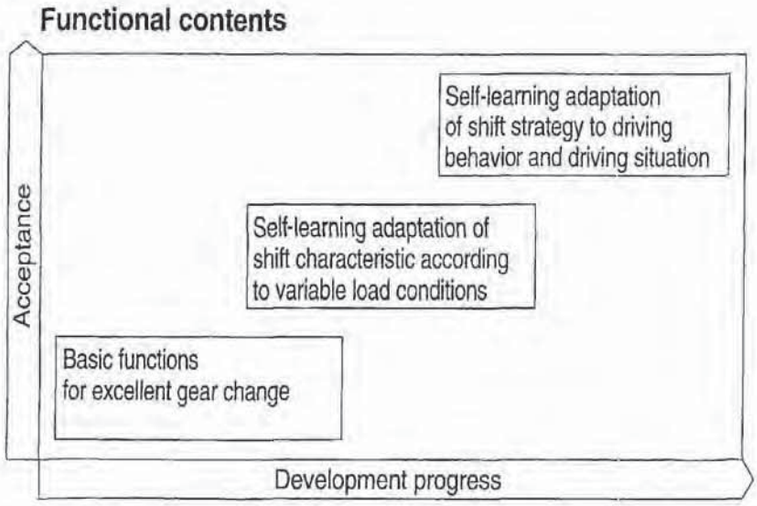


FIGURE 13.4 Relationship between driving characteristic and function complexity.

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US PATENT NO. 6,557,540

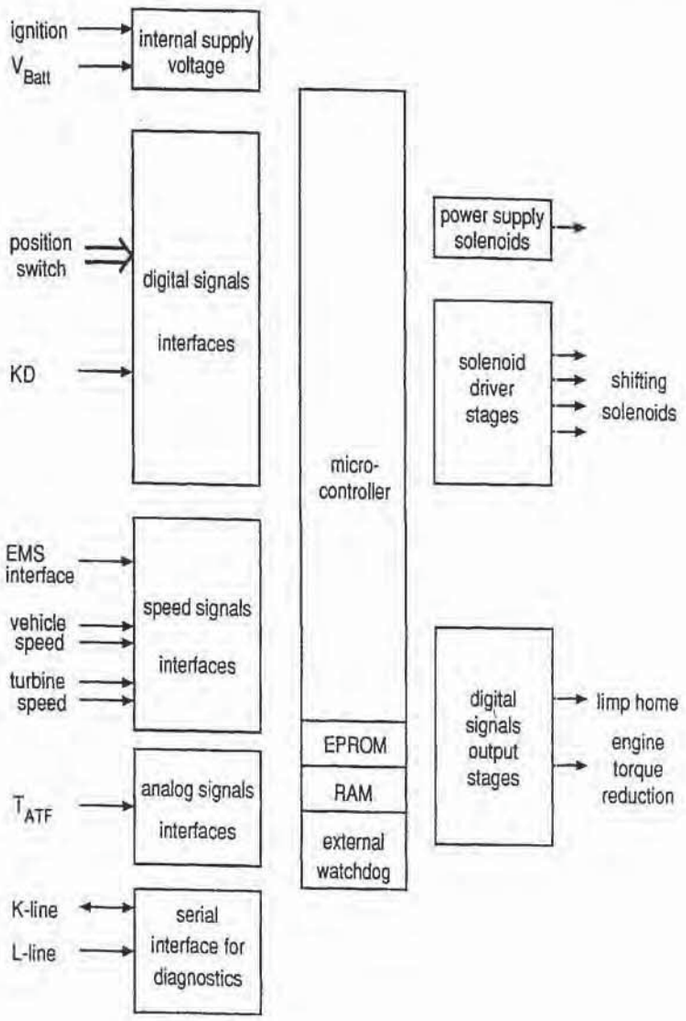


FIGURE 13.5 Structure of a basic transmission electronic control unit.

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

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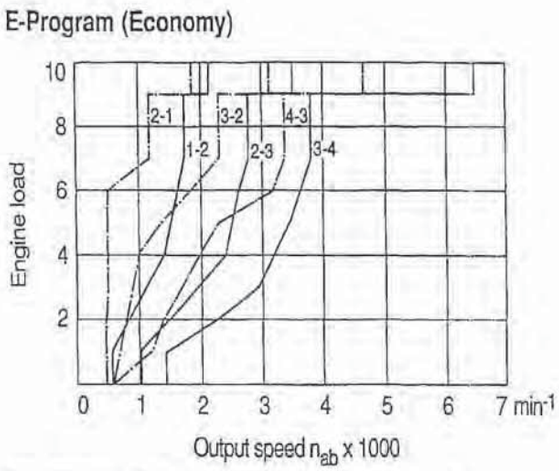


FIGURE 13.6 Shift characteristics of a four-speed application.

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

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US PATENT NO. 6,557,540

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

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

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.

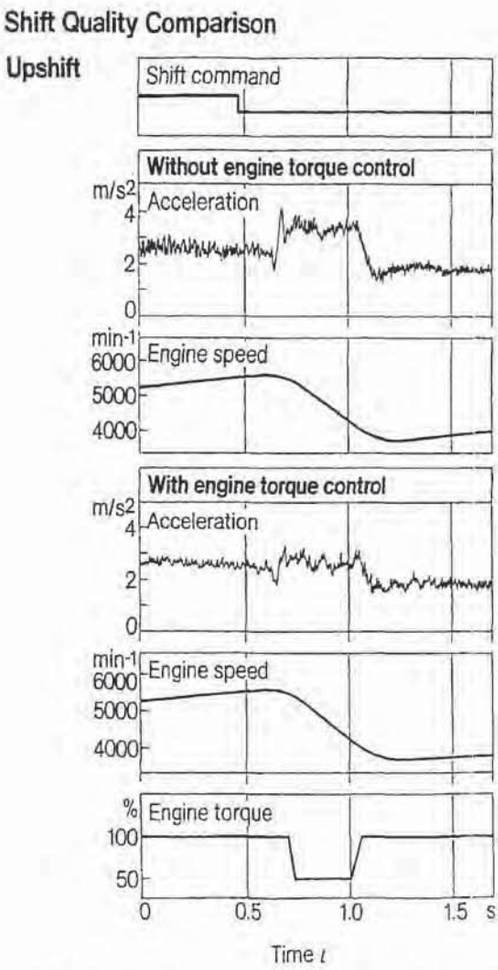


FIGURE 13.7 Timing of engine torque reduction during upshift.

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

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US PATENT NO. 6,557,540

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.

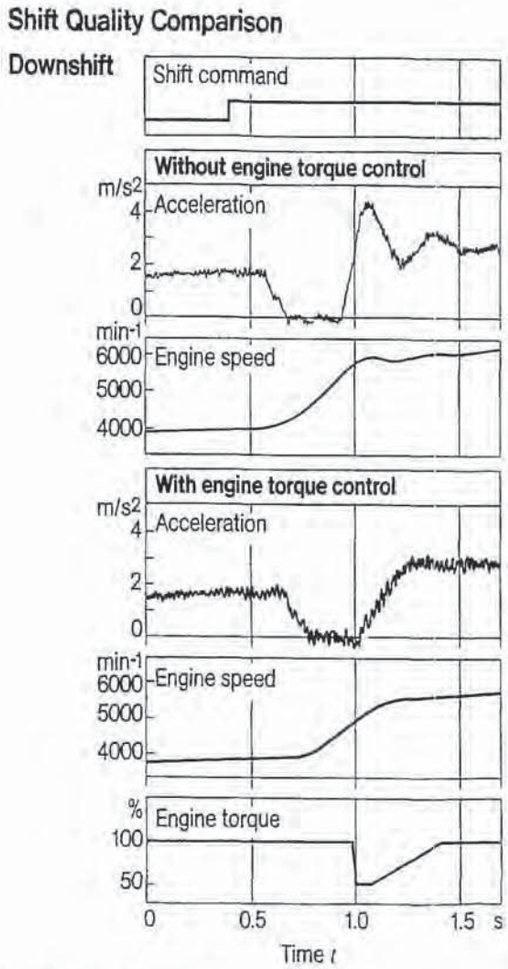


FIGURE 13.8 Timing of engine torque reduction during downshift.

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.

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US PATENT NO. 6,557,540

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

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 plausability 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

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US PATENT NO. 6,557,540

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

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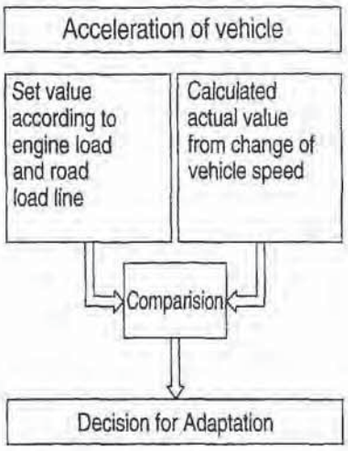


FIGURE 13.9 Basic principle of adaptive shift point control.

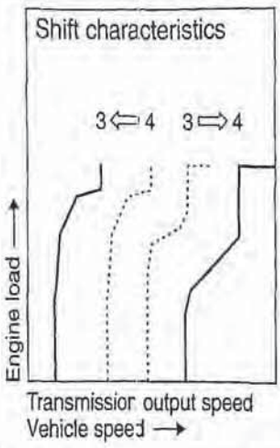


FIGURE 13.10 Shift characteristics before (- - -) and after (—) adaptation.

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-

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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 admissable 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 admissable 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-

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US PATENT NO. 6,557,540

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 registrated. 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

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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 Mastershift[10] 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.

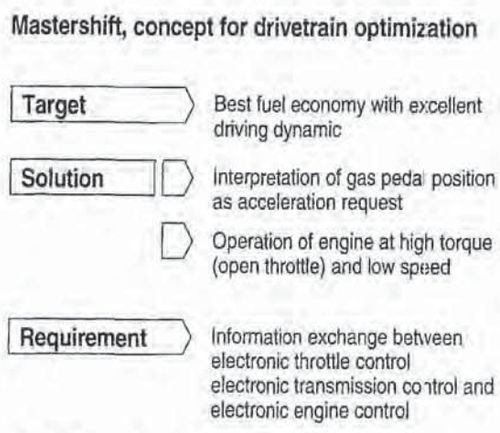


FIGURE 13.11 Drivetrain operation.

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US PATENT NO. 6,557,540

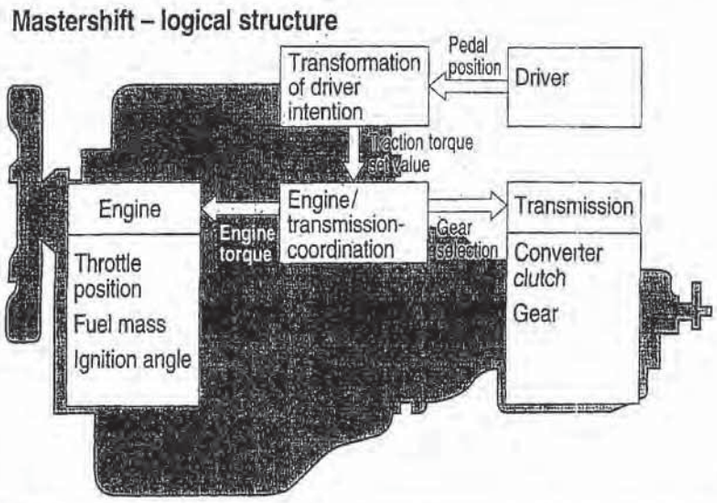


FIGURE 13.12 Mastershift: logical structure and communication between different control systems.

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.

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US PATENT NO. 6,557,540

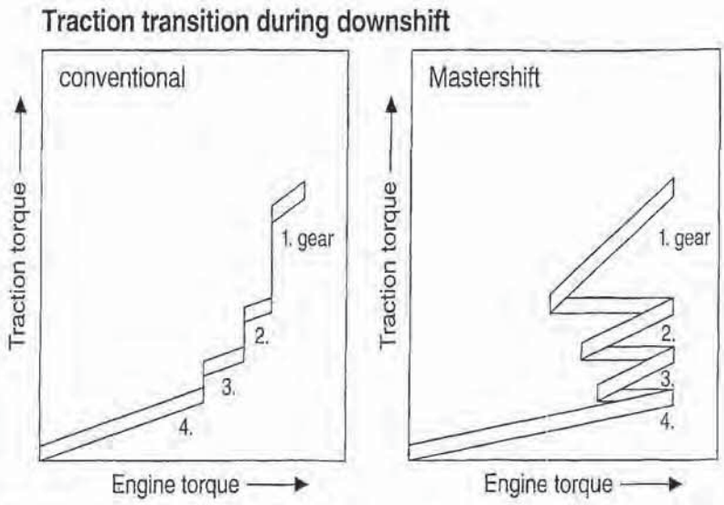


FIGURE 13.13 Constant traction torque by operation of throttle opening during gear shift.

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.

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PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

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US PATENT NO. 6,557,540

CHAPTER 14
CRUISE CONTROL

Richard Valentine
Motorola Inc.

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.

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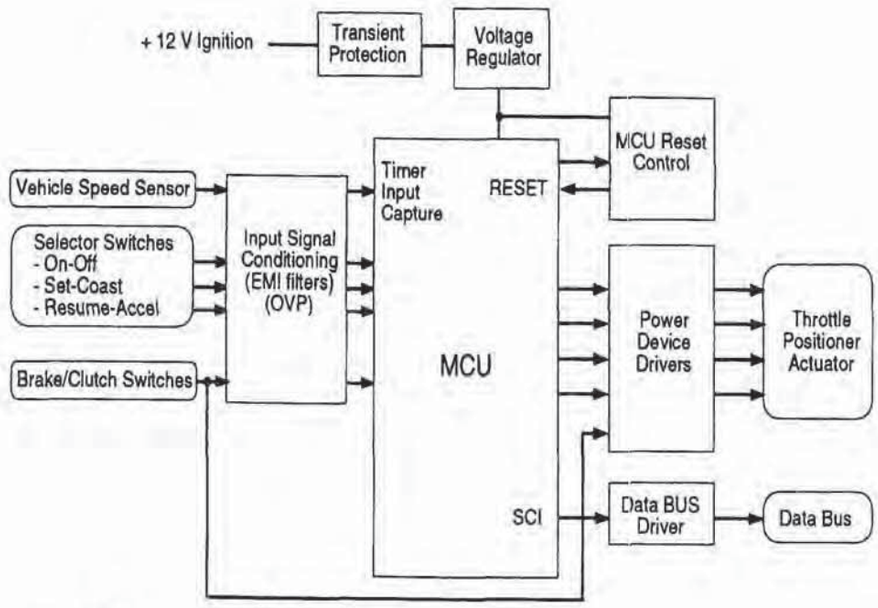


FIGURE 14.1 Cruise control system.

- *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.

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US PATENT NO. 6,557,540

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 ½ 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 Kp. Another signal i is generated that ramps up or down at a rate set by the error signal magnitude. The gains of both Ki and Kp 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 Ki will tend toward zero

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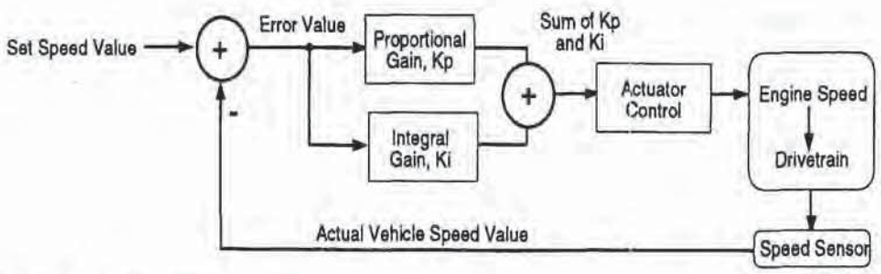


FIGURE 14.2 PI speed error control.

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.

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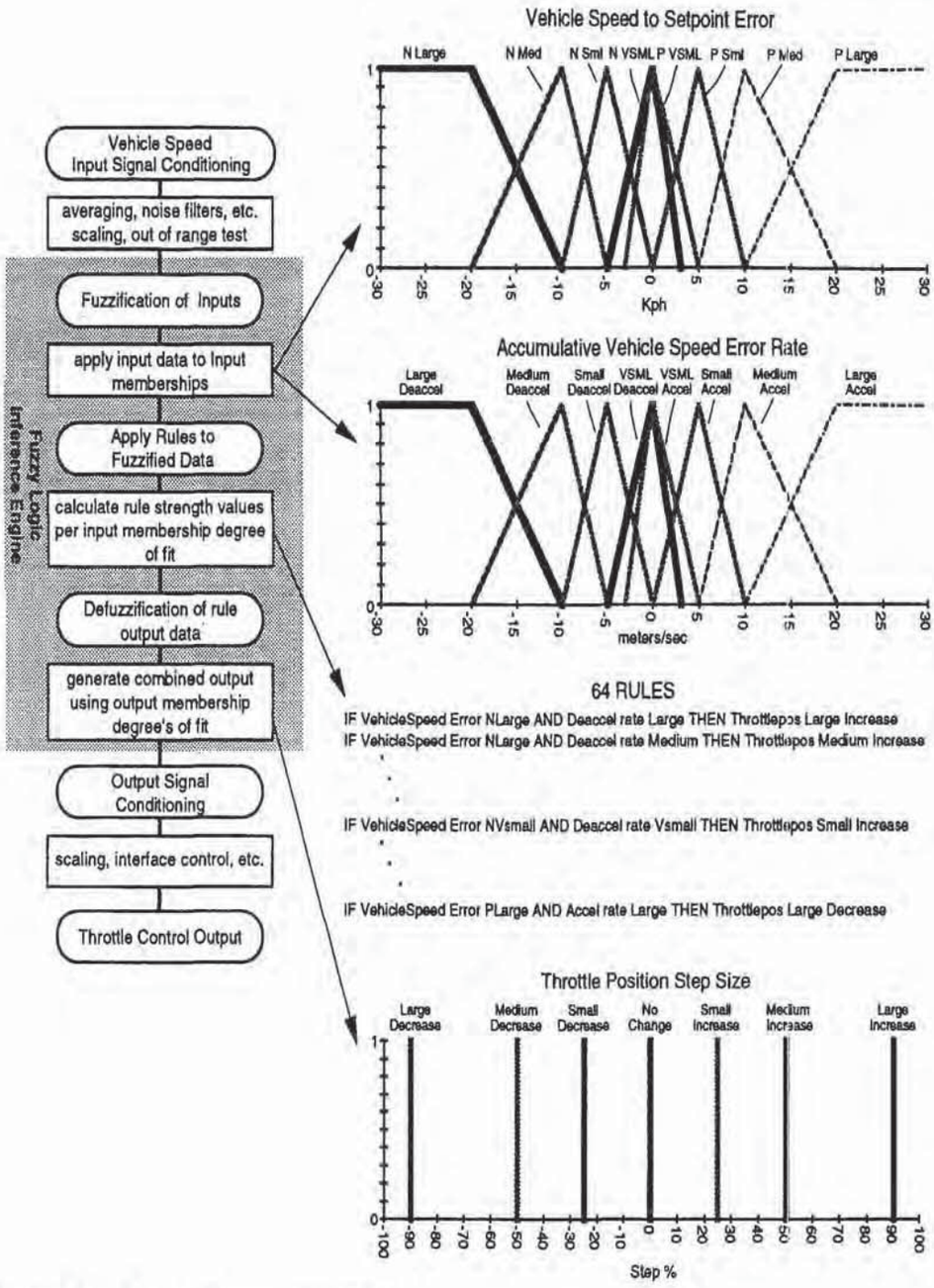


FIGURE 14.3 Fuzzy speed error program flow.

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

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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 setspeed; 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.

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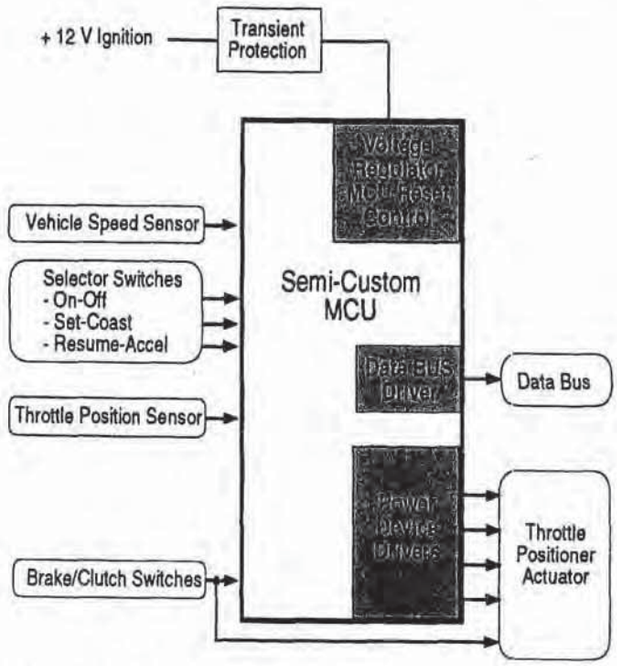


FIGURE 14.4 Automatic cruise control.

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.

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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 inputvarA is *x*, AND inputvarB 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.

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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).

$$F_x = \mu W_{wh} \tag{15.1}$$

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

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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:

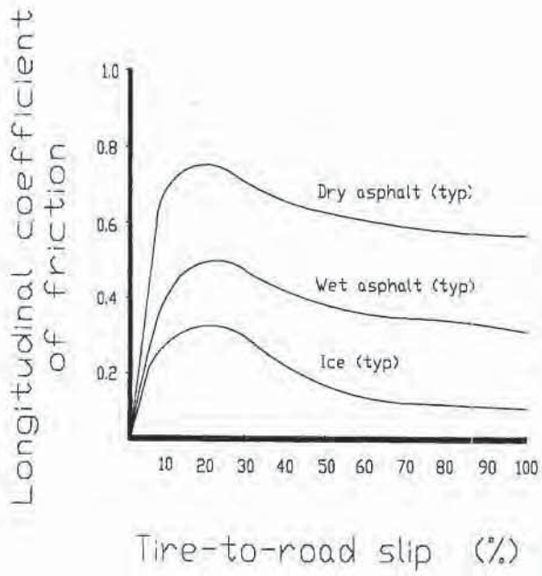


FIGURE 15.1 Longitudinal coefficient of friction as a function of wheel slip.

- 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:

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

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where F_y = friction force, by direction
$\mu_{lateral}$ = 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]

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

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 = $(R_{xf} + R_{xr}) / W\cos\Theta$

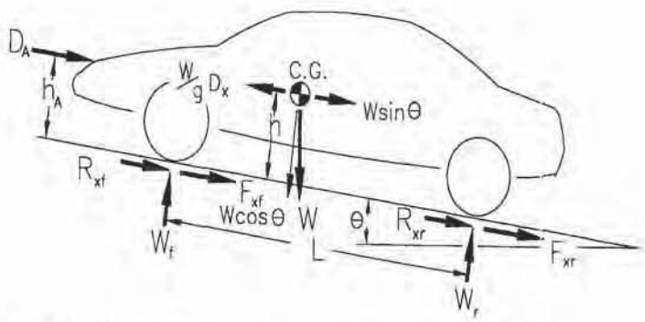


FIGURE 15.2 Significant forces' action on a vehicle during braking.

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]

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

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

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

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.

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

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.

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

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

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

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 $\Theta = 0°$ 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:

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

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

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.

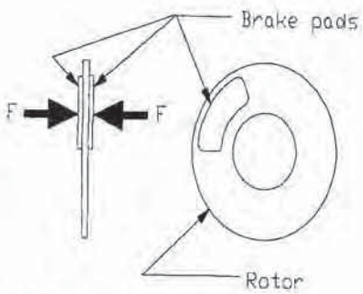


FIGURE 15.3 Disk brake schematic.

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:

$$T = PAER \tag{15.10}$$

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

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US PATENT NO. 6,557,540

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.

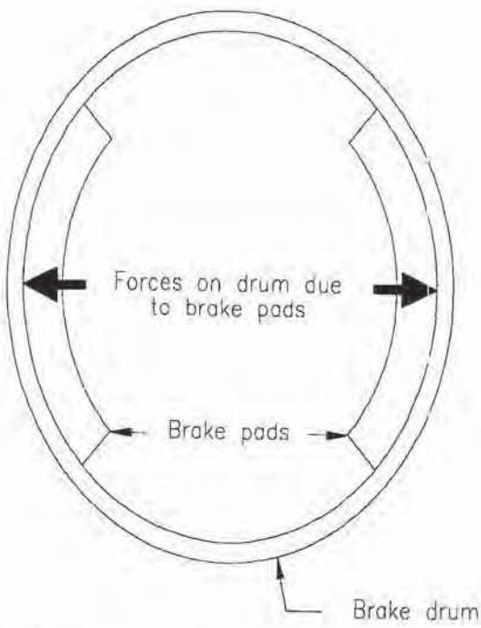


FIGURE 15.4 Drum brake schematic.

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

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US PATENT NO. 6,557,540

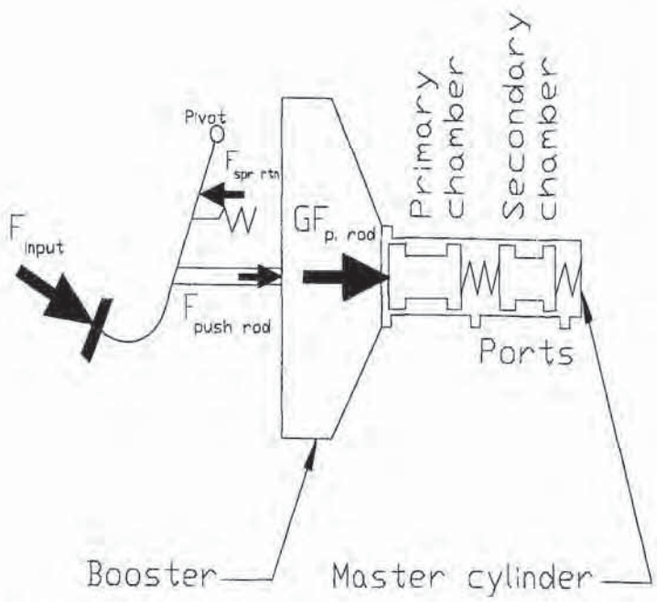


FIGURE 15.5 Brake pedal, vacuum booster, and master cylinder schematics.

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

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US PATENT NO. 6,557,540

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.

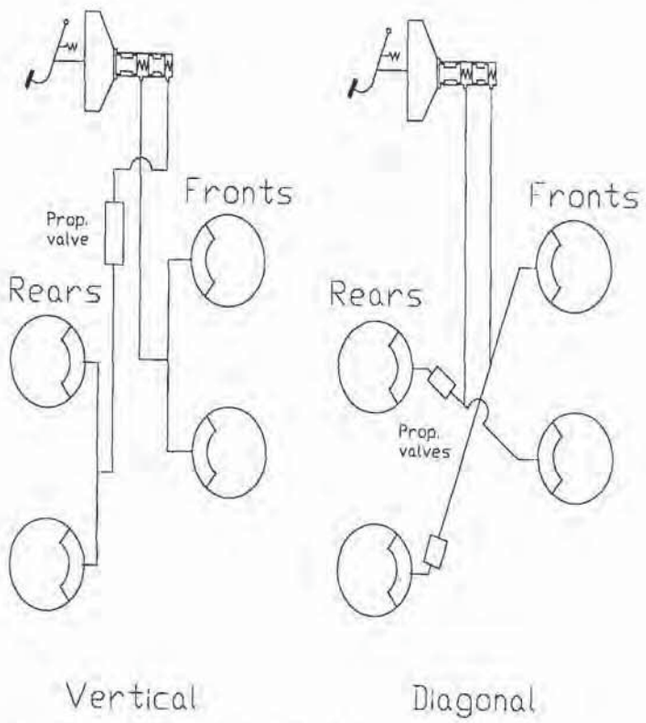


FIGURE 15.6 Vertical and diagonal split brake systems schematics.

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

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US PATENT NO. 6,557,540

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.

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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.

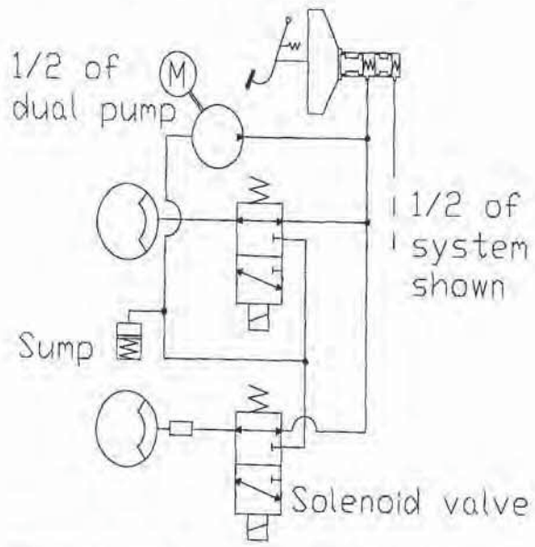


FIGURE 15.7 Simplified solenoid valve antilock system schematic.

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

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US PATENT NO. 6,557,540

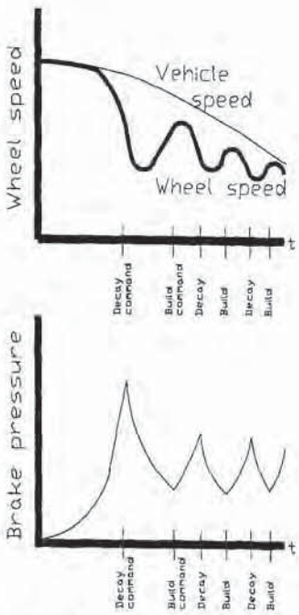


FIGURE 15.8 Typical antilock brake cycling.

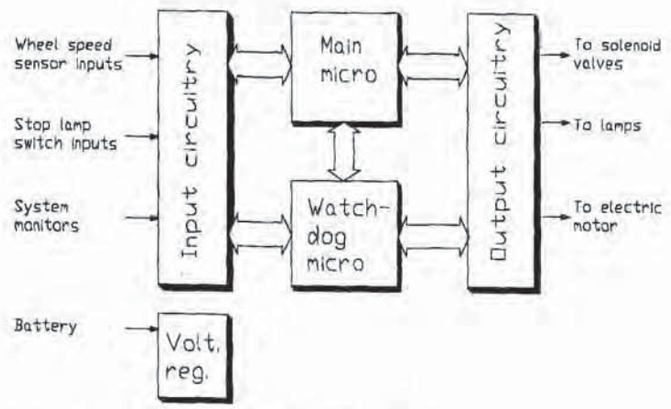


FIGURE 15.9 Electronic control unit block diagram.

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.

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US PATENT NO. 6,557,540

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

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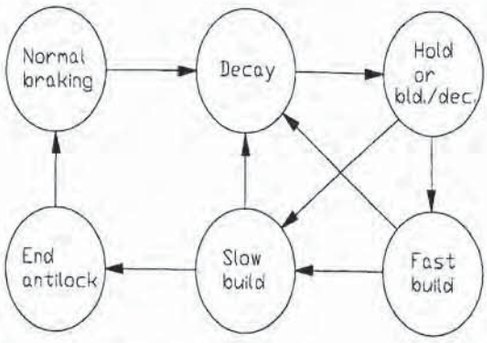


FIGURE 15.10 Simplified single-channel state diagram.

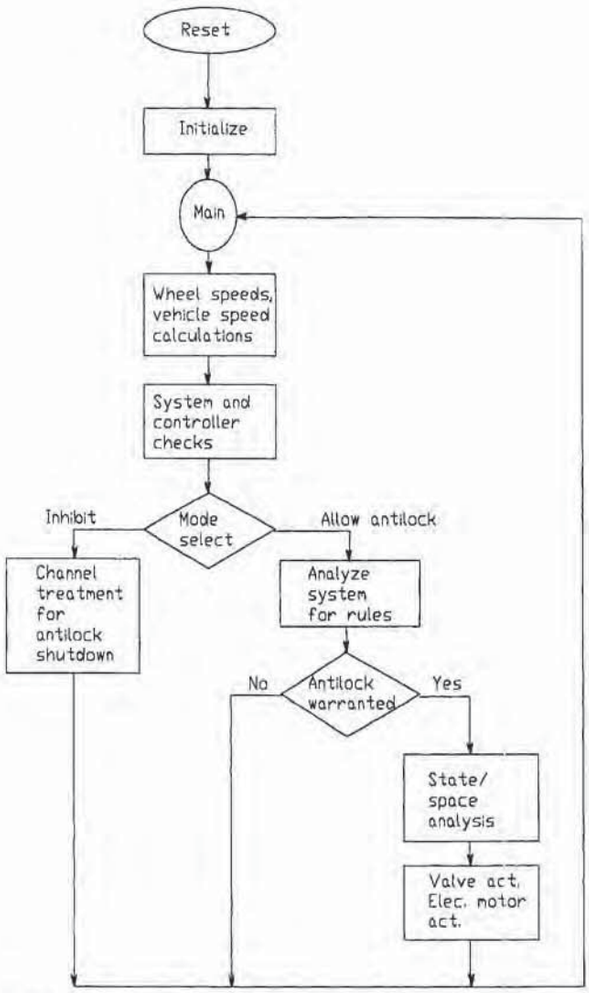


FIGURE 15.11 Simplified antilock flowchart.

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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.

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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.

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US PATENT NO. 6,557,540

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.

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US PATENT NO. 6,557,540

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.

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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 Θ_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.

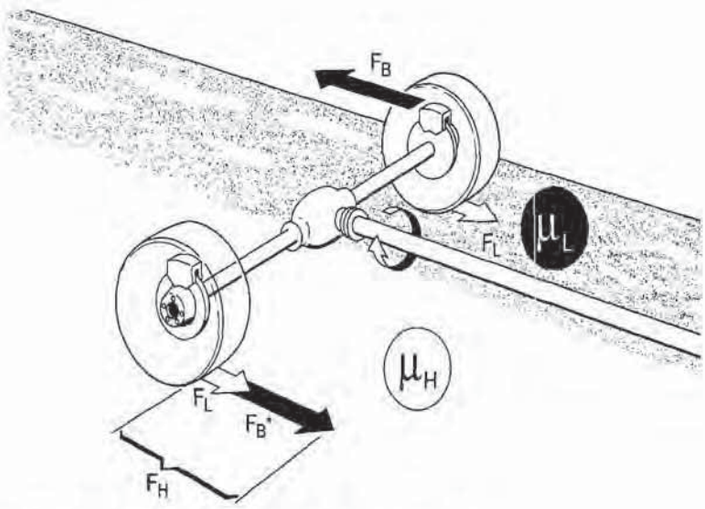


FIGURE 16.1 Braking intervention to limit differential slip.

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

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US PATENT NO. 6,557,540

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

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

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.

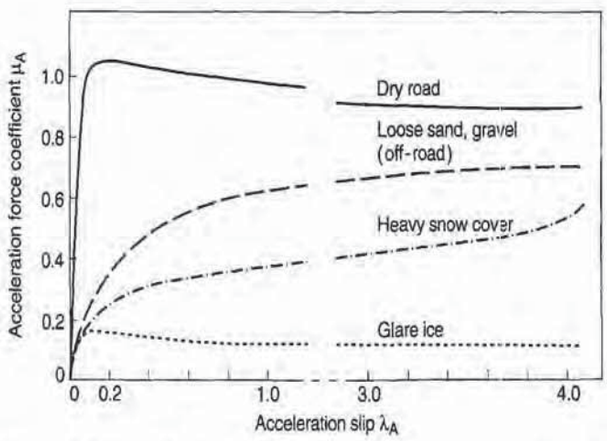


FIGURE 16.2 Adhesion coefficient for acceleration μ_A as a function of acceleration slip λ_A.

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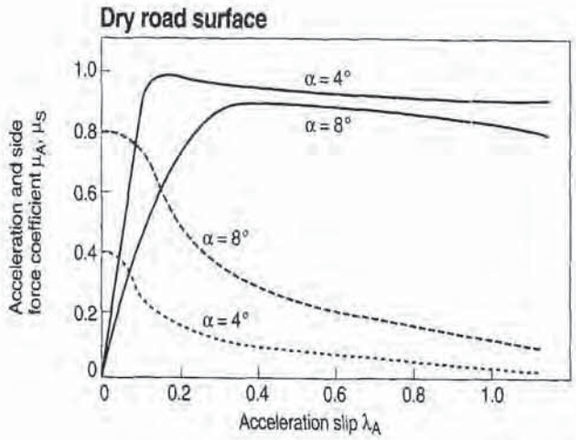


FIGURE 16.3 Acceleration and lateral traction coefficients at different slip angles α.

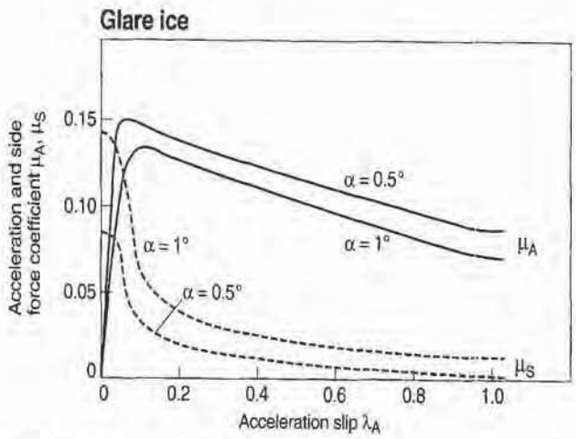


FIGURE 16.4 Acceleration and side-force coefficients at different slip angles α.

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.

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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. ≤2°). Relatively diminutive slip angles (≤0.05°) 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.

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US PATENT NO. 6,557,540

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).

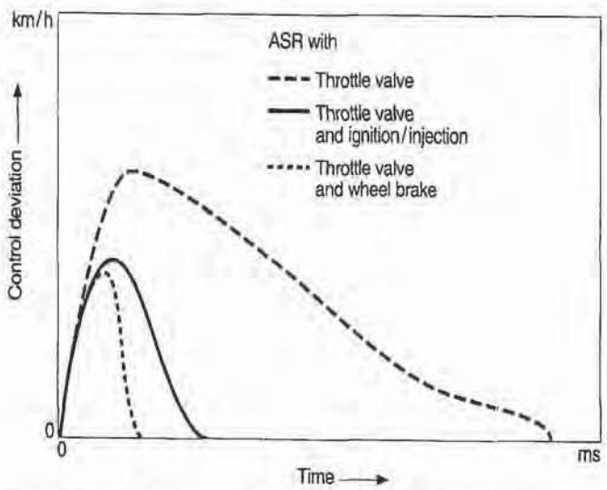


FIGURE 16.5 Deviation of the controlled variable during first control cycle with different actuators.

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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

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US PATENT NO. 6,557,540

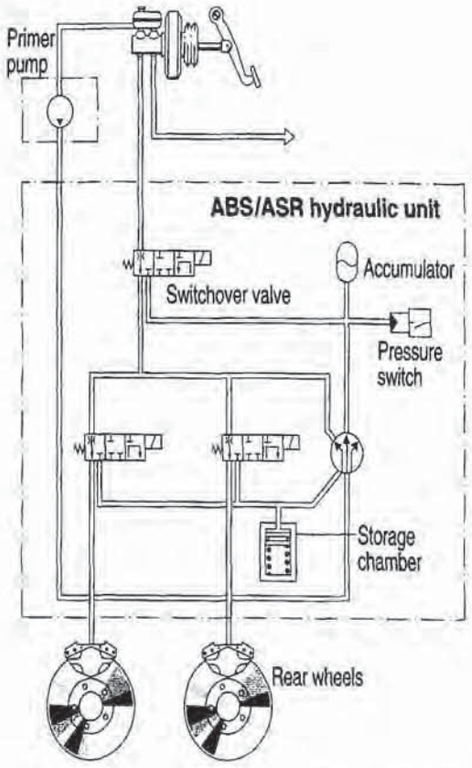


FIGURE 16.6 Diagram showing the ABS/ASR hydarulic circuit using braking force with stored energy (integrated system).

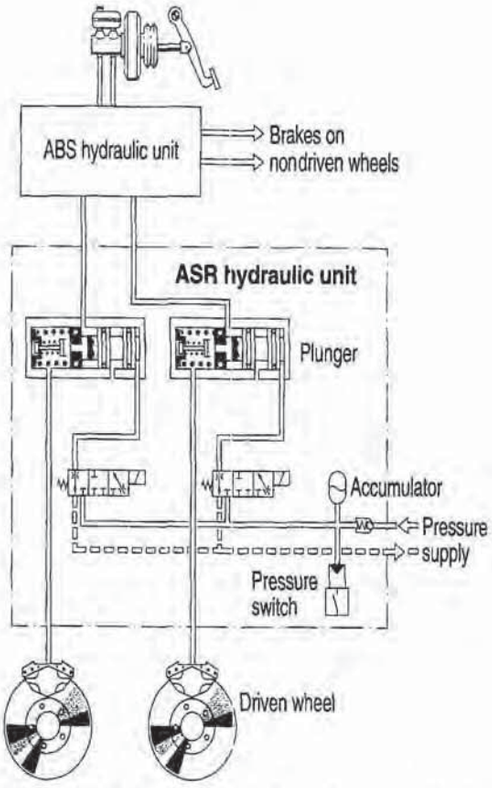


FIGURE 16.7 Diagram showing hydraulic unit for ASR system with braking intervention using stored energy via plunger.

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).

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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.

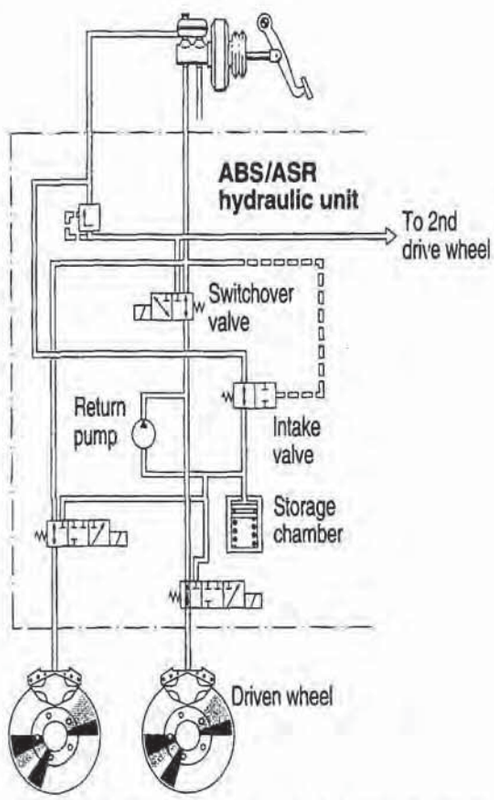


FIGURE 16.8 Diagram showing the ABS/ASR hydraulic unit employed for braking intervention without stored energy.

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.

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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.)

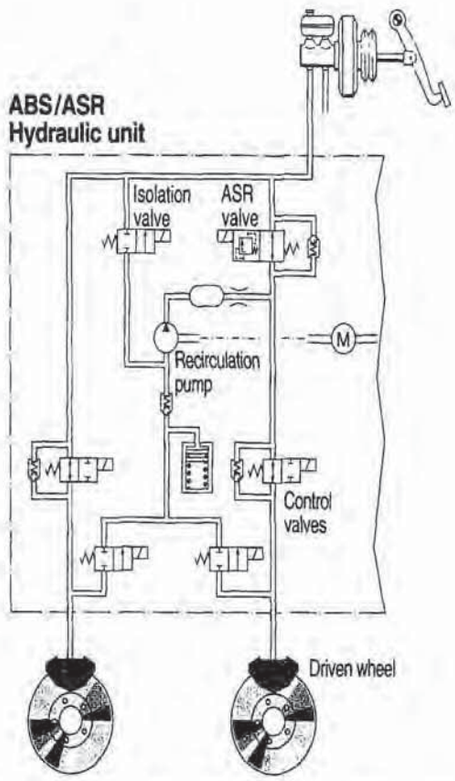


FIGURE 16.9 New design for brake torque control without stored energy.

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

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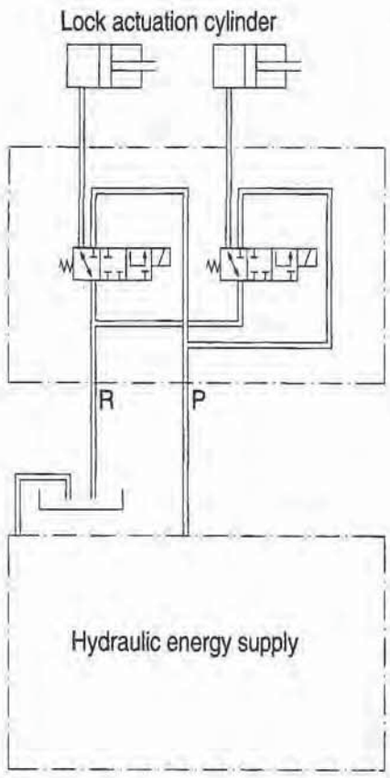


FIGURE 16.10 Hydraulic unit for control of rear-axle and interaxle differential locks.

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

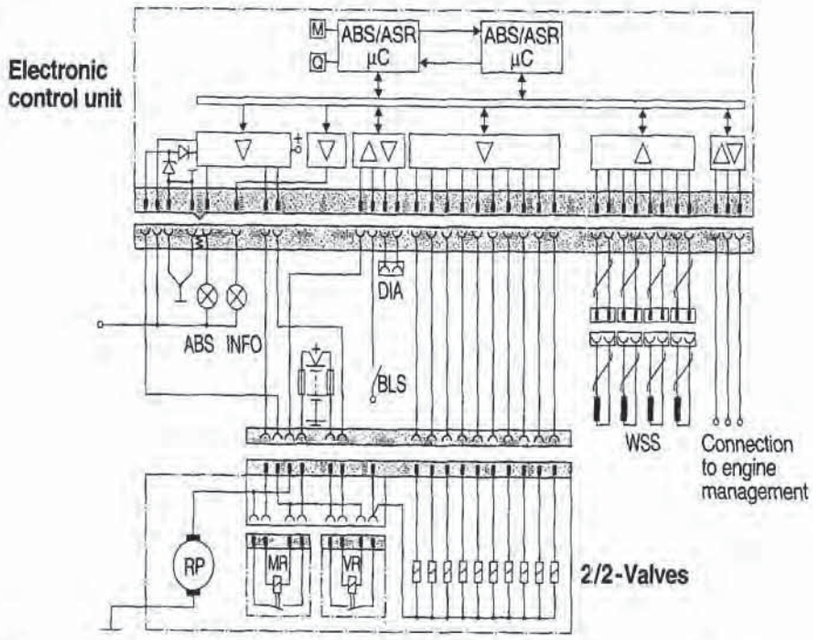
The system employs the same wheel-speed sensors that provide the information for the antilock braking system.

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.

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US PATENT NO. 6,557,540



M :	memory	BLS :	brake light switch
Q :	quartz	WSS :	wheel speed sensors
ABS/ASRµC :	ABS/ASR microcomputer	RP :	recirculation pump motor
ABS :	ABS warning lamp	MR :	motor relay
INFO :	information lamp	VR :	valve relay
DIA :	diagnosis connector		

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.

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With suppression according to individual cylinders, the number of control increments is the same as the number of cylinders. Because this limited number of control stages is still inadequate for a (as an example) four-cylinder engine, a supplementary strategy is employed: this is referred to as alternating injection suppression. It consists of varying the number of active cylinders by one after every two crankshaft rotations to produce a mean torque lying between the torques produced at the two cylinder stages. This method doubles the number of control stages to achieve an acceptable level of driving comfort, while complementary reductions in ignition advance can be employed to provide additional incremental adjustments.

In cases where the excess torque is substantial, injection suppression can be supplemented by short-term ignition cutout to provide extremely rapid output reductions. Figure 16.12 shows the design of the system. In addition to the modest expense, this system also offers vehicle manufacturers an additional advantage in the form of space savings (no additional space required) and simplicity (limited amount of extra wiring).

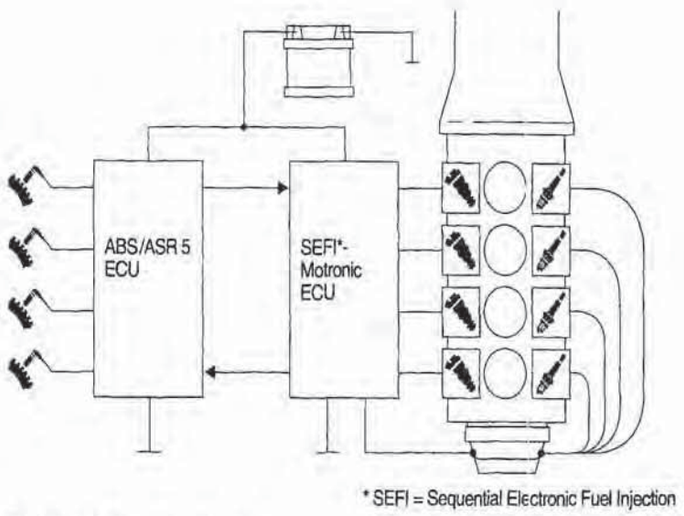


FIGURE 16.12 ASR EZ system.

16.6 APPLICATIONS ON HEAVY COMMERCIAL VEHICLES

Heavy commercial vehicles are used in a highly variegated range of applications. In principle, the ASR installed on these vehicles employs the same control strategies used for passenger cars: engine-output control and braking intervention.

Differences in vehicle application can make it necessary to employ ASR systems in various levels of complexity (for instance, relying on braking control or engine output control exclusively). The control unit recognizes the design stage and carries out its control functions accordingly. This makes it possible to employ the most economical ASR system for each vehicle type and particular application.

Figure 16.13 features a schematic diagram showing a typical 4×2 vehicle equipped with a top-of-the-line ABS/ASR system. During acceleration on μ-split surfaces, the brake-force regulator limits slip between the drive wheels. The ASR valve (4) is activated on the side with the spinning wheel, while the ABS pressure-control valve (3) allows graduated increases in pressure at the wheel cylinder.

If both drive wheels start to spin on a road surface affording equal traction on both sides, the engine-output controller responds by reducing the drive slip to optimum values. In this example, the electric performance control (EPC) (10) assumes the role of final-control ele-

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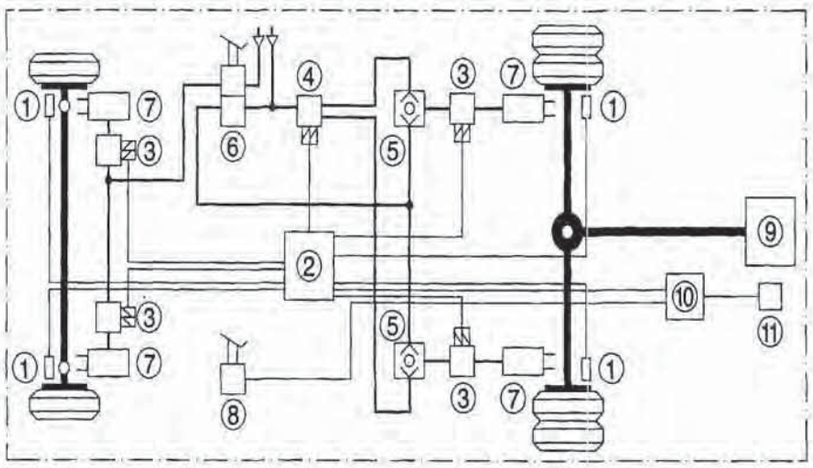


FIGURE 16.13 Four-circuit ABS/ASR, 4 × 2 vehicle.

ment. The ABS/ASR control unit (2) transmits the desired increment of reduction to the EPC control unit via interface, and also controls the electric motor at the injection pump.

16.7 FUTURE TRENDS

Ensuring driving stability is the most important task of ASR with rear-wheel-drive cars. This task can be achieved by a fast engine torque control or a combination of throttle control and a fast brake control.

ASR systems with a fast engine torque control (ignition and injection intervention) and brake control without stored energy will be widely used.

Although the application of controllable differential locks also offers efficient ASR control, the higher costs of this system will prevent wide usage.

The predominant demand on ASR systems for front-wheel-drive cars is that of traction optimization. Therefore, an ASR system with brake torque control is needed. The combination of brake torque control and engine torque control results in a complex, efficient system.

In the future, only ASR systems with brake torque control will be widely used with front-wheel-drive cars. Especially the combination of brake torque control and engine torque control with ignition and injection intervention will be widely used.

GLOSSARY

ABS return pump A piston pump that draws back brake fluid to the master cylinder.

Alternating injection suppression A variation of the number of active cylinders by one after every two crankshaft rotations in order to modulate the engine torque.

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US PATENT NO. 6,557,540

ASR deactivation switch A device to switch off ASR on sand and loose gravel in order to achieve maximum traction on these surfaces.

Automatic throttle valve actuator A simple actuator for automatic throttle angle reduction in case of excessive acceleration slip.

Braking intervention Automatic brake application at drive wheels in case of excessive acceleration slip.

D controller A controller with differentiating characteristics.

Electronic performance control Electronic accelerator control.

Engine torque control An actuator to modulate engine torque in case of excessive acceleration slip.

PI controller Controller with proportional and integral characteristics.

Slip threshold switch Switch to increase desired slip threshold on sand and loose gravel.

Switchover valve A valve to switch hydraulic performance from normal braking to ASR performance.

Throttle valve control ASR actuator to modulate the throttle angle.

3/3 ABS valve A valve with three connections and three positions for ABS wheel pressure modulation.

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Dipl. Ing. Armin Czinczel received a degree in mechanical engineering at the Technical University in Hanover, Germany. He started his professional career with development work on navigation systems. He joined Bosch in 1968, working on ABS development until his retirement in 1994.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

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US PATENT NO. 6,557,540

CHAPTER 17
SUSPENSION CONTROL

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The function of a suspension system in an automobile is to improve ride comfort and stability. An important consideration in suspension design is how to obtain both improved ride comfort and stability, since they are normally in conflict. Advances in electronic control technology, applied to the automobile, can resolve this conflict.

17.1 SHOCK ABSORBER CONTROL SYSTEM

During the past 20 years, many different damping control systems have been studied. The main purpose of all these systems is to select the optimum damping force for various driving conditions. The first function of a shock absorber is to control vehicle movement against inertial forces, such as roll when the vehicle turns and pitch when the vehicle is braked. The second function is to prevent vehicle vibration caused by road surface inputs. To satisfy both functions it is necessary to control damping forces.

There are three basic parts of a damping control system: a damping control device (actuator), sensors, and software (control strategy). Optimum damping forces should be set for various running conditions in order to improve ride comfort and handling stability.

17.1.1 System Configuration

One of the damping control system configurations is shown in Fig. 17.1. This system uses five sensors, including a supersonic road sensor, to detect running conditions. Control signals are sent to adjust the damping force of the variable shock absorbers to optimum values. A main advantage to this type of system is that, through the use of a road sensor, it can provide optimum control in accordance with the actual road conditions. This system incorporates three discrete damper characteristics. Sensors used are: a vehicle speed sensor, a steering angle sensor, an acceleration and deceleration sensor, a brake sensor, and a supersonic sensor to detect road conditions.

One system uses four piezo sensors and four piezo actuators on each wheel in order to change the damping forces as quickly as possible. This system incorporates two discrete damper characteristics. Sensors used are: four piezo sensors on each wheel, a stop lamp switch, a steering sensor, and a vehicle speed sensor.

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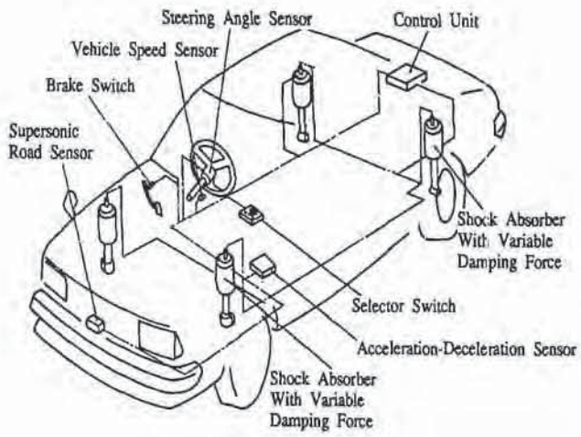


FIGURE 17.1 Principal components of a damping control system.

17.1.2 Actuator and Sensor

Actuator. Discrete damping control actuators often use a built-in motor to change the damping force. This motor turns a rotary valve to select the orifice diameter for three different damping levels: soft, medium, and hard. The stopping position of the rotary valve is controlled by encoder signals.

Another actuator is a piezo actuator consisting of 88 piezo elements, and it is installed in the piston rod of a shock absorber. When a high voltage (500 V) is applied to the piezo actuator, it expands about 50 μm with reverse piezoelectric effect. Elongation in the piezo actuator causes the plunger pin to be pushed out through the displacement hydraulic coupling unit. As a result, the plunger pin moves down to open the bypass of the damping force switching valve. The result is a soft damping force. Figure 17.2 shows this valve.

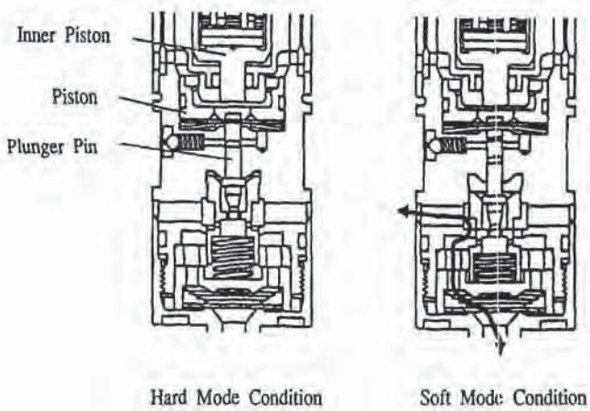


FIGURE 17.2 Damping force switching valve.

Sensor. A sensor using a supersonic wave to detect the road surface is shown in Fig. 17.3. The vehicle height from road to body is calculated on the basis of reflection time T. Judgment of the road condition is made by analyzing the pattern of change in vehicle height.

A sensor using the piezoelectric effect is shown in Fig. 17.4. Installed in the piston rod of a shock absorber, the sensor generates an electric charge in accordance with axial force from the road surface.

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US PATENT NO. 6,557,540

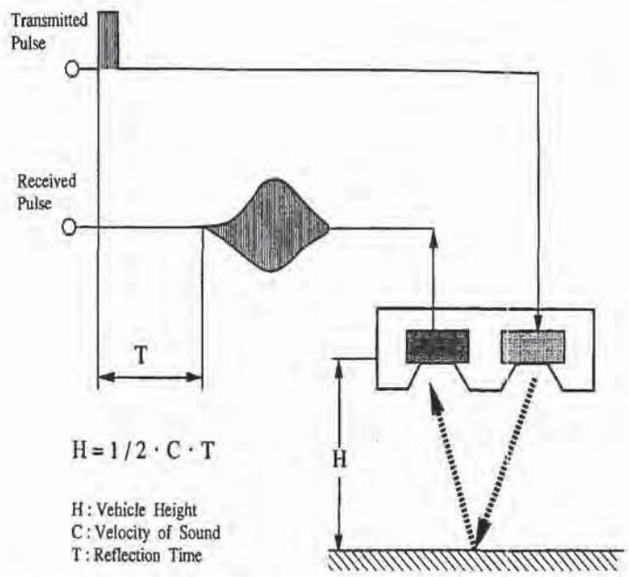


FIGURE 17.3 Supersonic sensor.

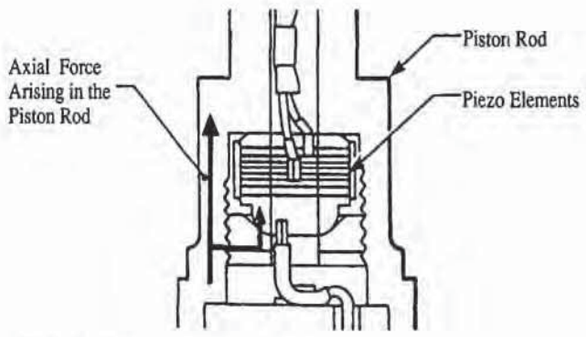


FIGURE 17.4 Piezoelectric sensor.

17.1.3 Control System

The control system of Fig. 17.1 is shown in Table 17.1. The occurrence of roll and pitch can be predicted from various sensors. Single bumps or dips are detected from changes in vehicle height. The outline of the road surface judgment logic is shown in Table 17.2.

The block diagram of the system using four piezo sensors and four piezo actuators is shown in Fig. 17.5. The damping force arising in an absorber increases instantaneously as the tire or wheel goes up or down in accordance with unevenness of the road surface. Each piezo sensor outputs to the electronic control unit continuous signals indicating damping force differential. If the value of damping force differential exceeds a predetermined level, the system switches from firm mode to soft mode. The system is designed so that the firm mode re-engages immediately after the vibrations due to poor road surface cease.

Many of the limitations and disadvantages of the conventional shock absorber can be eliminated by the damping control systems. They include semiactive suspension systems which are capable of providing both ride comfort and good handling.

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US PATENT NO. 6,557,540

TABLE 17.1 Damping Force Control System

Control objectives		Sensors used					Damping force Front / Rear
		Vehicle speed	Steering angle	Accel/ decel	Brake	Road condition	
Roll	Roll reduction for quick steering operation	○	○				Hard / Hard
Pitch	Reduction of nose diving by braking				○	○	Hard / Hard
	Reduction of pitching when accelerating and decelerating	○		○			Medium / Medium
Bouncing	Reduction of light, bouncy vibrations in bottoming	○				○	Medium / Medium
	Reduction of light, bouncy vibrations in bouncing on a heaving road	○				○	Medium / Medium
Road holding performance	Road holding performance improvement when running on rough roads	○				○	Medium / Medium
Others	Stability improvement at high speed	○					Medium / Soft
	Prevention of shaking when stopping and rocking when passengers exit or enter	○					Hard / Hard

TABLE 17.2 Road Condition Judgment Logic

High-frequency components	Low-frequency components	
	Small	Large
Small	Smooth road	Heaving road
Large	damping force control unnecessary	Rough road

17.2 HYDROPNEUMATIC SUSPENSION CONTROL SYSTEM

A hermetically sealed quantity of gas is used in the hydropneumatic suspension control system. The gas and hydraulic oil are separated by a rubber diaphragm, as shown in Fig. 17.6*a*. The mechanical springs are replaced by gas. The shock absorber damping mechanism is achieved by the orifice fitted with valves.

17.2.1 System Configuration

As shown in Fig. 17.6*b*, by adding an additional sphere to the hydropneumatic system, a controllable hydropneumatic system can be realized. If the regulator is closed, the system is in a firm mode. If the regulator is open, the spring constant of the suspension system becomes lower by increasing the total volume of the sphere, and the total damping force is reduced.

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US PATENT NO. 6,557,540

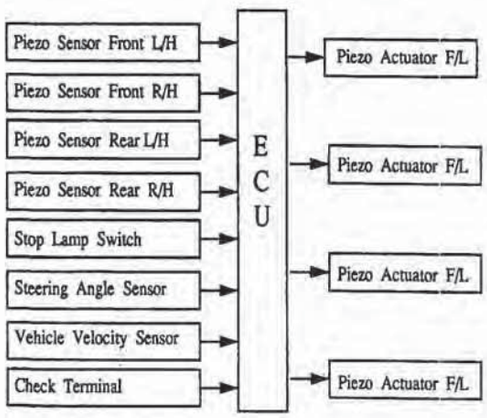


FIGURE 17.5 Block diagram of damping control system.

Depending on the sensors used, which detect vehicle driving and road surface conditions, this system can change the regulator characteristics in order to achieve both good ride comfort and handling stability.

17.3 ELECTRONIC LEVELING CONTROL SYSTEM

In a pneumatic and hydropneumatic suspension, the vehicle body can be maintained at a constant height from the road surface, keeping a low spring constant. The advantages of an electronic control system are:

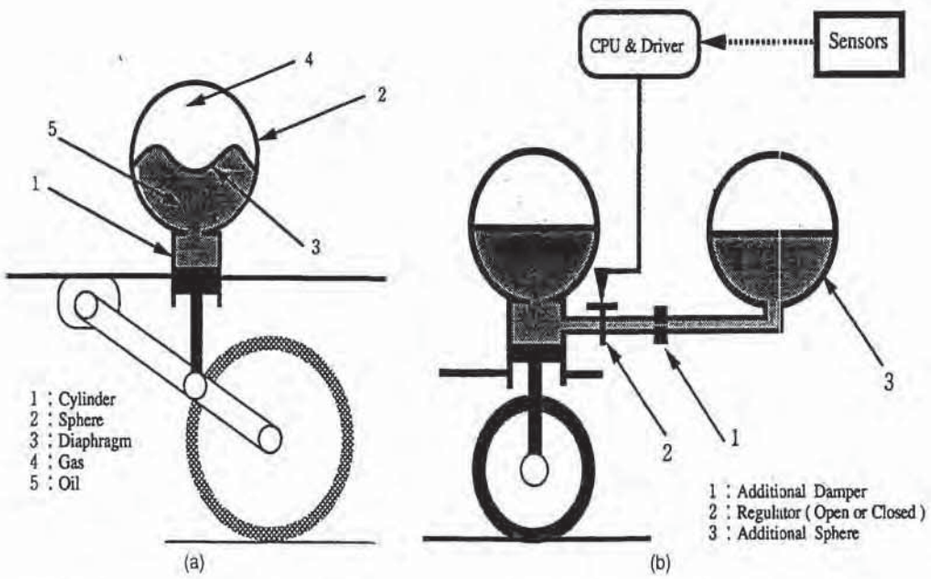


FIGURE 17.6 (*a*) Hydropneumatic suspension system; (*b*) Controllable hydropneumatic suspension system.

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US PATENT NO. 6,557,540

- Keeping a low spring rate to achieve good ride comfort independent of load conditions
- Increase in vehicle body height on rough road surfaces
- Changing spring rate and damping force in accordance with driving conditions and road surfaces

17.3.1 System Configuration

The system is shown in Fig. 17.7. It consists of eight sensors, a mode select switch, air spring/shock absorber units on four wheels, actuators to operate the changing valves in the unit, a compressor unit and five height control valves for air springs, and an electronic control unit (ECU). The system configuration is shown in Fig. 17.8.

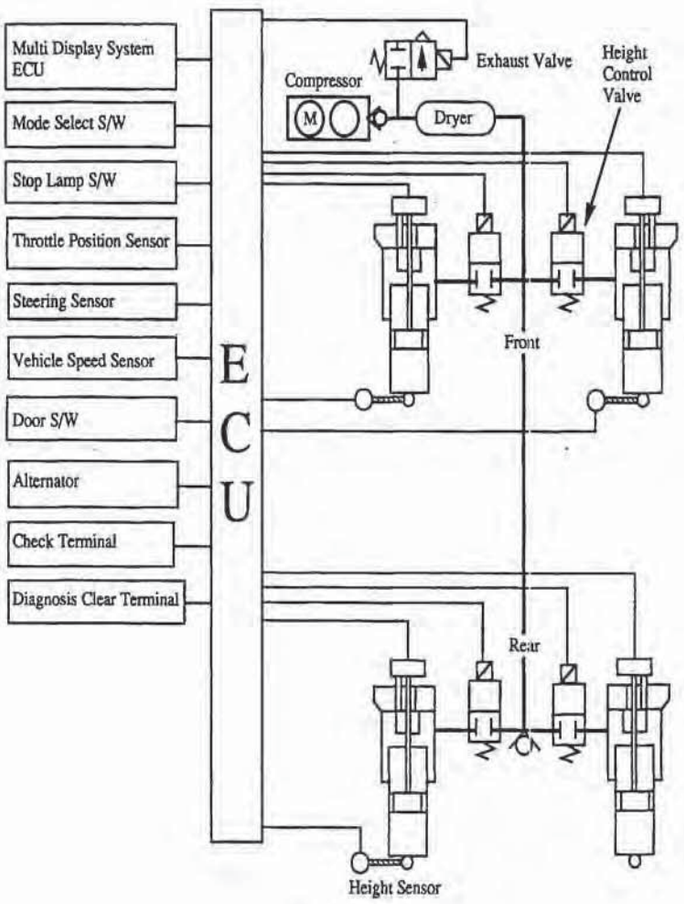


FIGURE 17.7 Principal components of an air suspension system.

17.3.2 Components

The structure of the air suspension unit consists of a shock absorber, a pneumatic piston surrounding the shock absorber, main and sub-air chambers, a rolling diaphragm, and valves which change the suspension stroke.

The actuator uses a dc motor, which has two shafts to operate the valves for the air spring and the shock absorber. The rotation of the motor is reduced by the sector gear and operates

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US PATENT NO. 6,557,540

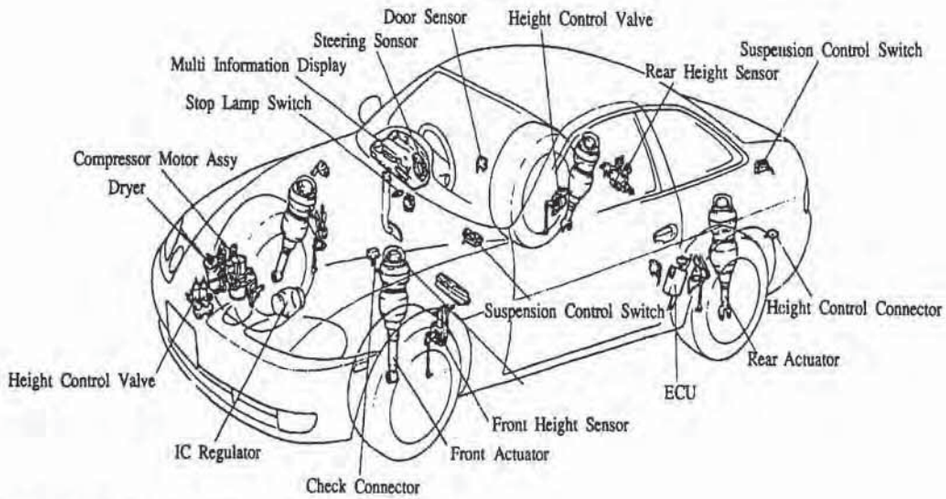


FIGURE 17.8 Configuration of an air suspension system.

the rotary valve to change the damping force. At the same time, another gear engaged with the sector gear operates the air valve to change the spring rate.

17.3.3 Control System

This system can change the spring rate and the damping force into three levels and vehicle height levels of low, normal, or high can be selected. One of the control logics is shown in Table 17.3. This is a control which changes the suspension characteristics in response to vehicle speed and road conditions. The spring rate/damping force and the vehicle height are controlled independently according to each control logic.

Electronic leveling control systems do not need much energy to control vehicle height. They control both spring rate and damping force. As a result of keeping the low spring rate, electronic leveling control systems can provide both good ride comfort and handling stability.

TABLE 17.3 Basic Control Logic of Air Suspension System

		Spring rate, damping force					
		Soft mode			Medium mode		
Function	Operating condition	Soft	Medium	Firm	Soft	Medium	Firm
Antiroll	Rapid steering	○		→ ○		○	→ ○
			○	↗			
Antidive	Braking at V* > 60 km/h	○		→ ○		○	→ ○
			○	↗			
Antisquat	Rapid starting at V < 20 km/h	○		→ ○		○	→ ○

* V: Vehicle speed.

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17.4 ACTIVE SUSPENSION

Suspension control systems for passenger cars have evolved through several stages over the years. Work in the field began with the air suspension for controlling vehicle height and then progressed to the hydropneumatic suspension and suspensions with variable damping force and spring rate control. Now efforts are underway to develop an active suspension. It is defined as one that has the following features:

- Energy is constantly supplied to the suspension and the force generated by that energy is continuously controlled
- The suspension incorporates various types of sensors and a unit for processing their signals that generates forces that are a function of the signal outputs

17.4.1 System Configurations

Basic Configuration. Hydraulic active suspension can be divided into two large systems: the hydraulic system and the control system, as shown in Fig. 17.9. The hydraulic pressure of each of the actuators located on each wheel is controlled in accordance with the output values from the G sensors to suppress changes in vehicle body position (bounce, pitch, roll) and reduce vibration from the road surface.

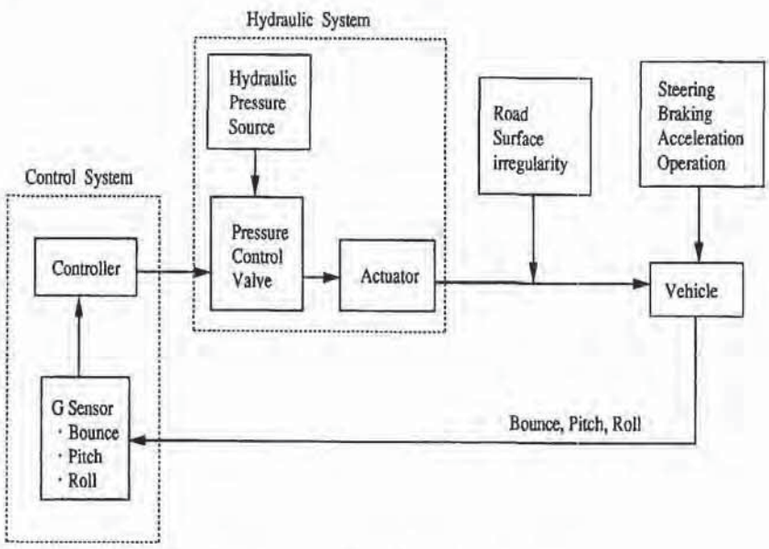


FIGURE 17.9 Hydraulic and control systems for an active suspension.

Hydraulic and Control System Configuration. A basic overview of the system is shown in Fig. 17.10. The functions of the main units of the hydraulic system are shown in Table 17.4.

As shown in Fig. 17.11, the control system contains the controller and all of the sensors including the vertical G sensors, lateral G sensors, fore and aft G sensors, and vehicle height sensors.

17.4.2 Components

Oil Pump. The oil pump has seven cylinders arranged around the circumference and can output a maximum oil flow of 12 liters per minute. The pump is connected in tandem with the power steering vane pump.

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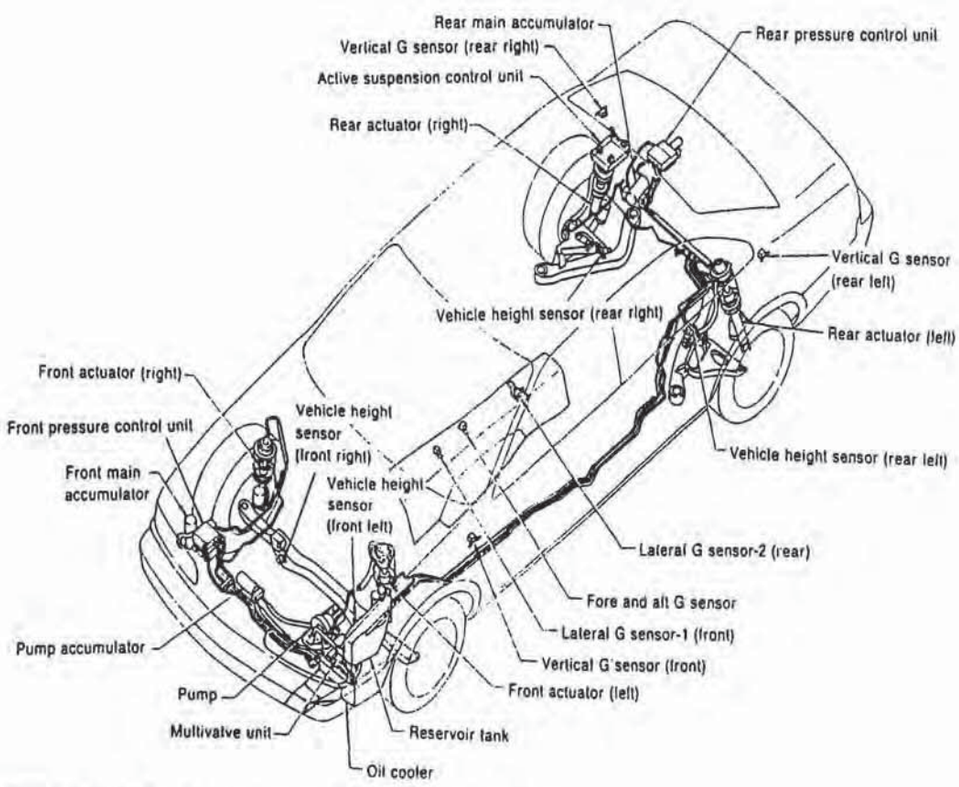


FIGURE 17.10 Principal components of an active suspension system.

Pump Accumulator. To dampen pulsating hydraulic pressure generated by the oil pump, pump accumulators are installed in the hydraulic supply unit, including one on the side of the oil pump. To dampen the high-frequency pulsations, a metal-bellows-type accumulator is used.

Multivalve Unit. As shown in Fig. 17.12, the multivalve unit contains valves for many different functions. The main purpose of the multivalve unit is the basic control of hydraulic pressure for the whole system. The function of the multivalve unit is shown in Table 17.5.

Main Accumulator. The main accumulator is positioned at both the front and rear axles. The main accumulator has two principal functions: it stores oil from the multivalve unit and provides extra flow to the actuators when necessary, and it preserves vehicle height when the engine is turned off.

TABLE 17.4 The Functions of Main Units of the Hydraulic Systems

	Main basic function
Oil pump	Supplies the necessary oil for system operation (power supply)
Pump accumulator	Removes the pulsating action from pressurized oil supplied by the oil pump
Multivalve unit	Controls the supply of pressurized oil, failsafe function, etc.
Main accumulator	Maintains oil pressure, compensates when large amount of flow is required and preserves vehicle body height
Pressure control unit	Controls the hydraulics for the actuators on each wheel according to signals received from the control unit
Actuator	Controls vehicle attitude and absorbs external forces from the road surface

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US PATENT NO. 6,557,540

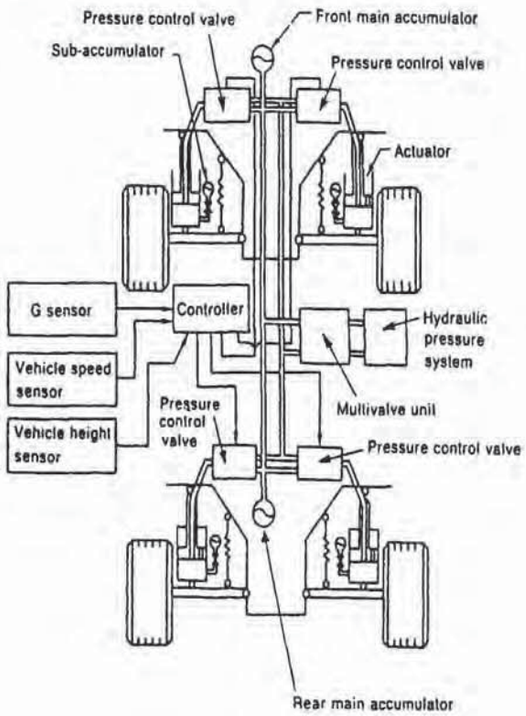


FIGURE 17.11 Controls for an active suspension system.

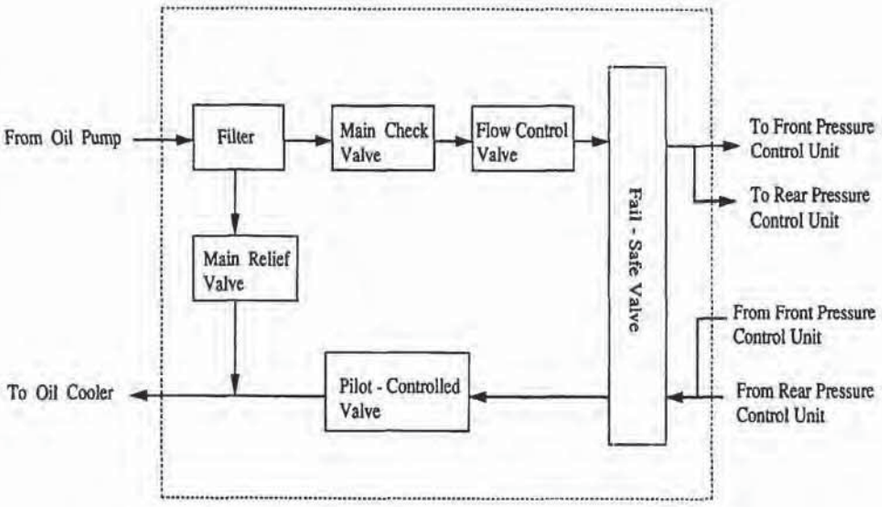


FIGURE 17.12 Multivalve unit configuration.

Pressure Control Unit. As shown in Fig. 17.13, the pressure control unit controls the hydraulic movement of the actuator of each wheel in accordance with instructions received from the control unit. Table 17.6 shows the valve's construction and operating principle.

The electrohydraulic pressure control system consists of a pressure actuator and a pressure control valve. The actuator is of the single acting type and is provided with a damping valve and an accumulator below the cylinder. The pressure control valve is built with three ports

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TABLE 17.5 Multivalve Unit Functions

Function	Valve	Outline
Pressure supply management function	Main relief valve	When the oil pressure exceeds a constant value, the main relief valve will return some of the oil flow. This keeps the oil supply pressure at a constant pressure.
Vehicle height maintenance function	Main check valve Pilot-controlled check valve	The main check valve is a nonreturn valve that controls the flow from the line filter and directs it to the flow control valve. The pilot controlled check valve is a supply-pressure-reaction-type open/closed valve. When the hydraulic pressure exceeds a constant value, the valve opens and when the hydraulic pressure falls below that level, it closes. In addition, it maintains the hydraulic pressure at a constant level when the engine is turned off.
Vehicle height control function	Flow control valve	When the engine is turned off, the flow control valve closes the main passage and directs the flow through the bypass passage orifice, slowly increasing the hydraulic pressure, after which the main passage is opened. This prevents any sudden changes in vehicle height when starting the engine.
Failsafe function	Failsafe valve	When any irregularities occur in the electronic system, it changes the hydraulic passage, preventing any sudden changes in vehicle height.

and employs a pilot type proportional electromagnetic control valve. This pressure control valve has two main functions:

- It controls the pressure of the actuator according to the control input. This is accomplished by driving the solenoid so that it adjusts the pilot valve, causing the spool to move.
- Feedback control is applied to move the spool in response to fluctuations in actuator pressure caused by road surface inputs; the action of the spool works to keep the actuator pressure at a certain fixed level.

Actuator. As shown in Fig. 17.14, the actuator consists of the hydraulic power cylinder, subaccumulator, damping valve, etc. Auxiliary coil springs are also employed to reduce the pressure necessary for the overall system and to reduce the amount of horsepower expended. The subaccumulator and damping valve at the bottom of the hydraulic power cylinder absorb and damp the high-frequency vibration from the road surface.

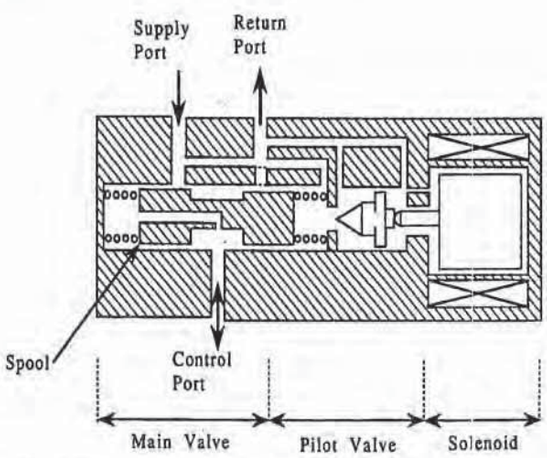


FIGURE 17.13 Pressure control valve construction.

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TABLE 17.6 Pressure Control Valve Functions

Active control function	The pressure in the control port (actuator) is controlled in response to the electric current applied to the solenoid, thus controlling the vehicle attitude.
Passive damping function	When various pressure levels are caused in the interior of the actuator by road surface forces, this pressure passes through the control port, causing feedback on the spool and the generation of appropriate damping forces.

Controller. As shown in Fig. 17.15, the controller is constructed using two 16-bit microcomputers, MCU1 and MCU2.

MCU1 processes signals from the G sensors and then sends attitude control signals to the pressure control valve solenoid drive circuit. MCU2 processes signals from the vehicle height sensors and then sends attitude control signals to the solenoid drive circuit.

MCU1 and MCU2 normally perform mutual transmission, but should an irregularity occur, the signal will be sent to the failsafe circuit, causing the failsafe valve to operate and thus guarantee a high degree of safety.

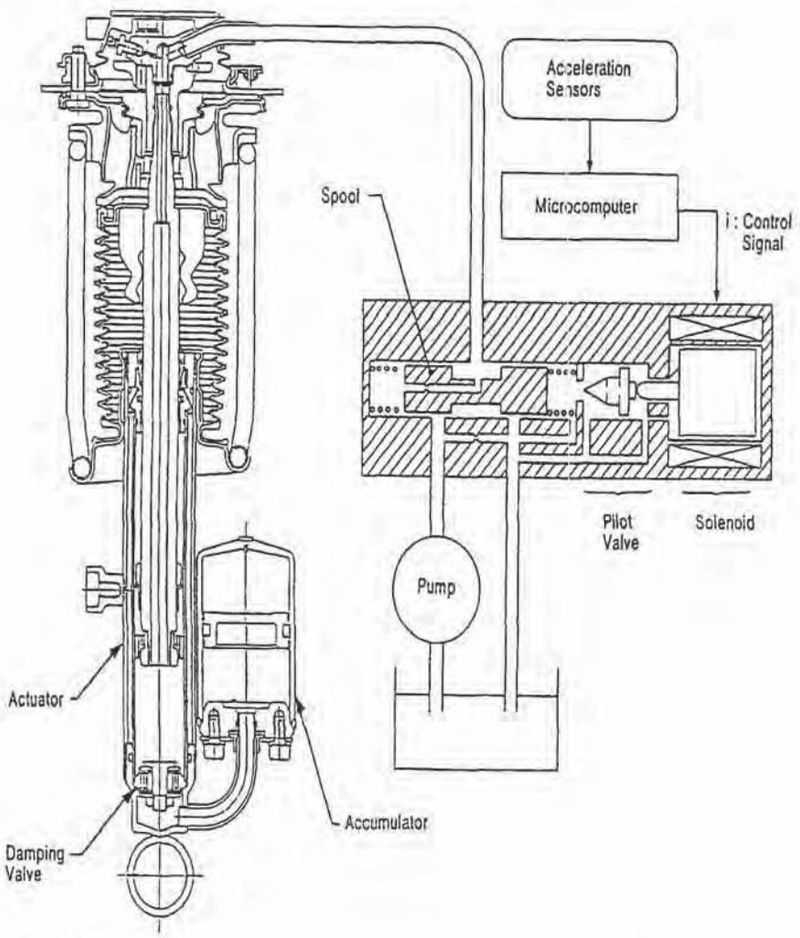


FIGURE 17.14 Schematic diagram of electrohydraulic pressure control system.

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US PATENT NO. 6,557,540

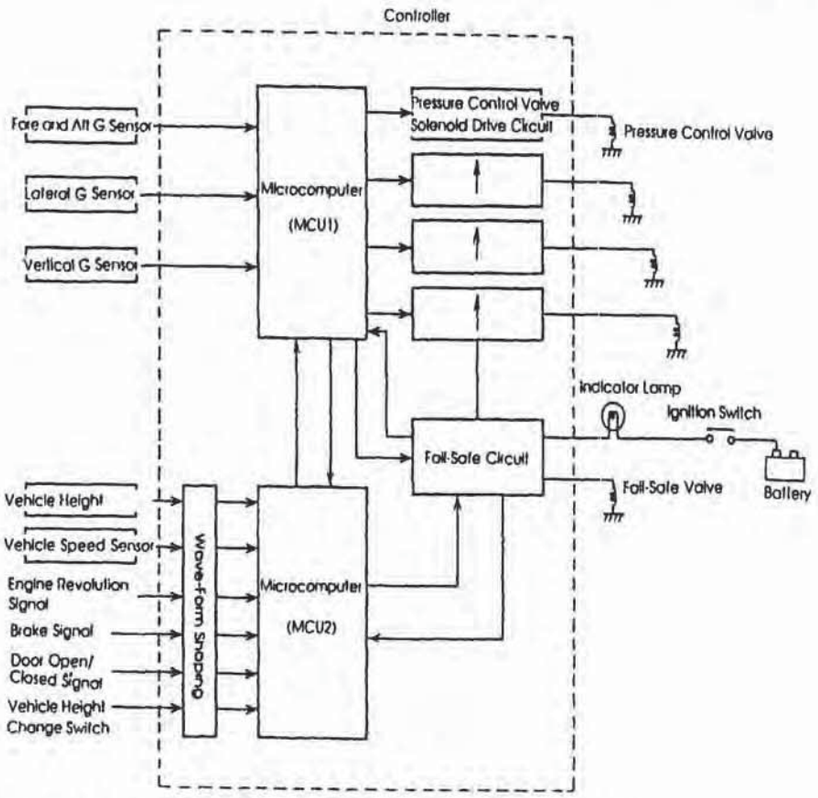


FIGURE 17.15 Interior construction of the electrohydraulic pressure controller.

G Sensors. The G sensors are ball position detection type sensors. They detect changes in the magnetic field caused by the position change of a steel ball as the result of acceleration.

17.4.3 Control System

Roll Control. The inertia force, which causes the car to roll, is detected by the lateral G sensor. Roll control is initiated by increasing the control pressure on the wheels on the outside of the turn and by decreasing the control pressure for the wheels on the inside of the turn. Figure 17.16 shows this operating principle.

The relation between the lateral G and the force generated by the actuator is:

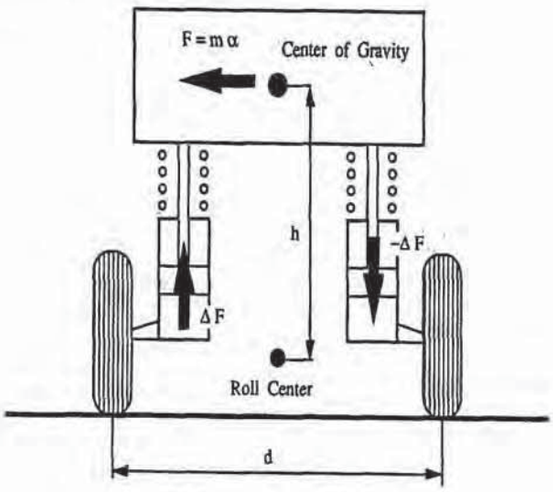
$$F = m\alpha \tag{17.1}$$

$$Fh = \Delta Fd$$

$$\Delta F = \frac{Fh}{d} = \frac{m\alpha h}{d}$$

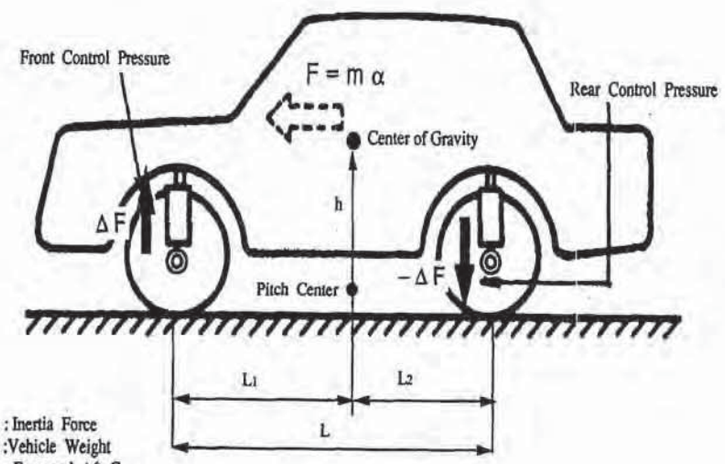
Pitch Control. During braking, inertia is generated at the vehicle's center of gravity and causes pitching. The longitudinal G sensor detects this inertia and cancels it to suppress nose dive by increasing the control pressure to the front and decreasing control pressure to the rear, as shown in Fig. 17.17. The relationship between longitudinal G and the actuator generated force is:

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US PATENT NO. 6,557,540



α : Lateral G
F : Inertia
ΔF : Actuator Generated Force
d : Tread

FIGURE 17.16 Operating principle for roll control.



F : Inertia Force
m : Vehicle Weight
α : Fore and Aft G
ΔF : Actuator - Generated Force
L : Wheel Base

FIGURE 17.17 Pitch control parameters.

$$Fh = \Delta F\,(L_1 + L_2)$$

$$\Delta F = \frac{Fh}{(L_1 + L_2)} = \frac{m\alpha h}{L} \tag{17.2}$$

Bounce Control. The vertical G sensor attached to the vehicle body detects the value for vehicle body acceleration. By integration of the acceleration, the absolute velocity of the body is calculated. A force proportional to the absolute velocity is generated by the pressure control valve.

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US PATENT NO. 6,557,540

This control method, called skyhook damper control, is adapted. It dampens the motion of the car body regardless of any input from the road surface.

In the case of a passive damper, the vertical motion of the body relative to the road surface inputs can be given as:

$$\frac{X_2}{X_1} = \frac{2j\omega_2\xi_2\omega + \omega_2^2}{-\omega^2 + 2j\omega_2\xi_2\omega + \omega_2^2} \tag{17.3}$$

The vibration transmission rate at the resonant point is:

$$\left|\frac{X_2}{X_1}\right|_{\omega=\omega_2} = \sqrt{1 + \frac{1}{4\xi_2^2}} \tag{17.4}$$

and always has a value greater than one.

By contrast, the vibration characteristics of the skyhook damper are given as:

$$\frac{X_2}{X_1} = \frac{2j\omega_2\xi_2\omega + \omega_2^2}{-\omega^2 + 2j\omega_2(\xi_2 + \xi_s)\omega + \omega_2^2} \tag{17.5}$$

The vibration transmission ratio at the resonant point is:

$$\left|\frac{X_2}{X_1}\right|_{\omega=\omega_2} = \frac{\sqrt{4\xi_2^2 + 1}}{2(\xi_2 + \xi_s)} \tag{17.6}$$

hence,

$$\xi_s \geq \sqrt{\xi_2^2 + \frac{1}{4}} - \xi_2 \tag{17.7}$$

and it is possible to reduce the ratio to less than one.

The effects of the hydraulic active suspension are organized in Table 17.7 according to those related to the vehicle.

17.4.4 Effectiveness

Figure 17.18 shows the lateral G and angle of roll during cornering. Figure 17.19 shows the fore and aft G force, angle of nose dive, and squat angle during starts and stops. In either case, the car with the hydraulic active suspension outperformed the other cars with conventional suspensions.

TABLE 17.7 Effect of Active Suspension Control

Control	Vehicle-related effects
Roll control	During transient control of wheel loading, as when changing lanes, the steering characteristics of the vehicle can be optimally controlled. The tires are used to their utmost performance ability because there is minimal roll, minimal change in the camber of the tires to the ground, and because the tires are continually kept in square contact with the road.
Pitch control	Nose dive and tail lift are minimized during braking. Squats are minimized during starts and rapid acceleration.
Bounce control	Vertical vibration of the vehicle is reduced and continuity is improved. Vertical load fluctuation is minimal, and the contact of the tires with the road is greatly improved.

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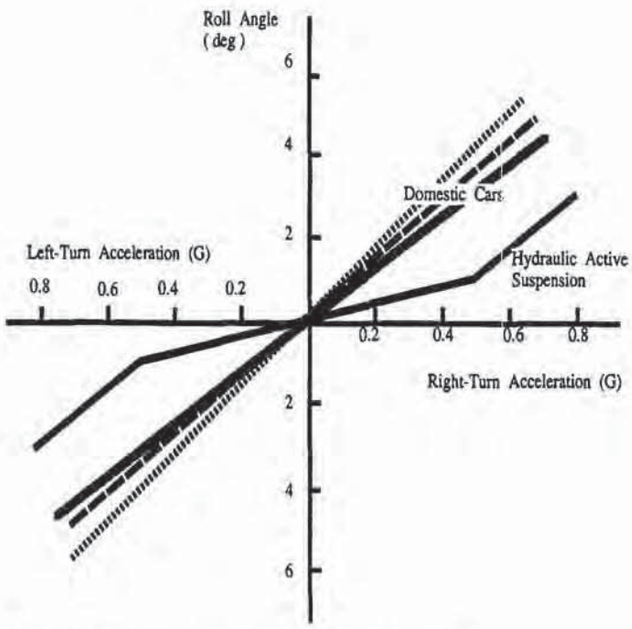


FIGURE 17.18 Comparison of roll angles during cornering.

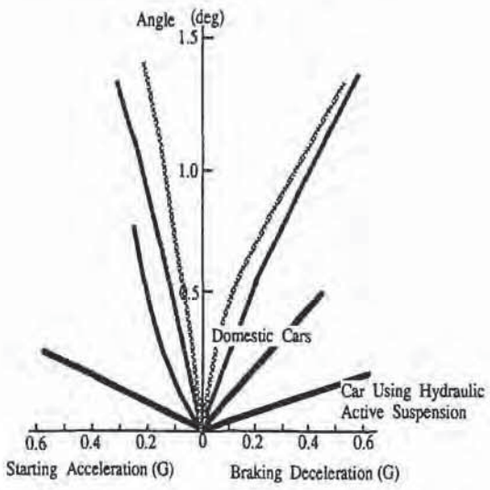


FIGURE 17.19 Comparison of nose dive and squat angles.

Figure 17.20 shows the effects of bounce control for the skyhook damper. Compared with cars using conventional suspension systems, the car with the hydraulic active suspension system exhibited superior performance and low vibration levels.

Figure 17.21 shows the ride characteristics and roll rate for various cars with some suspension systems. As the results clearly demonstrated, the hydraulic active suspension system, through advanced roll control and bounce control, provides a ride and a level of control far superior to that of other suspension systems.

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US PATENT NO. 6,557,540

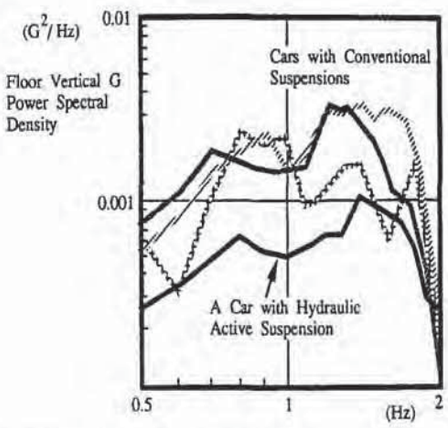


FIGURE 17.20 Comparison of vertical vibrations.

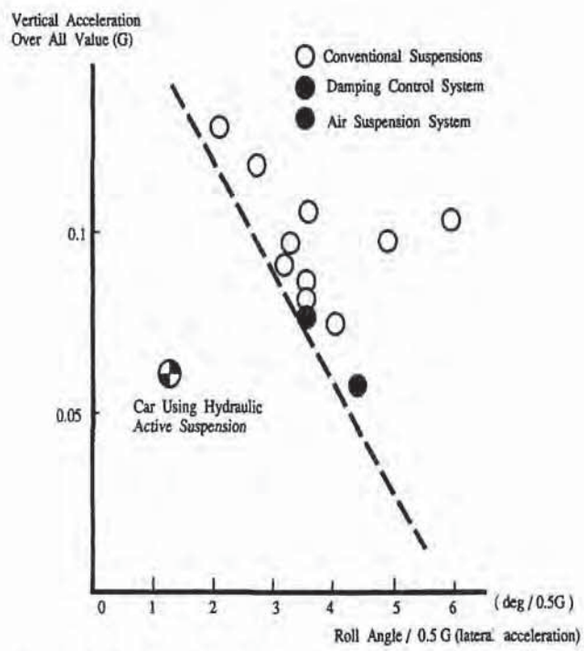


FIGURE 17.21 Ride characteristics and roll rate for various cars with different suspension systems.

17.5 CONCLUSION

The active suspension system provides outstanding levels of performance which are unobtainable with other suspension control systems and conventional passive suspensions. Evaluations made with actual vehicles confirmed the effectiveness of the active suspension system in improving ride comfort and handling properties.

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US PATENT NO. 6,557,540

GLOSSARY

Hydropneumatic suspension A suspension system using oil or air to support the car body.

Skyhook damper control The control law applied to control the vehicle as if it were fixed within absolute space suspended from the sky.

Supersonic sensor A sensor used to measure the distance between a car body and the road surface using supersonic waves.

NOMENCLATURE

C_2: passive damping coefficient
C_s: active damping coefficient
K_1: tire stiffness
K_2: spring stiffness
M_1: unsprung mass
M_2: sprung mass
$\omega_1 = (K_1/M_1)^{1/2}$ natural frequency of unsprung mass
$\omega_2 = (K_2/M_2)^{1/2}$ natural frequency of sprung mass
$\xi_2 = (C_2/2)\,(M_2K_2)^{1/2}$ active damping ratio
$\xi_S = (C_S/2)\,(M_2/K_2)^{1/2}$ passive damping ratio

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PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

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US PATENT NO. 6,557,540

CHAPTER 18
STEERING CONTROL

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The application of electronic control technology to vehicle steering systems is still in the development stage. The structure and functioning of such systems are not yet clearly defined. Accordingly, the material that follows is intended primarily to introduce systems which have already been published, and it is devoted entirely to the electronic aspects of those systems and does not include descriptions of the basic operation of hydraulic power steering systems.

18.1 VARIABLE-ASSIST STEERING

18.1.1 Fundamentals of Electronically Controlled Power Steering

Electronically controlled power steering improves steering feel and power-saving effectiveness, and increases steering performance. It does so with control mechanisms that reduce the steering effort. An electronic control system, for example, may be added to the hydraulic booster or the whole system may be composed of electronic and electric components.

The intent of electronic controls, initially, was to reduce the steering effort when driving at low speeds and to supply feedback for the appropriate steering reaction force when driving at high speeds. In order to achieve those goals, devices such as vehicle speed sensors were used to detect vehicle speed in order to make smooth and continuous changes in the steering assist rate under conditions ranging from steering maneuvers at zero speed to those at high speeds. However, as vehicles became equipped with electrohydraulic systems and fully electronic and electric systems, the emphasis for these systems started to include reduction in power requirements and higher performance.

The main functions required for electronically controlled power steering are listed in Table 18.1.

18.1.2 Types of Electronically Controlled Power Steering

Electronically controlled power steering systems presently available commercially can be classified according to their basic structure and basic principles into three types: hydraulic, hybrid, and full electric systems, as shown in Table 18.2. Detailed explanations of these systems are given as follows.

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TABLE 18.1 Functions Required for Electronically Controlled Power Steering

Reduction of driver's burden when turning the steering wheel and improvement in the steering feel	• Reduction in steering effort • Smoothness of steering operation • Feedback of appropriate steering reaction forces • Reduction of kickback[1] • Improvement in convergence[2] • Creation of other new functions
Power saving	
Failsafe	• Maintaining of manual steering function in the event of any malfunctions

TABLE 18.2 Classification of Electronically Controlled Power Steering System

Basic structure	Control method	Control objects	Sensors				Actuator	Major effects	
			Vehicle speed	Steering torque	Angular velocity	Current		Steering force responsive to vehicle speed	Power saving
Electronically controlled hydraulic system	Flow	Flow supply to power cylinder	O			O	Solenoid	O	O
	Cylinder bypass	Effective actuation pressure given to cylinder	O			O	Solenoid	O	
	Valve characteristics	Pressure generated at control value	O			O	Solenoid	O	
	Hydraulic reaction force control	Pressure acting on the hydraulic reaction force mechanism	O			O	Solenoid	O	
Hybrid system	Flow	Flow supply to power cylinder	O		O	O	Motor	O	O
Full electric system	Current	Motor torque	O	O		O	Motor	O	O
	Voltage	Motor power	O	O	O	O	Motor	O	O

18.1.3 Explanations of Each System

Electronically Controlled Hydraulic System. This system consists of a linear solenoid valve, a vehicle speed sensor, and other electronic devices located in part of the hydraulic circuit of the hydraulic system. The opening of the solenoid valve is controlled based on signals from the vehicle speed sensor. The flow and pressure of the hydraulic fluid is controlled by means of the opening of the solenoid valve.

The assist rate is smoothly and continuously varied in response to the vehicle speed, so that when the vehicle is stationary, the opening of the solenoid valve is small to ensure that the steering effort is appropriately light. When the vehicle is moving at high speed, the opening of the solenoid valve is large to ensure that the steering effort is appropriately heavy.

Flow Control Method. In this method, a solenoid valve is located at the pump discharge port as shown in Fig. 18.1. The electronic control device regulates the solenoid valve opening at high vehicle speeds to reduce the pump discharge volume, thus increasing the required steering effort. By reducing the resistance of the circuit between the pump and the power cylinder, power requirements are reduced. Figure 18.2 shows the position of the solenoid

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US PATENT NO. 6,557,540

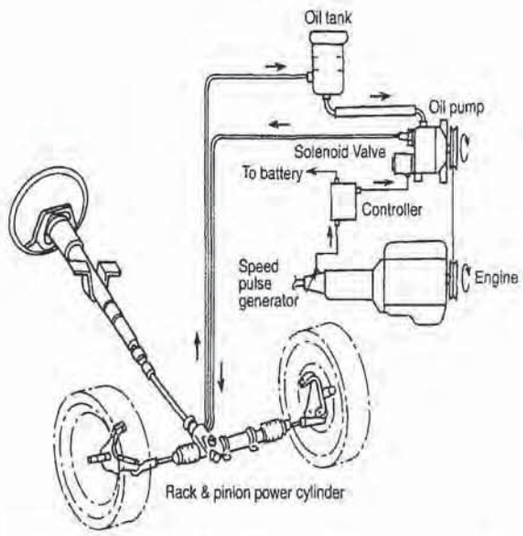
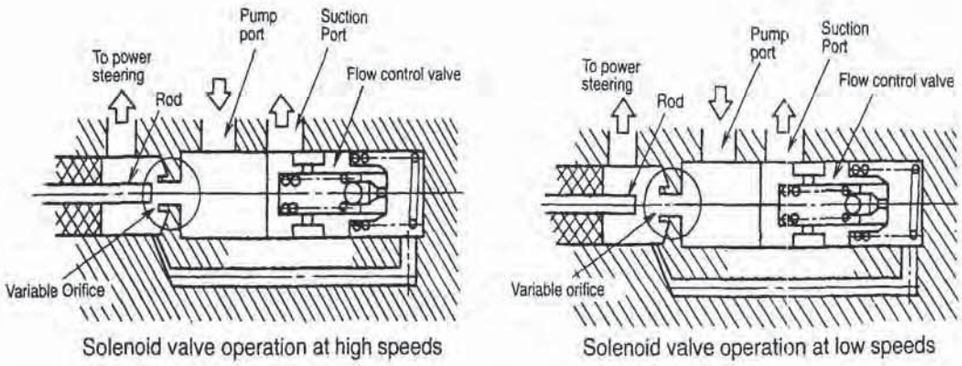


FIGURE 18.1 Vehicle speed-responsive pump discharge flow volume control type.[1]



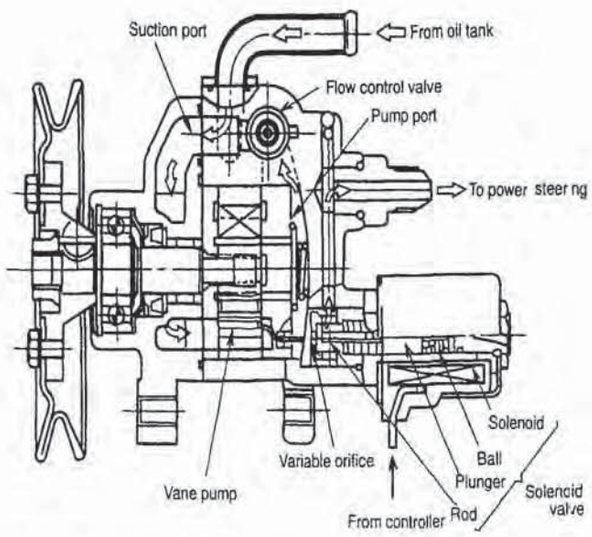


FIGURE 18.2 Structure and operation of pump with solenoid valve.[1]

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valve at the pump, with separate diagrams showing the operation at high and low vehicle speeds.

The flow of hydraulic fluid to the power cylinder is reduced when driving at high speeds, so that, for this method, the magnitude of the steering response rate and the steering reaction force are balanced at a point of equilibrium.

Cylinder Bypass Control Method. In this method, a solenoid valve and a bypass line are located between both chambers of the power cylinder.[2] The opening of the valve is extended by the electronic control equipment in accordance with increases in vehicle speed, thus reducing the hydraulic pressure in the power cylinder and increasing the steering effort. Like the flow control method, this system may also seek the equilibrium point for the steering response rate and the steering reaction force.

Valve Characteristics Control Method. In this method, the pressure control restrictions of the rotary valve (control valve) mechanism, which control the volume and pressure of the hydraulic fluid supplied to the power cylinder, are divided into second and third parts. A fourth part, controlled by means of the vehicle speed signal, is provided in the hydraulic line between the second and third parts as shown in Fig. 18.3. The structure of this system is shown in Fig. 18.4. The steering effort is controlled by carrying out variable control of the fourth part to vary the assist ratio. Because the structure is simple and the flow from the pump to the cylinder is supplied efficiently without waste, this system exhibits a good response rate. Figure 18.5 shows the hydraulic pressure characteristics in the valve characteristics control method with the driving current of the solenoid valve in accordance to the vehicle speed control signal as a parameter. When the current is 0.3 A, the valve is fully open, and this represents the high-speed driving condition.

Hydraulic Reaction Force Control Method. In this method, the steering effort is controlled by means of a hydraulic reaction force mechanism, which is located at the rotary valve (control valve). A hydraulic reaction force control valve increases the hydraulic pressure (reaction pressure) introduced into the hydraulic reaction force chamber in accordance with increases in the vehicle speed. The rigidity of the reaction force mechanism (equivalent spring constant) is variably controlled so as to directly control the steering effort.

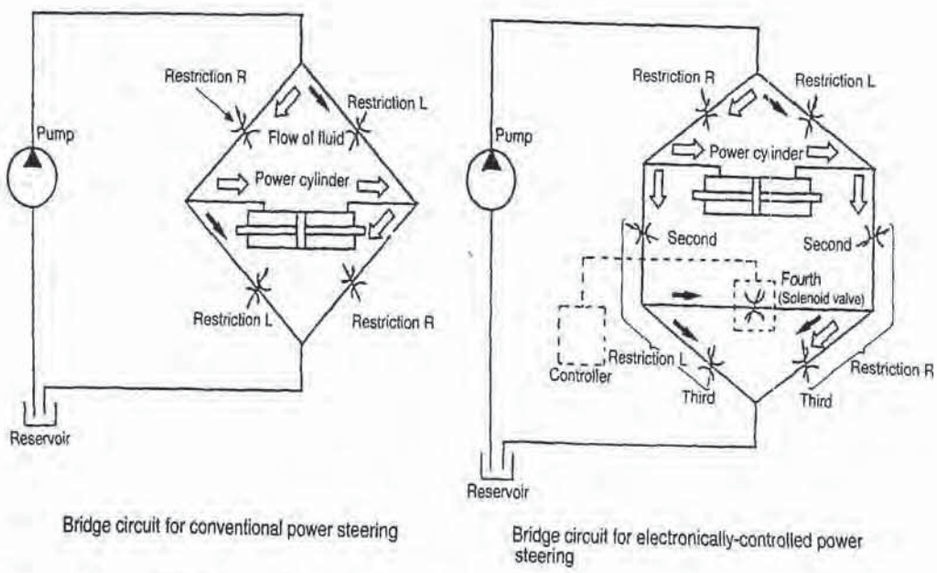


FIGURE 18.3 Closing bridge circuit for control valve.[2]

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US PATENT NO. 6,557,540

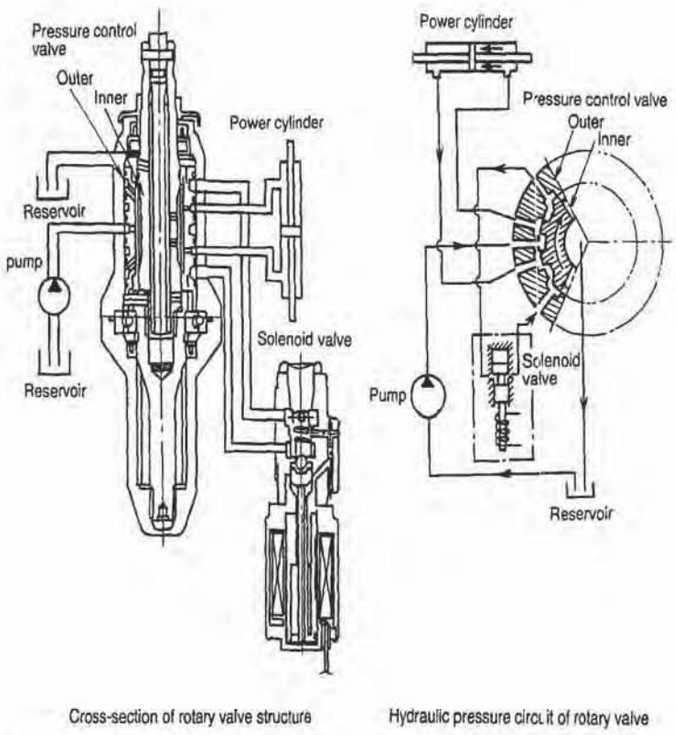


FIGURE 18.4 Valve characteristics control method.[1]

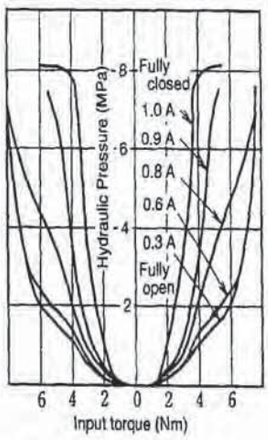


FIGURE 18.5 Valve characteristics in response to vehicle speed.[1]

This method requires the inclusion of a reaction force mechanism, which makes the structure of the control valve more complex, which in turn increases the cost. However, because the rigidity of the reaction force mechanism increases in accordance with increases in the vehicle speed, there is no vagueness in the steering feel in the area around the straightforward steering position. Because this method assigns the steering reaction force irrespective of the volume of hydraulic fluid supplied to the power cylinder, the magnitude of the steering reaction force can be set freely without the need to sacrifice any of the steering response rate.

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US PATENT NO. 6,557,540

18.1.4 Hybrid Systems

Hybrid systems utilize a flow control method in which the hydraulic power steering pump is driven by an electric motor. The steering effort is controlled by controlling the rotating speed of the pump (discharge flow).

The drive efficiency of the generator and motor are low compared to that of the hydraulic pump, which is driven by the vehicle engine. But because any residual flow is not discharged, the power loss is lower than that of the engine pump when driving at high speeds.

Because the pump is not driven by the vehicle engine, there is also a large degree of freedom in the selection of the mounting location for the pump.

Driving Mode Responsive Method. In this method, the control system consists of a vehicle speed sensor, steering angular velocity sensor, an electronic control unit, and a motor driven hydraulic pump, as shown in Fig. 18.6.

As is shown in Fig. 18.7, the driving conditions (such as driving in urban areas, country areas, winding regions, or highways) are automatically judged, and the pump flow rate is controlled in accordance with this condition in order to provide the appropriate steering effort for the driving conditions. Fine control adjustments are achieved by means of this method as compared with vehicle speed-responsive types mentioned previously.

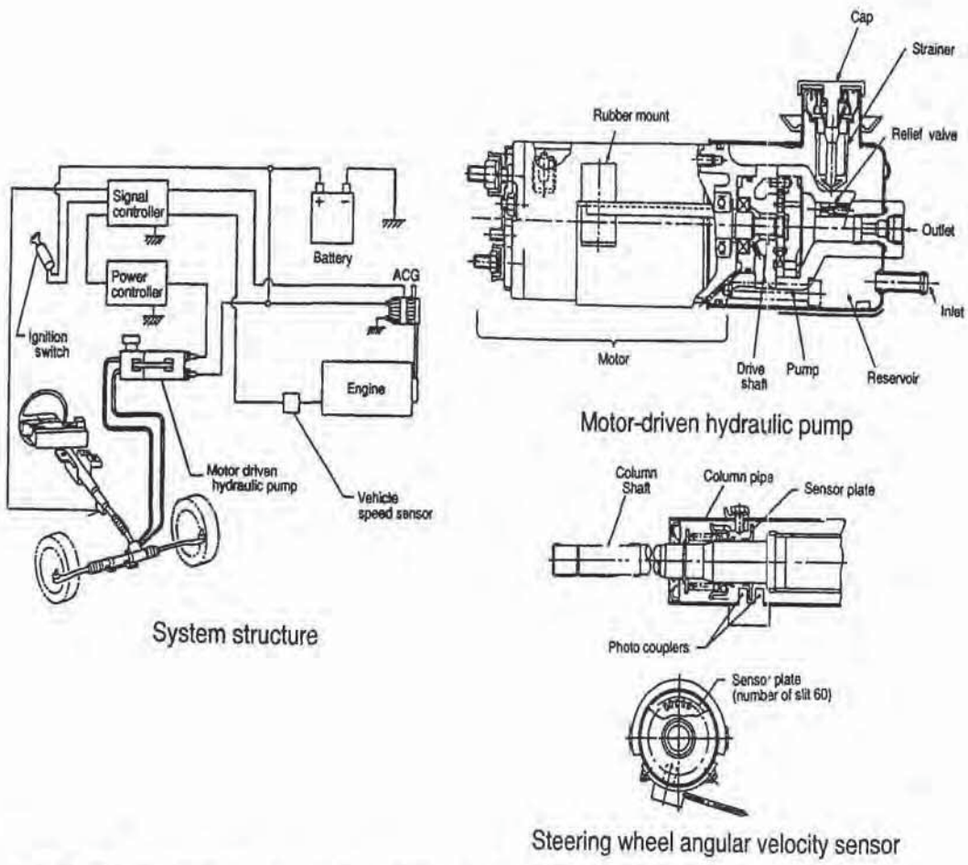


FIGURE 18.6 Driving mode responsive-type hybrid power steering.[3]

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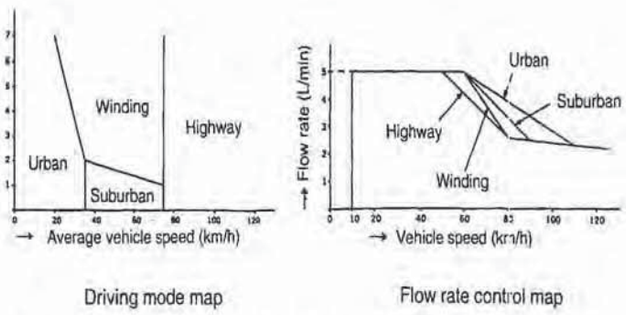


FIGURE 18.7 Driving mode and pump flow rate.[3]

Steering Wheel Speed Responsive Method. For this method, the system consists of components such as a vehicle speed sensor, steering wheel angular velocity sensor, an electronic control unit, and a motor-driven hydraulic pump, as shown in Fig. 18.8. The pump flow volume is controlled in accordance with the angular velocity of the steering wheel and the vehicle speed as shown in Fig. 18.9. As mentioned previously, the discharge flow volume of the pump is reduced and the steering response drops when the vehicle is driven at high speeds. Therefore, in this system, the speed of the motor is increased in accordance with the detected angular velocity of the steering wheel in order to increase the discharge flow volume to solve the problem. Accordingly, losses in power resulting from the circulation of residual flow within the system are kept to the minimum possible level, and the magnitude of the reaction force can be controlled freely without sacrificing any of the steering response rate.

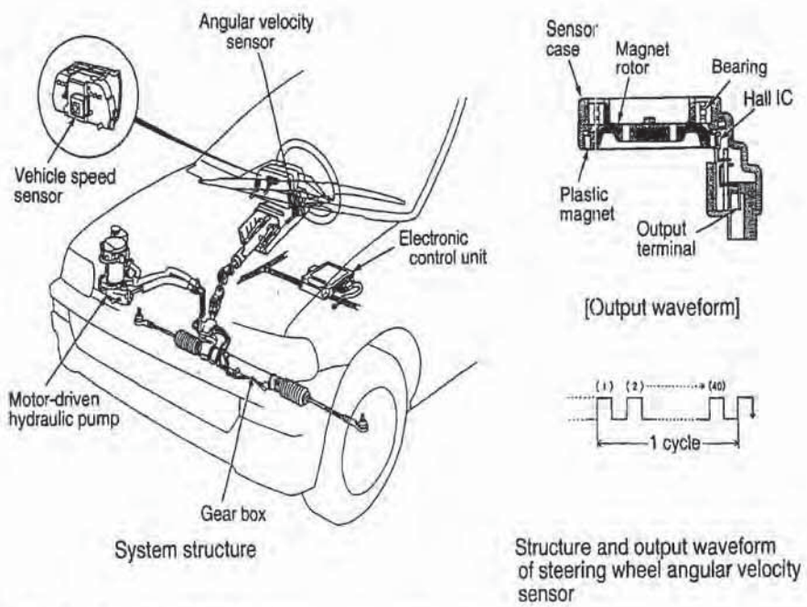


FIGURE 18.8 Steering speed-responsive-type hybrid power steering.[4]

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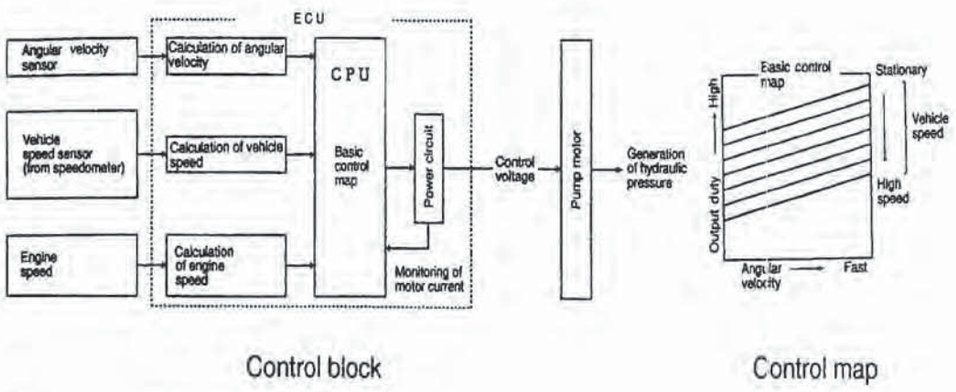


FIGURE 18.9 Basic control.[4]

18.1.5 Electric Power Steering

Electric power steering (EPS) is a fully electric system, which reduces the amount of steering effort by directly applying the output from an electric motor to the steering system. This system consists of vehicle speed sensors, a steering sensor (torque, angular velocity), an electronic control unit, a drive unit, and a motor, as shown in Fig. 18.10. Signal outputs from each sensor are input to the electronic control unit, where the necessary steering assistance is calculated and applied by the drive unit to control the operation of the motor.

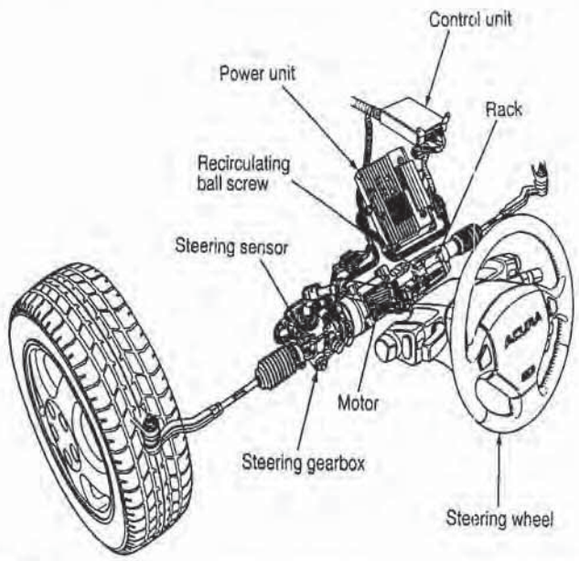


FIGURE 18.10 Structure of EPS System[5] (rack assist-type ball screw drive).

Because the motor output is controlled directly in this system, the setting range for the steering effort is large, and also because it is possible to supply only the amount of power that is necessary when the steering wheel is turned, a large reduction in power requirements can be effectively achieved with no power losses. This means, in contrast to hydraulic systems, that it is not necessary for the pump to keep operating continuously when the steering wheel is not being turned.

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US PATENT NO. 6,557,540

TABLE 18.3 Classification of Motor Drive Mechanism in EPS

Method	Motor drive method	Power transmission mechanism	Figure
Pinion assist	Column shaft drive	Motor→worm gear→column shaft→pinion shaft	18.11
	Pinion shaft drive	Motor→gear train→pinion shaft	18.12
Rack assist	Another shaft pinion drive	Motor→planetary gear train→another shaft pinion→rack shaft	
	Ball screw drive	Motor→ball screw→rack shaft	18.10

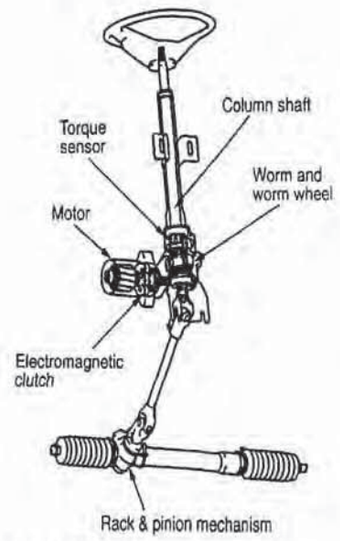


FIGURE 18.11 Column shaft drive method.[1]

In rack-and-pinion steering mechanisms, the EPS system applies the motor power to the pinion gear shaft or to the rack shaft. Several reduction gears are incorporated to amplify the torque of the motor. This system can be classified according to the drive method as given in Table 18.3 and Figs. 18.10 to 18.12. The maximum amount of assist, the smoothness of the steering feel, and the degree of noise occurring during steering are, by and large, determined by the power transmission systems in this table. In general, it is possible to obtain a greater amount of assist from the rack assist method than from the pinion assist method, and the rack assist method is optimal for vehicles in which the front axle load is high.

Details of the respective sensors, controls, and the results achieved thereby are given as follows under common headings.

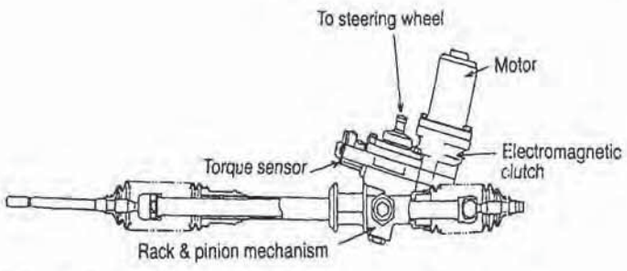


FIGURE 18.12 Pinion shaft drive method.[1]

Sensors. The EPS system utilizes a variety of sensors to control the motor. These sensors include a torque sensor, which detects the steering effort of the steering wheel; a steering wheel angular velocity sensor, which detects the angular velocity of the steering wheel; a battery sensor, which detects the battery voltage; a current sensor, which detects the motor current and the battery current; and a vehicle speed sensor. Of these sensors, the torque sensor and the steering wheel angular velocity sensor, which form the core of the EPS system, are described as follows. (Also see Chap. 3.)

Torque Sensor. The pinion shaft in the rack-and-pinion steering mechanism is divided into two sections of input shaft and pinion gear. The torque sensor comprises a torsion bar

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US PATENT NO. 6,557,540

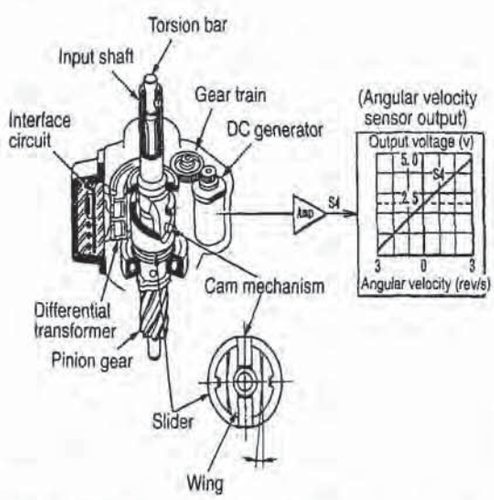


FIGURE 18.13 Torque sensor and angular velocity sensor.[5]

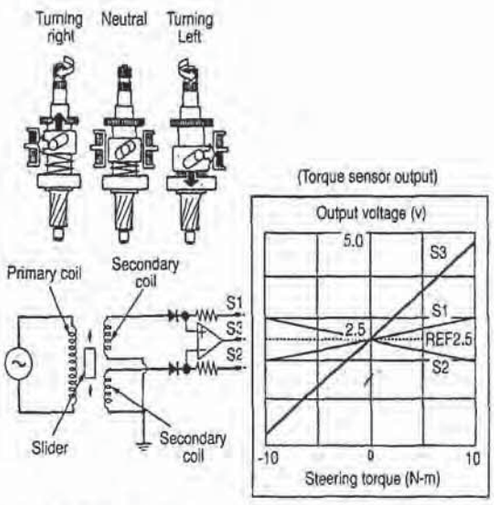


FIGURE 18.14 Diagram of torque sensor operation.[5]

that connects the two sections, a slider with a movable iron core, a cam mechanism that converts the twist torque of both sections of the shaft into an axial direction displacement, and a differential transformer that converts the axial direction displacement of the slider into an electric signal.

The structure is shown in Fig. 18.13. Figure 18.14 shows examples of the detection of the magnitude and direction of the slider displacement. Signal S3 is the output signal of torque and S1 and S2 are outputs for diagnosis use. The differential transformer-type torque sensor has a dual electric structure to provide differential outputs, so that a high sensing accuracy and good temperature characteristics can be obtained, and detection of failure can also be made accurately.

In addition to this type of torque sensor, there are other torque sensors that use a potentiometer instead of a differential transformer, and also types in which the relative displacement in the turning direction is measured in a noncontact manner using a coil, without there being a mechanically movable part such as the slider.

Angular Velocity Sensor. The angular velocity sensor consists of a gear train, which is located around the input shaft, and a dc generator, which is driven at an increase of speed by this gear train. The structure is shown in Fig. 18.13. The turning direction and angular velocity of the steering wheel are detected by the turning direction and angular velocity of this dc generator. Signal S4 indicates the output from the steering wheel angular velocity sensor.

Electronic Control Unit (ECU). The ECU consists of an interface circuit that coordinates the signals from the various sensors, an A/D converter and a PWM unit that are all built into an 8-bit one-chip microprocessor, a watchdog timer (WDT) circuit that monitors the operation of this microprocessor, and a PWM drive circuit that drives the power unit mentioned previously. The ECU conducts a search for data according to a table lookup method based on the signals input from each sensor and carries out a prescribed calculation using this data to obtain the assist force.

In addition, trouble diagnosis for the sensors and the microprocessor is also carried out. When a problem is detected, power to the motor is interrupted, an indicator lamp illuminates, and the problem condition is memorized. Then this problem mode flashes on a display as necessary.

Power Unit. The power unit comprises a power MOSFET (metal oxide semiconductor field effect transistor) bridge circuit which drives the motor in a forward or reverse direction, a drive circuit which controls the respective power MOSFET of this power MOSFET bridge circuit, a current sensor, and a relay which turns the motor current ON and OFF.

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US PATENT NO. 6,557,540

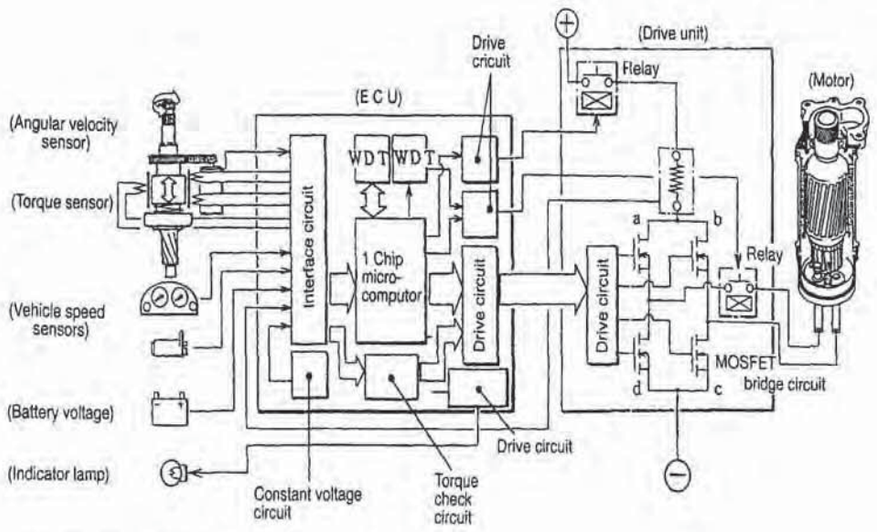


FIGURE 18.15 ECU and drive unit.[6]

The structure is shown in Fig. 18.15. The motor is driven based on instructions from the ECU. The current at this time is monitored by the ECU, and the power supplied to the motor is interrupted in the event of an abnormality.

Depending on the magnitude of the current, some systems are provided with an integrated ECU and power unit, while other systems have each section separate.

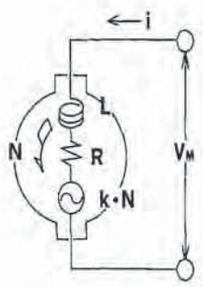


FIGURE 18.16 Motor equivalent circuit.[6]

Motor Control Methods. In the equivalent circuit of the motor, which is shown in Fig. 18.16, the relationship between the terminal voltage V_M, the impedance L, the resistance R, the induced voltage constant k, the revolution speed N, the current i, and the time t, is expressed by the following equation:

$$V_M = L(di/dt) + R \cdot i + k \cdot N \quad (18.1)$$

$$\doteqdot R \cdot i + k \cdot N \quad (18.2)$$

And it is known that the current **i** is proportional to the motor torque T_M.

As can be understood from Eq. (18.2), there are two control methods. In the motor current control method (refer to Fig. 18.17), the target motor current I_T, which is proportional to the motor assist torque T_M, is determined from the signal output T from the torque sensor, and control is performed so that there is no difference between this target current value I_T and the value detected through feedback from the current sensor I_M.

In the motor voltage control method (refer to Fig. 18.18), the motor voltage component ($V_{M1} = R \cdot i = k_T \cdot T_M$; k_T is a proportional constant) which corresponds to the motor assist torque as calculated from the output signal T from the torque sensor, and the motor voltage component ($V_{M2} = k \cdot N$) which corresponds to the motor speed as calculated from the output signal $\dot{\theta}_1$ from the steering wheel angular velocity sensor. These two voltage components are then added and output.

Current Control Method. In this method, the target value for the motor current, which corresponds to the motor assist torque, is set so that it is equal to the vehicle speed response type derived from the signal of the vehicle speed sensor.

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US PATENT NO. 6,557,540

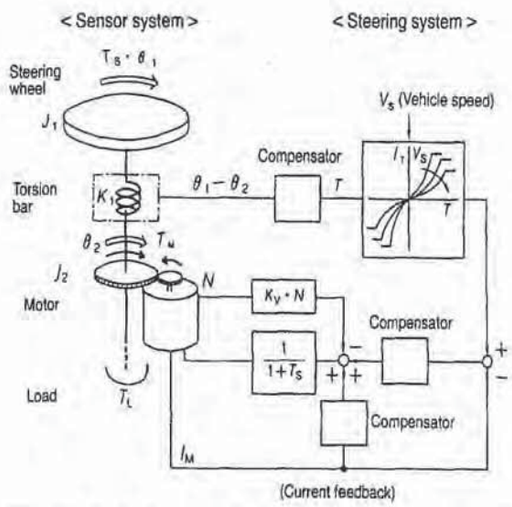


FIGURE 18.17 Diagram of principle of motor current control method.[7]

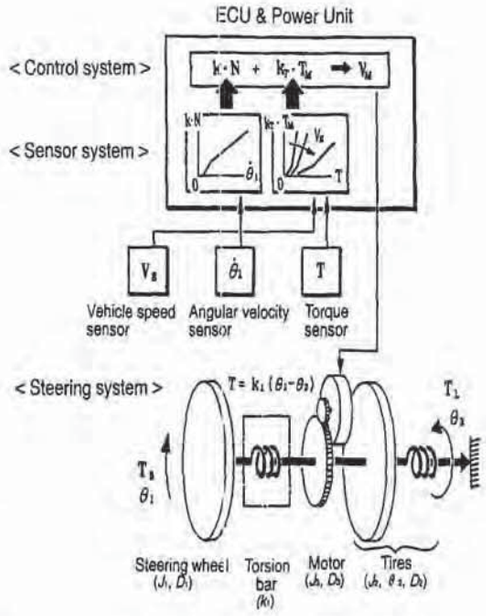


FIGURE 18.18 Diagram of principle of motor voltage control method.[8]

Voltage Control Method. In this method, both the motor torque and the motor speed can be controlled by the output from the torque sensor and the steering wheel angular velocity sensor. When the vehicle is traveling at low speeds, *normal control* is carried out. With this control, the value for $(R \cdot i + k \cdot N)$ in Eq. (18.2) is output to the motor to obtain a good motor response rate (steering response rate) and thus provide a comfortable steering performance. When the vehicle is traveling at high speeds, it is possible to carry out two types of control. In the first method, *return control*, the value for $(k \cdot N)$ is made smaller so that a damping torque which is proportional to the motor speed is generated. In the second method, *damper control*, motor torque is generated in the opposite direction of motor rotation with $V = 0$ when the steering wheel is released in turning.

Normal Control. This is a method of drive control for steering with a reduced steering effort and a good steering response rate. As is shown in Table 18.4, when the steering wheel is turned to the right, FET (a) is ON at the same time that FET (c) is carrying out PWM drive, and current flows to the FET bridge circuit as shown in Fig. 18.19 based on the value of V_M in Eq. (18.2). If the angular velocity of the steering wheel increases, the PWM duty also increases.

An example of the steering characteristics in an actual vehicle can be seen in Fig. 18.20. Figure 18.20*a* represents the assist characteristics during stationary swing, with the dotted line representing the steering characteristics when no assist force is provided. Figure 18.20*b* is the steering response rate; Fig. 18.20*c* is the steering characteristics corresponding to the vehicle speed; and Fig. 18.20*d* is the steering characteristics relative to the lateral acceleration of the vehicle.

TABLE 18.4 FET Drive During Normal Control[5]

Steering condition	FET (a)	FET (b)	FET (c)	FET (d)	Motor operation
Steering to right	ON	OFF	PWM	OFF	Operates in direction steering to the right
Straight ahead	OFF	OFF	OFF	OFF	Stops
Steering to left	OFF	ON	OFF	PWM	Operates in direction steering to the left

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US PATENT NO. 6,557,540

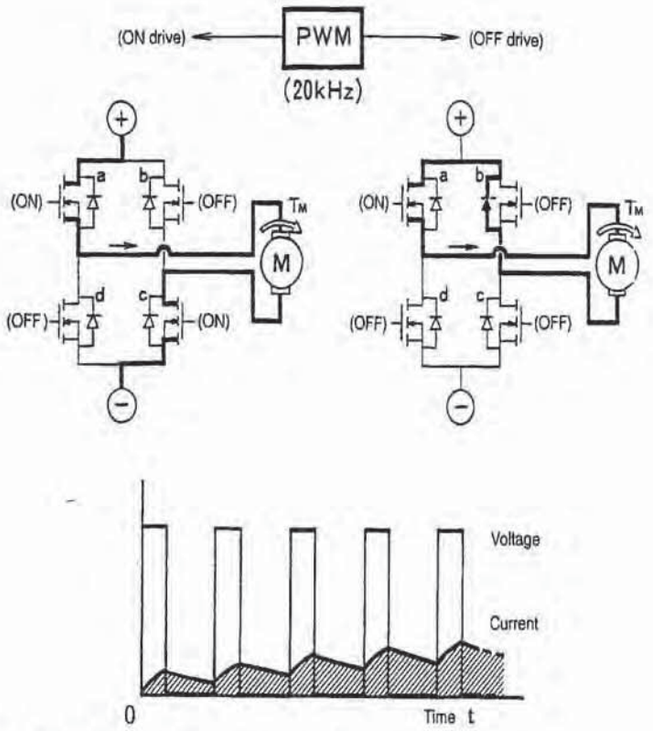


FIGURE 18.19 Diagram of bridge circuit operation.[6]

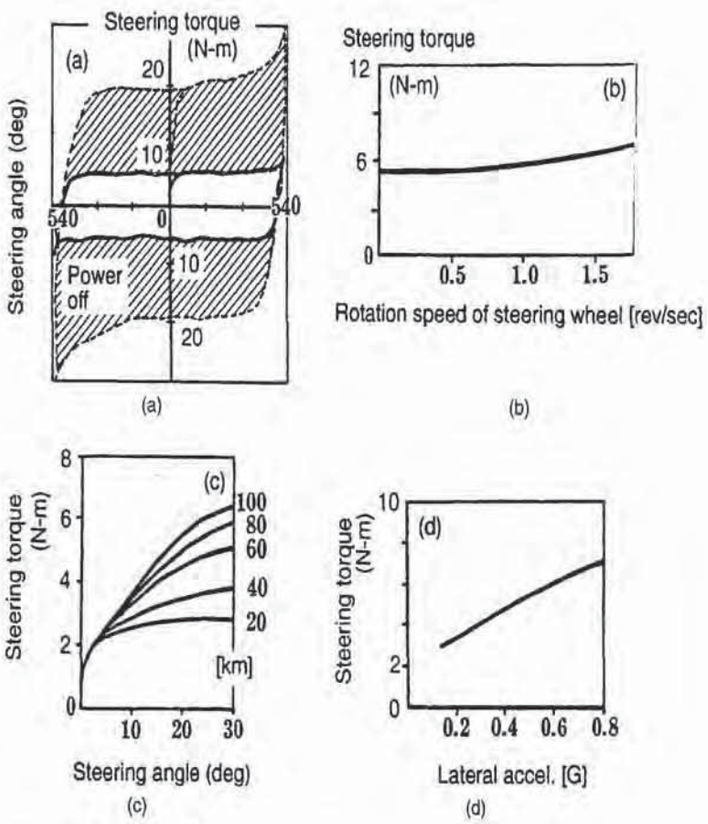


FIGURE 18.20 Steering characteristics.[8]

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Return Control. This method of drive control is for varying the steering wheel return characteristics. When the driver is returning the steering wheel to the neutral position at low-speed driving, the motor current is immediately reduced in order to make the motor operate in the reverse direction to the torque generating directions. As a result, a good returnability of the steering wheel should be obtained. At high-speed driving, motor current is gradually reduced in order to suppress returnability and obtain more stable steering characteristics.

As shown in Table 18.5, when the steering wheel returns to the straightforward position after being turned to the right, FET (a) carries out PWM-r drive based on the signals from the steering wheel angular velocity sensor, and, at the same time, FET (c) also carries out PWM drive based on the signals from the torque sensor.

TABLE 18.5 FET Drive During Return Control[5]

Steering condition	FET (a)	FET (b)	FET (c)	FET (d)
Return from right steering to straight ahead	PWM-r	OFF	PWM	OFF
Return from left steering to straight ahead	OFF	PWM-r	OFF	PWM

The results obtained by means of this return control are shown in Fig. 18.21. Figure 18.21*a* represents the steering characteristics when return control is not applied, and Fig. 18.21*b* represents the steering characteristics when return control is operating. The hysteresis is greater and the returnability is weaker than in Fig. 18.21*a*.

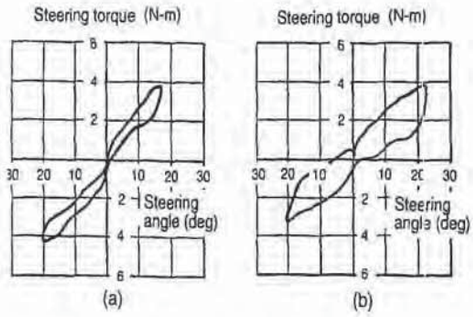


FIGURE 18.21 Comparison of steering wheel return characteristics.[6]

Damper Control. This method of drive control is for improving the convergence of the steering wheel when the vehicle is traveling at high speeds and for eliminating wandering of the steering wheel caused by the tire inputs.

As shown in Fig. 18.22, when the motor terminals are shorted, it is possible to generate motor torque in the reverse direction in proportion to the speed of the motor as is shown by the equation in Fig. 18.22, and this characteristic is utilized for control. Figure 18.23 shows the results of the application of damper control during a convergence test carried out at a vehicle speed of 120 km/h.

18.1.6 Power-Saving Effectiveness

Because the EPS system is a "power-on-demand" system, which supplies only the necessary amount of power at the necessary time, almost no power losses occur when the steering wheel is not being turned. Because of this, the system has extremely high fuel efficiency. Table 18.6 shows the results of measurements of fuel consumption during different driving modes to compare EPS with the hydraulic system. The LA-4 mode corresponds to urban road driving.

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US PATENT NO. 6,557,540

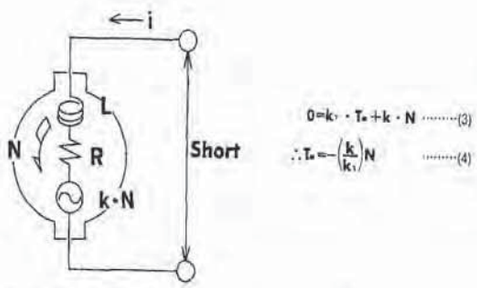


FIGURE 18.22 Principle of damper control.

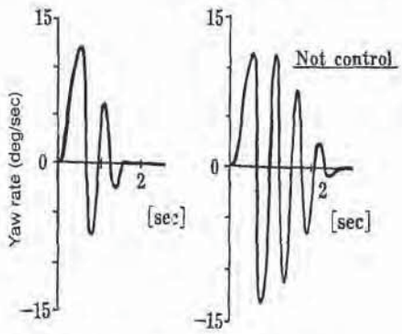


FIGURE 18.23 Effect of damper control.

18.1.7 Trends in Research and Development

The demands for faster speed, higher quality, and reduced power requirements in vehicles is continually increasing. In order to respond to these demands, research and development is under way on the application of electronic control systems with the aim of further improving functions and performance. Features that are being proposed include the introduction of fuzzy logic and the application of power steering, which responds to the driving environment by varying the assist amount in accordance with the traffic conditions or the road surface conditions in order to provide steering feel to fit the sensitivities of human operators. The most important of these is probably active reaction power steering, which provides feedback to the driver regarding the behavior of the vehicle in the form of steering reaction force. Such a system provides the driver with information regarding the operating conditions of the vehicle, for instance, the yaw velocity and/or lateral acceleration, as steering reaction forces. Not only would it improve the relationship between the driver and the vehicle to make it possible to achieve a steering feel that better suits the sensitivities of the driver, but a function that automatically compensates for irregularities in vehicle behavior caused by disturbances could also be expected.

Figure 18.24 is a system control block diagram in which the yaw rate is fed back as a steering reaction force.

Figure 18.25 shows an example of the effect of suppressing irregularities in vehicle behavior caused by side disturbances by comparing the amount of lateral removal when braking on a rut of road surface with the case for conventional power steering.

18.2 FOUR-WHEEL STEERING SYSTEMS (4WS)

For vehicles with extremely long wheel bases and vehicles which need to be operated in narrow places, the concept of a four-wheel steering system is attractive. In such systems, the rear wheels are turned in the opposite direction to the steering direction of the front wheels in

TABLE 18.6 Measurement Results on Mode Fuel Consumption[6]

			Improvement in fuel consumption	
Mode	EPS (mpg)	Hydraulic power steering (mpg)	Fuel consumption difference (mpg)	Improved rate (%)
LA-4	20.51	20.01	0.80	2.50
Highway	28.88	28.18	1.14	2.52

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US PATENT NO. 6,557,540

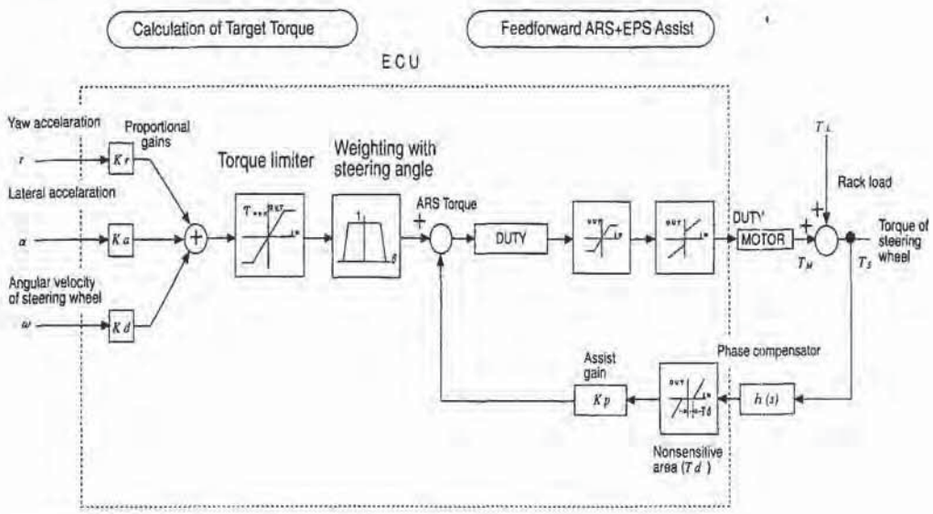


FIGURE 18.24 Block diagram of active reaction control.

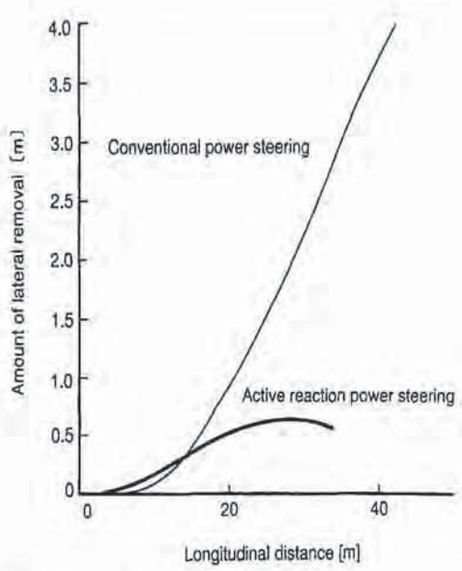


FIGURE 18.25 Effect of active reaction power steering.

order to make the turning radius as small as possible and to improve the handling ability. Such 4WS systems have been under development for some time. However, the concept of the system being used in passenger vehicles for the purpose of improving vehicle stability and steering response at medium to high speeds is relatively new.

A 4WS system for passenger cars has the following two aims:

- Reducing the turning motion (yawing) of the vehicle by steering the rear wheels in the same direction as the front wheels, thus improving the vehicle stability at high speeds
- Improving the steering response at medium speeds, while at the same time reducing the turning circle radius at low speed, by steering the rear wheels in the opposite direction to the front wheels

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US PATENT NO. 6,557,540

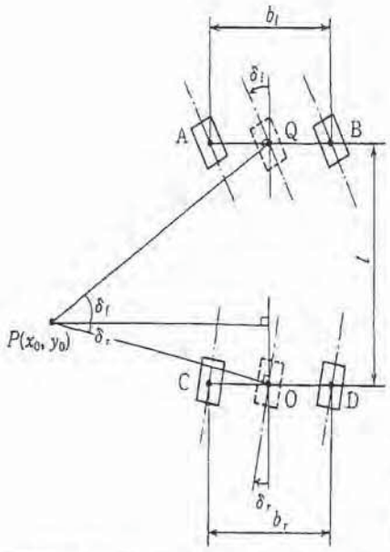


FIGURE 18.26 Half vehicle model of 4WS.[1]

18.2.1 Basic Principles of 4WS

Shortening the Minimum Turning Radius. As shown in Fig. 18.26, provided the origin point of the coordinate is set at the center of the rear tread, the coordinates of the turning center of the vehicle $P(X_0, Y_0)^v$ when the rear wheels are turned in the opposite direction from the front wheels are given in the following equations:

$$x_0 = \iota / (\tan\delta_f + \tan\delta_r) \tag{18.3}$$

$$y_0 = \iota\,(\iota \cdot \tan\delta_1) / (\tan\delta_f + \tan\delta_r) \tag{18.4}$$

And, if the turning radius for the front outer wheel is R and the difference between the turning radius of the front and rear outer wheels is ΔR, then:

$$R = \overline{BP} = \sqrt{\left(\frac{b_f}{2} + X_0\right)^2 + (\iota - y_0)^2} \cdot \sqrt{\left(\frac{b_f}{2} + \frac{\iota}{\tan\delta_f + \tan\delta_r}\right)^2 + \left(\frac{\iota \cdot \tan\delta_f}{\tan\delta_f + \tan\delta_r}\right)^2} \tag{18.5}$$

$$\Delta R = \overline{AP} - \overline{CP}$$

$$= \sqrt{\left(-\frac{b_f}{2} + X_0\right)^2 + (\iota - y_0)^2} - \sqrt{\left(-\frac{b_r}{2} + X_0\right)^2 + y_0^2}$$

$$= \sqrt{\left(-\frac{b_f}{2} + \frac{\iota}{\tan\delta_f + \tan\delta_r}\right)^2 + \left(\frac{\iota \cdot \tan\delta_f}{\tan\delta_f + \tan\delta_r}\right)^2}$$

$$- \sqrt{\left(-\frac{b_r}{2} + \frac{\iota}{\tan\delta_f + \tan\delta_r}\right)^2 + \left(\frac{\iota \cdot \tan\delta_f}{\tan\delta_f + \tan\delta_r}\right)^2} \tag{18.6}$$

In these equations, δ_f is the front wheel steering angle (average of left and right wheels), δ_r is the rear wheel steering angle (average of left and right wheels), b_f is the tread of the front wheels, b_r is the tread of the rear wheels, and ι is the wheelbase.

$$R = \overline{PQ} = \frac{\iota}{\sin\delta_f + \cos\delta_f + \tan\delta_f} \tag{18.7}$$

$$\Delta R = \overline{PQ} - \overline{PO} \tag{18.8}$$

$$= \frac{\iota}{\sin\delta_f + \cos\delta_f + \tan\delta_f}$$

$$- \frac{\iota}{\sin\delta_f + \cos\delta_f + \tan\delta_f}$$

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US PATENT NO. 6,557,540

It can be seen from Eq. (18.7) that when the rear wheels steer in the opposite direction as the front wheels, the turning radius becomes smaller than when the rear wheels are not steered ($\delta_r = 0$). In addition, it can be seen from Eq. (18.8) that when the steering amount for the front and rear wheels is the same ($\delta_f = \delta_r$), then it becomes possible to obtain a difference of 0 between the turning radius of the front and rear wheels

Improvement in Stability and Maneuverability When Driving at Medium to High Speeds. The steering characteristics in yaw velocity and lateral acceleration of the vehicle with 4WS in which the rear wheels are steered in proportion to the front wheels steering angle in the same direction, are shown in Table 18.7, along with that for a vehicle yaw 2WS system. However, the half-car vehicle model shown in Fig. 18.26 is used as the vehicle model.

From Table 18.7, in a system in which the rear wheels are turned in the same direction as the front wheels, the stability factor K, the damping ratio ζ, and the natural oscillation of yawing ωn are not different from the values for conventional vehicles with 2WS, so that the intrinsic stability of the vehicle will not vary. On the other hand, because the lateral acceleration response delay will become smaller as the coefficients of numerators s and s^2 increase and the steady state gain in the yaw velocity will drop by the proportion to $(1 - k)$, the yawing movement which accompanies the lateral movement of the vehicle will become less and the stability within the range of practical use will be improved. If, however, the rear wheels are steered in the opposite direction to the front wheels, $k < 0$ and so $1 - k > 1$. This means that the steady state gain of the yaw velocity will increase, with the result that the steering response will be improved. The symbols used in Table 18.7 and Fig. 18.27 are explained in Table 18.8.

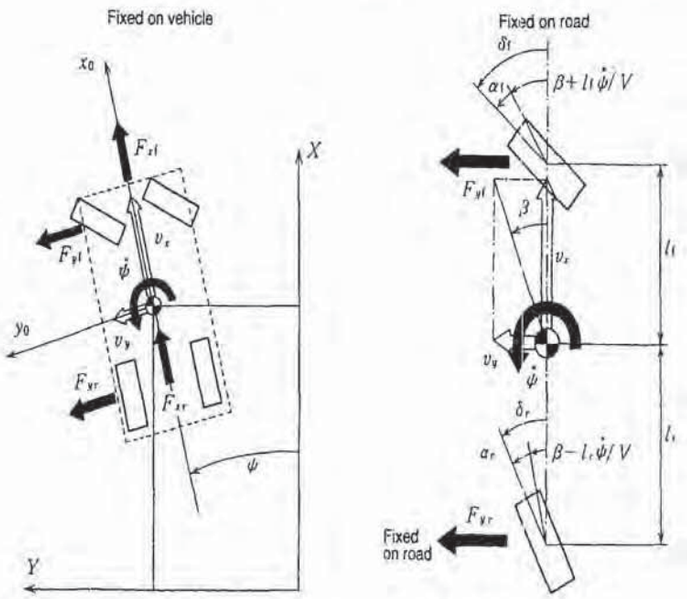


FIGURE 18.27 Coordinates fixed on road and vehicle in half vehicle model.[1]

18.2.2 Classification of 4WS

Four-wheel steering systems that are currently being implemented in vehicles are classified according to their functions and mechanisms. The aims and characteristics of each system are briefly explained in Tables 18.9 and 18.10. Because the fully mechanically controlled system shown in Table 18.10 has a low degree of control freedom, there is less tendency for it to be used. Regardless of its low cost, however, this system provides every basic function of 4WS, so that there is a possibility that it may be used in the future, primarily in smaller vehicles.

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US PATENT NO. 6,557,540

TABLE 18.7[1]

Symbol	Name	
K	Stability factor (s^2/m^2)	
A	Steady state gain in yaw velocity	(1/s)
B_f, B_r	Steady state gain vehicle side slip angle for front and rear	(s)
T_f, T_r	Time constant for front/rear yaw velocity	(s)
τ_f, τ_r	Time constant for front/rear side slip angle	(s)
ω_n	ω_n : Response frequency	(rad/s)
	$f_n(\omega_n/2\pi)$: Natural frequency	(Hz)
$\zeta \cdot \omega_n$	$\zeta \cdot \omega_n$:	(1/s)
	$\zeta \cdot \omega_n \times V$: Steering capacity	(m/s^2)
	ζ : Damping rate	

	Vehicle with 2WS	Vehicle with 4WS in which the rear steering angle is proportional to front
Stability factor	$\dfrac{m(l_r K_r - l_f K_f)}{2l^2 K_f K_r} = K$	K
Steady state gain in yaw velocity	$\dfrac{1}{1+KV^2} \cdot \dfrac{V}{l}$	$\dfrac{1}{1+KV^2} \cdot \dfrac{V}{l}$
Damping rate	$\dfrac{1}{2} \cdot \dfrac{(k_f + K_r I_z + (K_f l_f + K_r l_r^2)m}{\sqrt{m I_z K_f K_r l^2 (1+KV^2)}} = \zeta$	ζ
Response frequency	$\dfrac{2l}{V} \cdot \sqrt{\dfrac{K_f K_r (1+KV^2)}{I_z m}} = \omega_n$	ω_n
Steering capacity	$\dfrac{1}{V}\left(\dfrac{K_f + K_r}{m} + \dfrac{K_f l_f^2 + K_r l_r^2}{I_z}\right) = \zeta \cdot \omega_x$	$\zeta \cdot \omega_n$
$\dfrac{\dot{\varphi}}{\delta_H}$	$\dfrac{1}{i_s} G_\varphi(0) \dfrac{1 + T_f s}{1 + \dfrac{2\zeta}{\omega_n} s + \dfrac{1}{\omega_n^2} s^2}$ $G\varphi(0) = \dfrac{1}{1+KV^2} \cdot \dfrac{V}{l}$	$\dfrac{1-k}{i_s} G\varphi(0) \dfrac{1 + (1+\lambda\phi) T_f s}{1 + \dfrac{2\zeta}{\omega_n} s + \dfrac{1}{\omega_n^2} s^2}$ $G\phi(0) = \dfrac{1}{1+KV^2} \cdot \dfrac{V}{l}$ $\lambda\varphi = \dfrac{k}{1-k} \cdot \dfrac{T_f - T_r}{T_f}$
$\dfrac{\alpha_y}{\delta_H}$	$\dfrac{1}{i_s} G_{ay}(0) = \dfrac{1 + T_{ay1} s + T_{ay2} s^2}{1 + \dfrac{2\zeta}{\omega_n} s + \dfrac{1}{\omega_n^2} s^2}$ $G_{ay}(0) = \dfrac{1}{1+KV^2} \cdot \dfrac{V^2}{l}$ $T_{ay1} = \dfrac{l_r}{V}, \quad T_{ay2} = \dfrac{l_z}{2K_f l}$	$\dfrac{1-k}{i_s} G_{ay}(0) \cdot \dfrac{1 + (1+\lambda_{ay1}) T_{ay1} s + (1+\lambda_{ay2}) T_{ay2} s^2}{1 + \dfrac{2\zeta}{\omega_n} s + \dfrac{1}{\omega_n^2} s^2}$ $G_{ay}(0) = \dfrac{1}{1+KV^2} \cdot \dfrac{V^2}{l}$ $\lambda_{ay1} = \dfrac{k}{1-k} \cdot \dfrac{T_{ay1} + T'_{ay1}}{T_{ry1}}$ $\lambda_{ay2} = \dfrac{k}{1-k} \cdot \dfrac{T_{ay2} + T'_{ay2}}{T_{ry2}}$ $T_{ay1} = \dfrac{l_r}{V}, \quad T_{ay2} = \dfrac{l_f}{V}$ $T_{ay2} = \dfrac{l_z}{2K_r l}, \quad T'_{ay2} = \dfrac{l_z}{2K_f l}$

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

TABLE 18.8[1]

Symbol	Dimension	Name
m	kg	Mass of the vehicle
I_z	kg · m^2	Moment of the inertia of vehicle
l	m	Wheel base
l_f, l_r	m	Distance between center of gravity and front/rear wheel shaft
V	m/s	Vehicle speed $V = \sqrt{v_x^2 + v_y^2}$
v_x, v_y	m/s	Longitudinal and lateral velocity of the center of gravity
a_x, a_y	m/s^2	Longitudinal and lateral acceleration of the center of gravity
ψ, ϕ	rad	Yaw angle, roll angle
$\dot{\psi}, \dot{\phi}$	rad/s	Yaw velocity roll rate
β	rad	Side slip angle of the center of gravity
δ_f, δ_r	rad	Front/rear wheel steering angle
α_f, α_r	rad	Front/rear side slip angle
K_f, K_r	N/rad	Equivalent cornering power of front/rear
δ_H	rad	Steering wheel angle
i_s	I	Steering ratio in over all

TABLE 18.9[1]

Classification by functions	Aims
Small range of rear steer angle only controlled electronically	Improvement of steering response and vehicle stability in medium to high speed
Not only small range in medium to high speed but large range in low speed of rear steering angle are controlled electronically	In addition to the above, making the minimum turning radius small

TABLE 18.10[1]

Classification by mechanism	Feature
Full mechanical system	Simple mechanism
Electronic-hydraulic system	High degree for control freedom (compact actuator)
Electronic-mechanical-hydraulic system	High degree of freedom (mechanism is not simple)
Full electric system (electronic-electric system)	High degree of control freedom simple mechanism

18.2.3 Introduction to Each System

Fully Mechanical System. This was the first 4WS system used in passenger vehicles. It was adopted in the Honda Prelude in 1987 and in the Honda Accord in 1989. This system is a front-wheel steering angle responsive-type system in which the rear-wheel steering angle is determined wholly by the steering angle of the front wheels. The structure and fabrication of this system are shown in Figs. 18.28 and 18.29.

In rack-and-pinion type steering gearboxes for the front wheels, a rear-wheel steering pinion is provided in order to transmit the steering angle of the front wheels to the rear wheels. The displacement in the steering angle is transmitted to the rear steering gearbox via the center steering shaft. As shown in the figure, the rear steering gearbox consists of a combination of an eccentric shaft and a planetary gear. The input and output characteristics that can be obtained are shown in Fig. 18.30. The result of this is that when the steering angle for the front

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US PATENT NO. 6,557,540

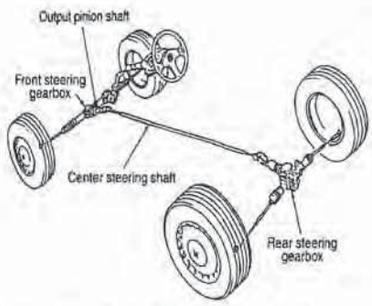


FIGURE 18.28 Fully mechanical 4WS.

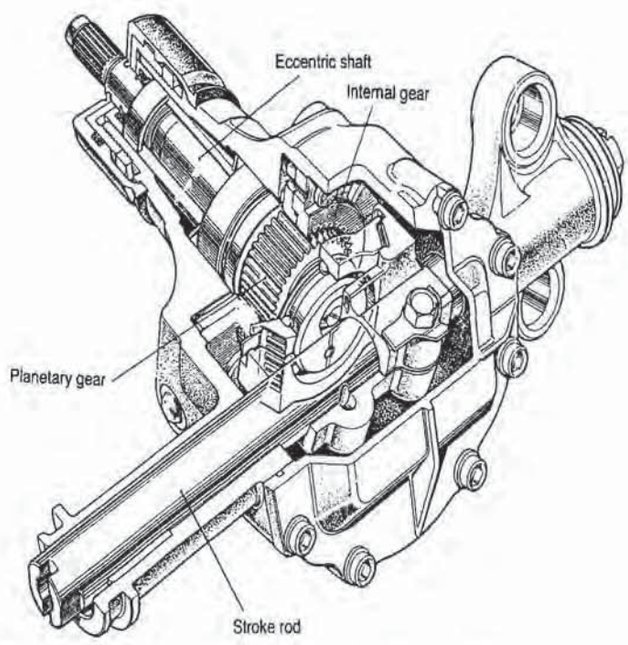


FIGURE 18.29 Rear steering actuator of full mechanical 4WS.

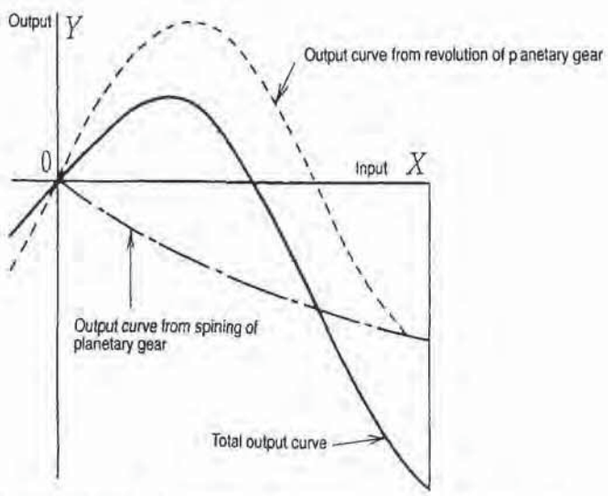


FIGURE 18.30 Input-output characteristics of rear steering gearbox.

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US PATENT NO. 6,557,540

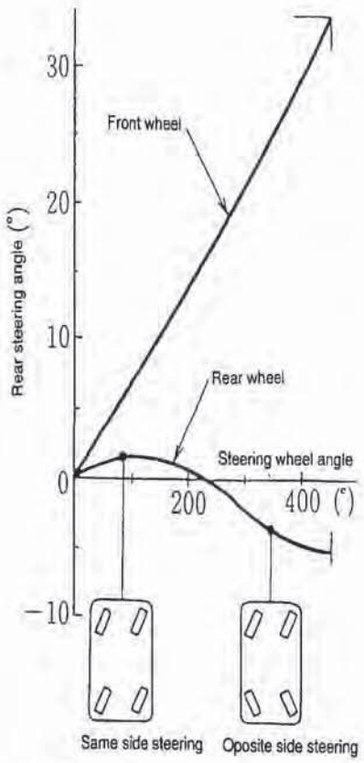


FIGURE 18.31 Characteristics of front steering speed-responsive 4WS.

wheels is small, the rear wheels turn in the same direction, but when the steering angle for the front wheels is large, the rear wheels turn in the opposite direction, as shown in Fig. 18.31. During high-speed driving, since the front wheels are only turned by very small amounts, the rear wheels thus turn in the same direction, and driving stability is improved. At extremely low speeds where the front wheels are steered through much larger angles, the rear wheels turn in the opposite direction and the turning radius becomes small, thus, the working ability should be improved.

Electronic-Hydraulic Control System

Vehicle Speed/Lateral Acceleration Responsive Type. This system was installed in vehicles like the 1986 Nissan Skyline. The structure of the system and the construction of the rear gearbox are shown in Figs. 18.32 and 18.33, respectively. In this system, a special hydraulic valve, which generates hydraulic pressure balanced with the reaction force from the front wheels in proportion to the lateral acceleration, is provided in the front-wheel power steering system, and this hydraulic pressure is transmitted to the actuator for the rear wheels. The rear-wheel actuator contains a high spring rate spring which allows displacement of the output rod to the position that balances the hydraulic pressure received. The rear wheels are steered in the same direction as the front wheels by means of the displacement of this rod. Accordingly, the relationship between the vehicle speed and the steering angle of the rear wheels varies in accordance with

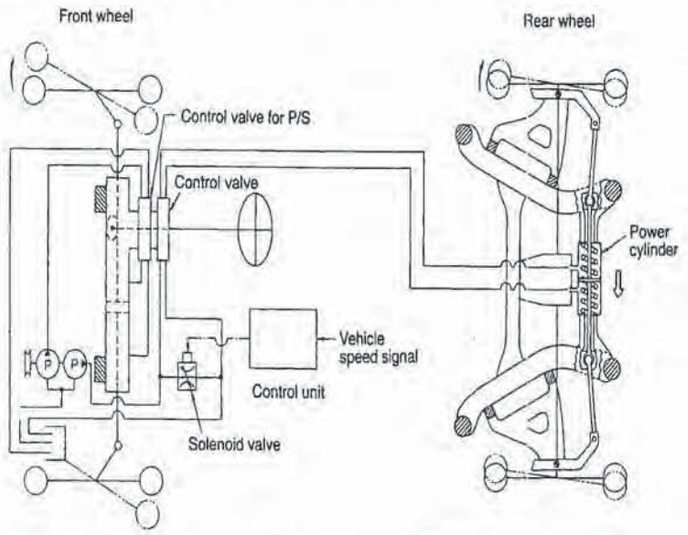


FIGURE 18.32 Lateral acceleration-responsive 4WS.[1]

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US PATENT NO. 6,557,540

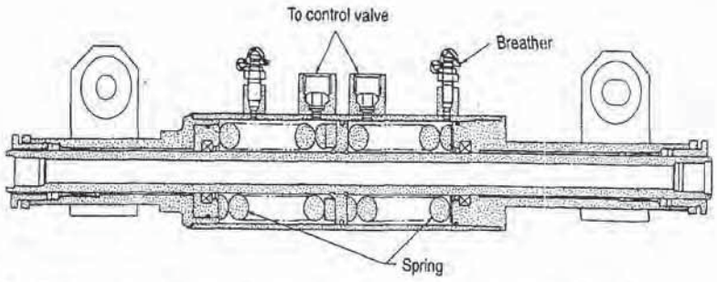


FIGURE 18.33 Rear steering actuator of lateral acceleration-responsive 4WS.[1]

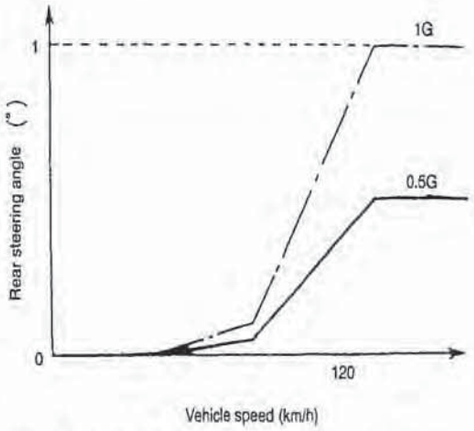


FIGURE 18.34 Characteristics of rear steering angle.[1]

the lateral acceleration as shown in Fig. 18.34. In this system, the maximum steering angle of the rear wheels is confined to a fairly low value, so that it is not intended to be a system for improving the minimum turning radius of the vehicle at very low speeds.

Vehicle Speed/Front-Wheel Steering Angle/Steering Wheel Velocity Responsive Type. This system was installed in vehicles such as the 1988 Nissan Sylvia, and was developed for the purpose of further improvement of vehicle stability and maneuverability during medium- to high-speed driving. The structure of the system and the construction of the rear-wheel actuator (solenoid servo valve) are shown in Figs. 18.35 and 18.36, respectively. In this system, the working fluid discharged from the hydraulic pump is directly introduced to the solenoid servo valve and controlled by instructions from an ECU (electronic control unit), after which it is sent to the power cylinder.

The ECU detects the front-wheel turning angle and the steering speed of the steering wheel by means of signals from the steering wheel angle sensor located in the steering wheel. The values thus calculated and the vehicle speed are used to determine the steering angle for the rear wheels. The maximum steering angle of the rear wheels is confined to a fairly low value in this system also, so that it was intended only to be a system for improving the vehicle stability and maneuverability at medium to high speeds. However, it does not improve only the stability by turning the rear wheels in the same direction, but also improves the maneuverability by momentarily turning the rear wheels in the opposite direction to the front wheels when the steering wheel is turned quickly at medium vehicle speeds. That is, the initial yawing movement (yaw velocity) of the vehicle is improved and steering response is thus improved. Figure 18.37 shows the steering angle pattern of the rear wheels with time expressed along the horizontal axis.

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

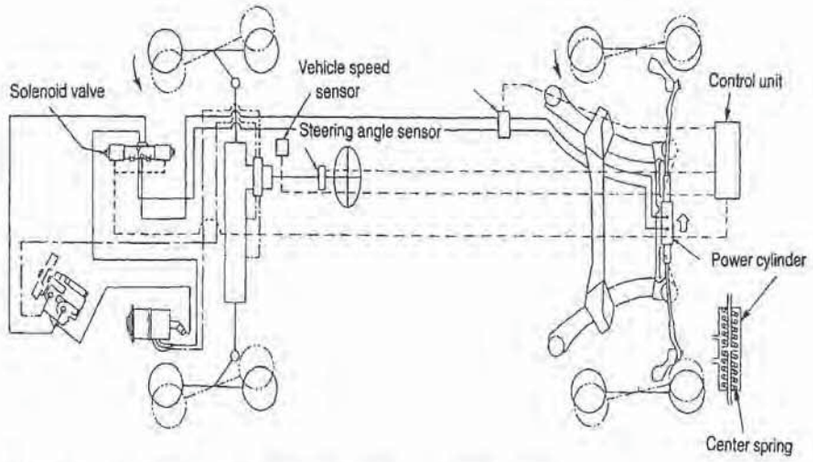


FIGURE 18.35 System construction of front-wheel-responsive 4WS.[1]

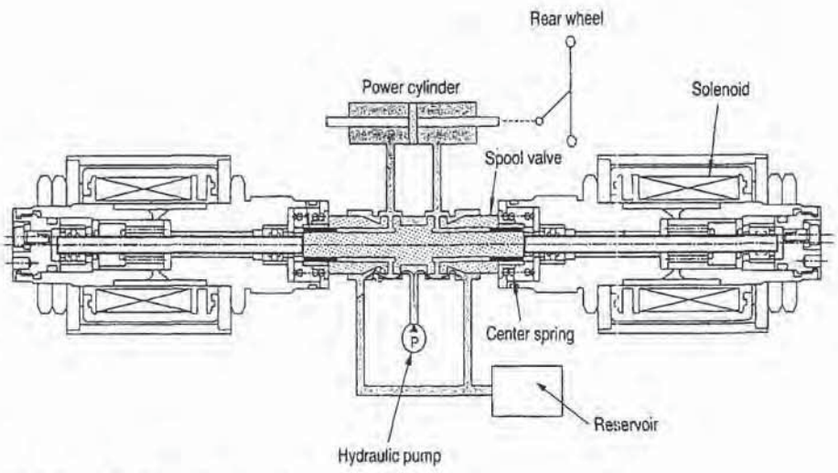


FIGURE 18.36 Solenoid servo valve of front steering angle-responsive 4WS.[1]

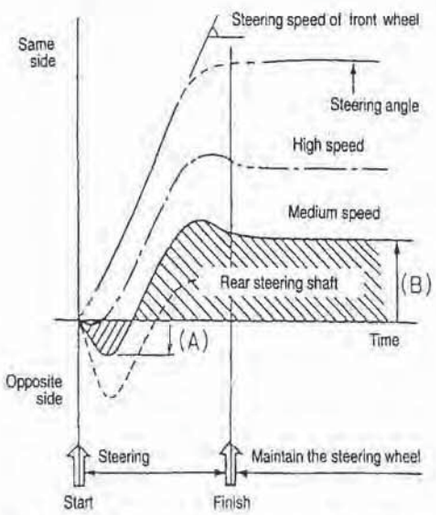


FIGURE 18.37 Characteristics of front steering angle-responsive 4WS.[1]

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

Electronic-Hydraulic-Mechanical Systems

Vehicle Speed/Front-Wheel Steering Angle Responsive Type. This system was introduced in the 1987 Mazda Capella, and was designed with the aim of being a system for improving both stability when driving at high speeds, like the fully mechanical system, and vehicle minimum turning radius at low speeds. The structure of the system and the construction of the rear actuator are shown in Figs. 18.38 and 18.39, respectively. This system can be broadly divided into the power assist section and the phase control section. The power assist section consists of a linear spool valve and a power piston, and utilizes hydraulic pressure as the dynamic force. The phase control section comprises a bevel gear which engages with the input shaft, a control yoke, and a control rod which is connected to a valve. The angle of the control yoke is driven by a stepping motor via a worm gear, thus controlling the turning direction of the rear wheels so that the rear wheels can be turned, not only in the same direction, but also in the opposite direction as the front wheels.

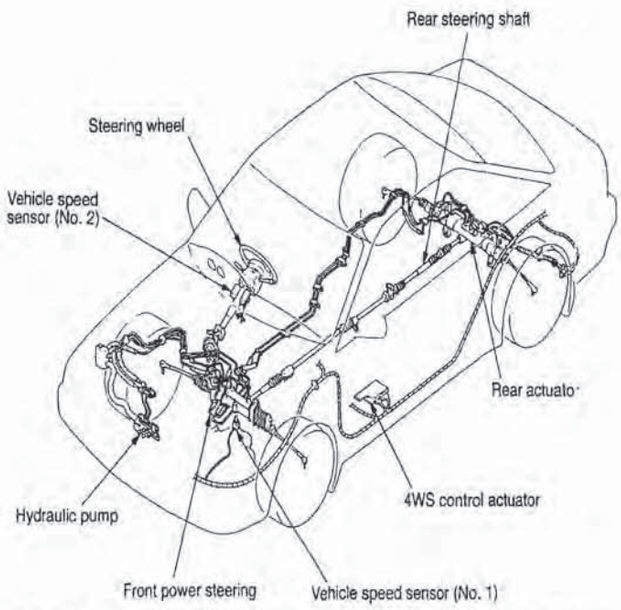


FIGURE 18.38 Front steering angle and vehicle speed-responsive 4WS.[11]

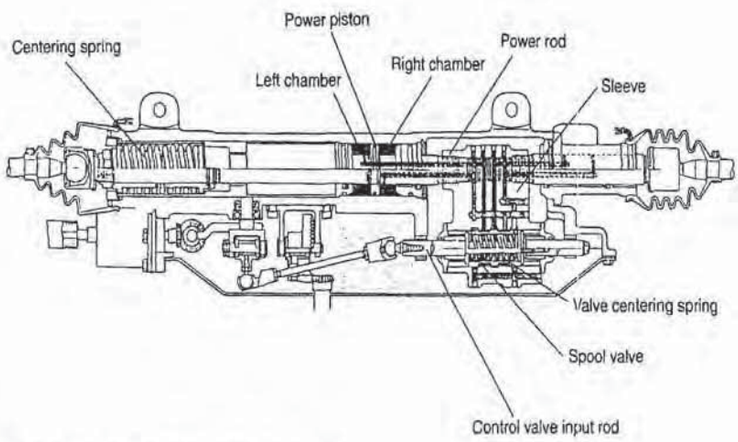


FIGURE 18.39 Rear steering actuator.[11]

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US PATENT NO. 6,557,540

In Fig. 18.40, *a* represents the situation in which the rear wheels are turned in the same direction, and *b* represents the situation in which the rear wheels are in the opposite direction. The direction of movement of the valve spool in relation to the turning direction of the input shaft is opposite for *a* and *b* as shown. Figure 18.41 shows the situation in which the ratio of the steering angle of the rear wheels with respect to the steering angle of the front wheels is continuously changing with respect to the vehicle speed.

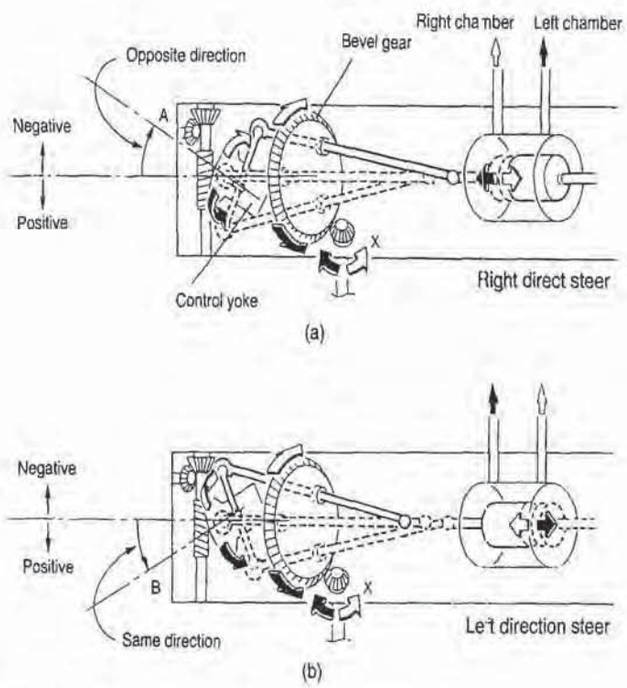


FIGURE 18.40 Explanation diagram of rear actuator.[11]

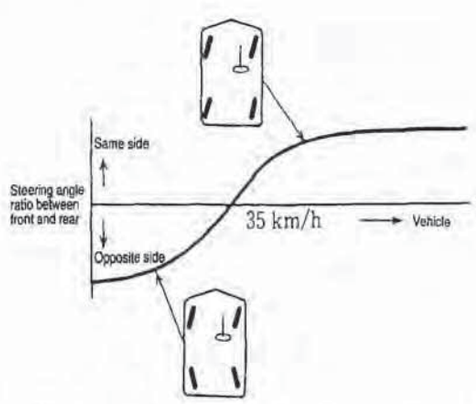


FIGURE 18.41 Characteristics of front steering angle and vehicle speed-responsive 4WS.[1]

Vehicle Speed/Front-Wheel Steering Angle/Yaw Velocity Responsive Type. This system was introduced in the 1992 Toyota Soarer. Figures 18.42 and 18.43 show the system structure and the construction of the rear-wheel steering actuator, respectively. The system consists of two sections: a *mechanical-hydraulic type* steering mechanism to steer the rear wheels by a

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

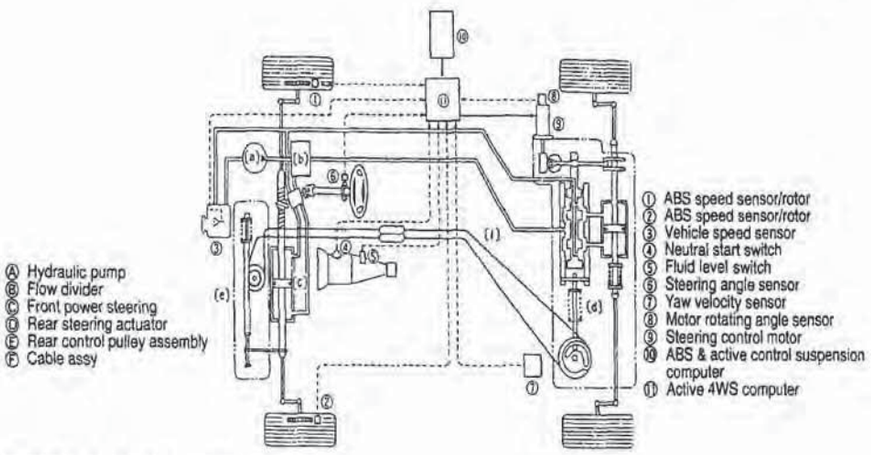


FIGURE 18.42 Active 4WS system.[10]

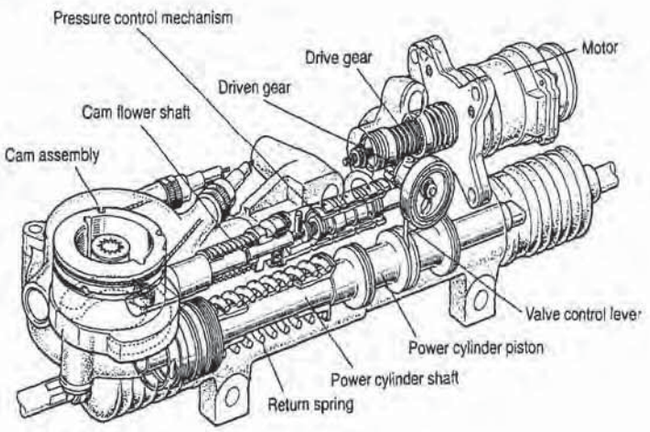


FIGURE 18.43 Rear steering actuator unit.[10]

considerably large steering angle in the opposite direction to the front wheels in order to make the turning radius small at low speeds, and an *electronic-hydraulic type* steering mechanism to steer the rear wheels by a considerably smaller value in order to not only improve the steering response and the stability during middle- to high-speed driving, but also to suppress the unexpected movements due to the outside disturbances.

Mechanical-Hydraulic Type Steering Mechanism. This mechanism consists of steering angle transmission cables, which transmit information concerning the steering angle of the front wheels to the rear-wheel steering actuator; a joint to connect the front and rear cables; a pulley assembly, which converts the displacement of the front-wheel steering rack into cable movement; a cam, which converts the movement of the cable into displacement of the sleeve valve inside the rear-wheel steering actuator; and a copy valve, which steers the rear wheels as far as the instruction given by the sleeve valve and maintains them in position. Furthermore, a dead zone is provided in the pulley assembly so that the rear wheels are not steered in the opposite direction to the front wheels when the vehicle is traveling at high speed.

Electronic-Hydraulic Type Steering Mechanism. This mechanism is designed to improve the steering response and the stability of the vehicle and to suppress the unexpected movements from outside disturbances when the vehicle is traveling at medium to high speeds. For this purpose, the system consists of five types of sensors for detecting the driving conditions and the steering situations, an ECU which determines the steering angle for the rear wheels

PETITIONER'S EXHIBIT 1002
US PATENT NO. 6,557,540

based on the signals from these sensors, a pulse motor which drives the gear mechanism based on instructions from the ECU, a gear mechanism which converts the rotation of the pulse motor into a spool valve displacement, and a hydraulic copy valve which steers the rear wheels as far as the instruction given by the sleeve valve and maintains them in that position.

Control Method. The basic algorithm to determine the steering angle for the rear wheels is given in Fig. 18.44. When driving in the medium- to high-speed range, the front-wheel steering angle proportional gain for the rear wheels is set to 0; instead of that, the yaw velocity proportional gain in the direction which suppresses the yaw rate is set. Accordingly, in the medium- to high-speed range, the rear wheels are turned in the same direction as front wheels, with the same effect as that of the other system mentioned before. The feature of this system is that a yaw rate sensor is provided to detect the vehicle dynamics and utilizes its signal as a control parameter. The effect of this is the addition of a function which automatically compensates the confusion in vehicle behavior caused by outside disturbance.

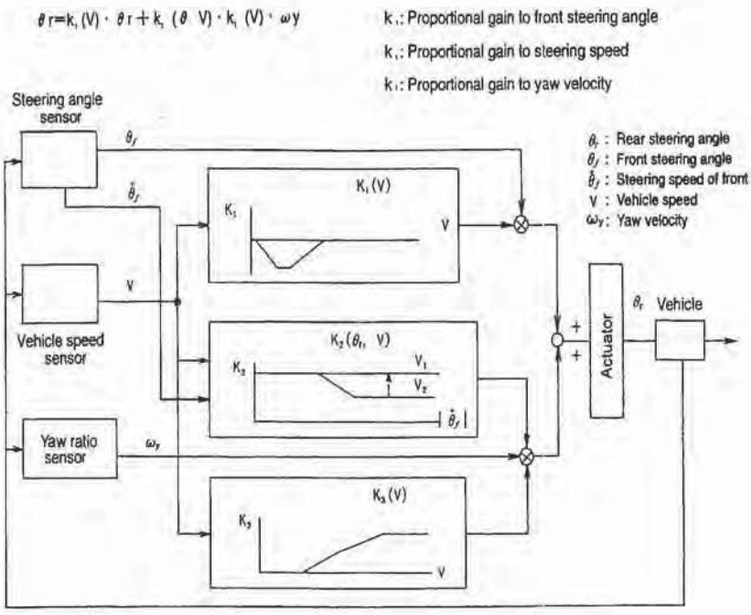


FIGURE 18.44 Control block diagram.[10]

18.2.4 Electronic-Electric Control Systems

Vehicle Speed/Front-Wheel Steering Angle/Steering Speed Responsive Type. This system was introduced in the 1991 Honda Prelude. The system structure is shown in Fig. 18.45, and the construction of the rear-wheel actuator is shown in Fig. 18.46. This system has a structure whereby the rotation of an electric motor that is controlled by the ECU is converted into linear motion by means of a ball screw to directly drive the rear wheels. The system is easy to install. Cross sections of the structures of the front-wheel steering angle sensor and the rear-wheel steering angle sensor are shown in Figs. 18.47 and Fig. 18.48, respectively.

Also in this system, the turning of the rear wheels is determined by the vehicle speed, the front steering angle, and the steering wheel velocity. The determining algorithm is similar to the system mentioned previously. In this system, improvements in the stability at high speeds, improvement in the steering response when the steering wheel is turned quickly, and improvement in the turning radius of the vehicle at low speeds have been realized simultaneously.

The most recent example of an electronic-electric system is the system[12] that has been installed in the 1993 Nissan Laurel. The approach to control in this system is the same as in the

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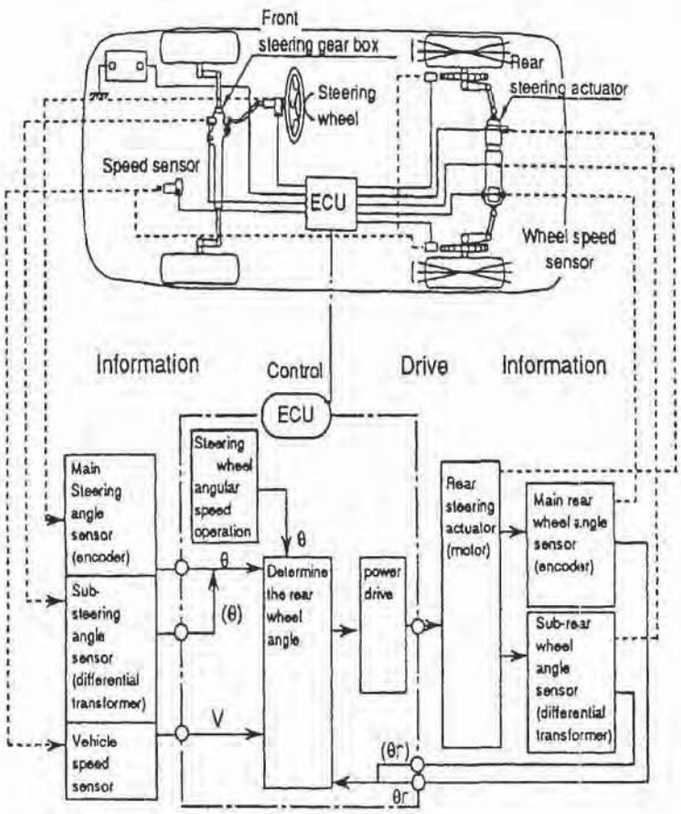


FIGURE 18.45 Block diagram of the full-electric 4WS.

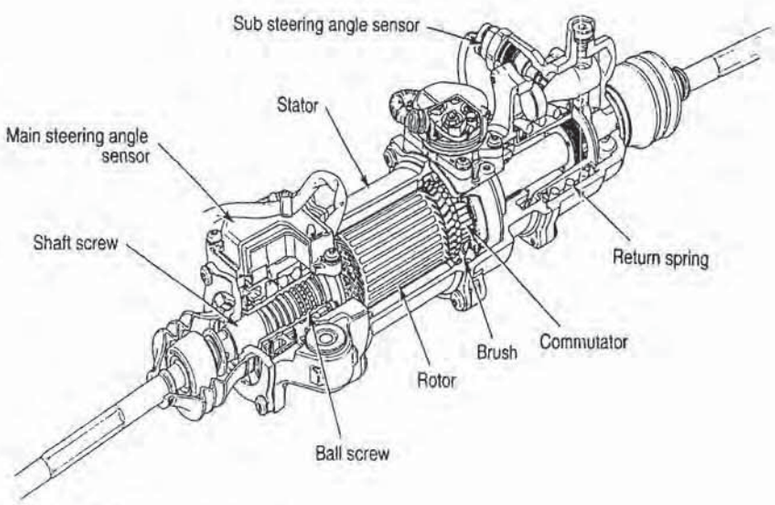


FIGURE 18.46 Rear steering actuator.

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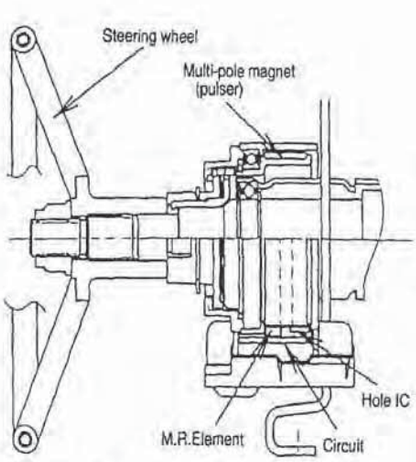


FIGURE 18.47 Main steering angle sensor.

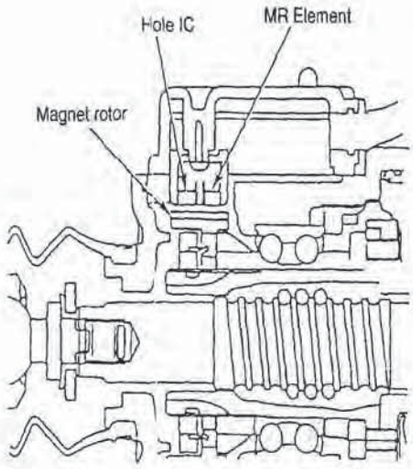


FIGURE 18.48 Rear steering angle sensor (main).

vehicle speed/front-wheel steering angle/steering wheel velocity responsive type system mentioned before, but by using an electric actuator, the space taken up by the system has been reduced and ease of installation improved.

Figure 18.49 shows the system structure, and Fig. 18.50 shows the steering angle generation modes for the rear wheels.

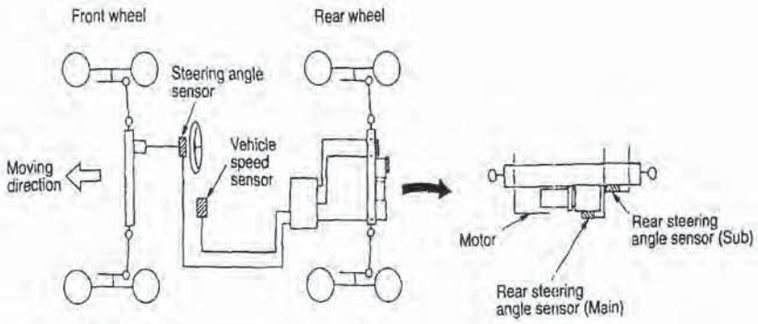


FIGURE 18.49 System construction of HICAS.[12]

18.2.5 Trends in Research and Development

Four-wheel steering systems that have been installed in current vehicles mainly adopt the program control technique in which the rear-wheel steering angle is programmably determined based on the beforehand scheduling relationship with the vehicle speed, the front steering angle, and steering wheel velocity. The aim of the system is the improvement of the insufficient performances in handling characteristics of the vehicle under special steering situations.

Recently, however, the system based on the new concept is being earnestly researched in order to drastically improve the handling characteristics in any region for practical use. Such technology is similar to that of CCV (control configured vehicle), which is used in airplanes, and named as *active control technology* in general. In this section, a basic introduction to the following two representative concepts concerning the active four-wheel steering system (A-4WS), which adopts the active control technology in the 4WS systems, will be given.

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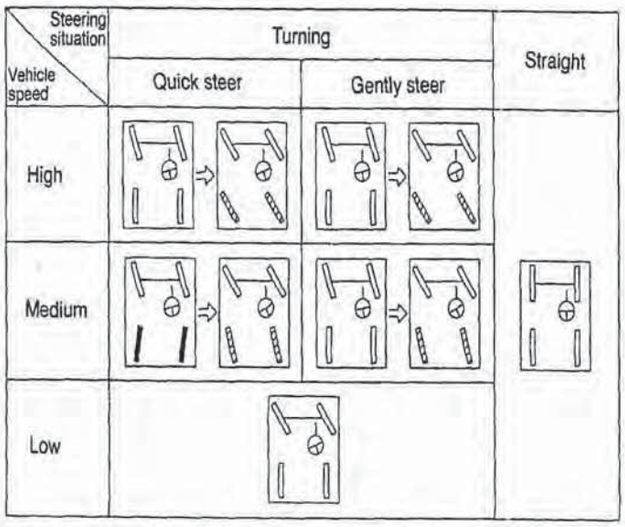


FIGURE 18.50 Rear steering angle generation mode.[12]

Vehicle's Slip Angle Control System. Generally, the moving direction of the center of gravity of the vehicle is not the same as the heading direction of the vehicle while the vehicle is turning, and the angle between these two directions is called the *side slip angle of the vehicle.* When the side slip angle of the vehicle is large, the driver is forced to drive with an oblique line of vision. This can be considered one cause of increased driving difficulties. From this point of view, the control concept has been proposed whereby the side slip angle of the vehicle while driving is always kept close to 0. In addition, the cause of the increasing of phase delay in lateral acceleration according to the vehicle speed is that the steady state gain in the sideslip angle is reduced together with the vehicle speed and becomes negative at high speed. In order to eliminate such phenomena, the concept which keeps the sideslip angle close to 0 may also be desirable.

The sideslip angle B can be expressed in general by the following equation:

$$\beta = \frac{1}{POL(s)}\{B_f(1+\tau_f \cdot s)\delta_f + B_r(1+\tau \cdot s)\delta_r\} \tag{18.9}$$

$$B_f: \frac{2l_r lk_r - l_f m v^2}{2lk_r V} \cdot A, B_r: \frac{2l_f lk_f - l_r m v^2}{2lk_f V} \cdot A$$

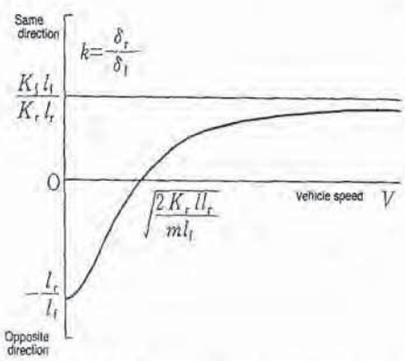


FIGURE 18.51 Steering angle ratio for making the vehicle sideslip angle equal to 0.[1]

Provided the numerator s in Eq. (18.9) = 0, the steering angle ratio between the front and rear wheels that is required for realizing a sideslip angle of 0 can be obtained from Eq. (18.10) as a function of the vehicle speed. This is shown graphically in Fig. 18.51.

$$\frac{\delta_r}{\delta^f} = -\frac{B_f - B_f \tau_{fs}}{{}_B r + B_r \tau_r s} \tag{18.10}$$

$$= \frac{-\iota_{f+} + (\iota_f m/2k_r\iota)V^2 - (I_z/2K_r\iota)V \cdot s}{\iota_{f+} + (\iota_r m/2k_f\iota)V^2 - (I_z/2K_f\iota)V \cdot s}$$

If the front- and rear-wheel control principle that includes the phase proceeding func-

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US PATENT NO. 6,557,540

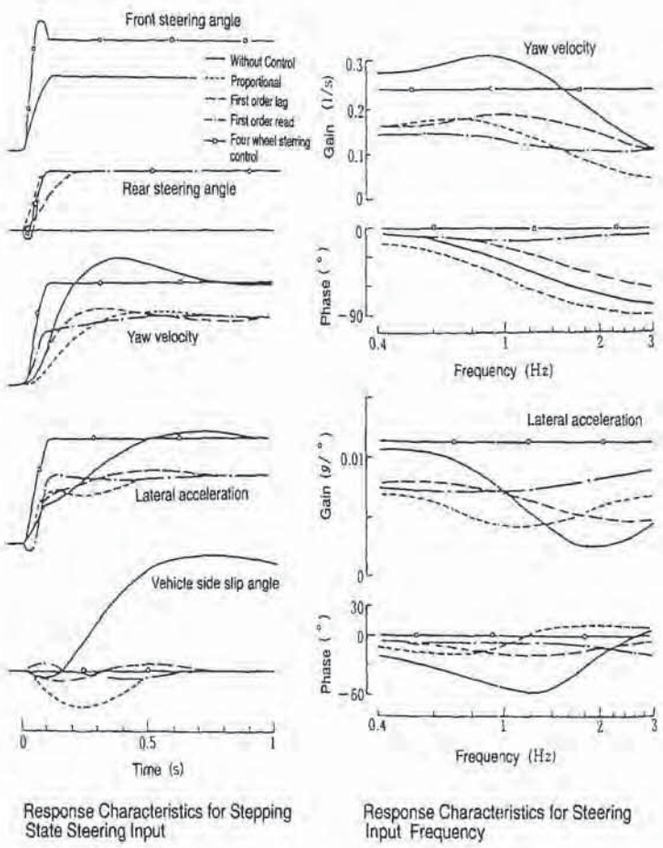


FIGURE 18.52 Affections in steering response characteristics by using the delay/advanced control means.[1]

tion for the front-wheel steering angle is employed, the vehicle characteristics can be set more freely. An example of the simulation analysis when the front and rear wheels are steered with a control principle such as the above is shown in Fig. 18.52.

Model Following Control System. In this system, the desirable vehicle behavior in accordance with the driver's steering operations is predetermined as a reference model, and the control is carried out so that the actual vehicle behavior fits this model. Generally, the lateral acceleration and the yaw velocity are used as the parameters for expressing the lateral movement of the vehicle. If the independent models are set for both of these characteristics, and make the actual vehicle behavior follow them, it is necessary to actively control both of the front and rear wheels. Research into systems such as this is currently being carried out, but in this section, a system for controlling the steering angle of the rear wheels that uses a range model for D^* which is defined by the linear combined Eq. (18.11) for the yaw velocity and the lateral acceleration will be introduced.

$$D^* = da_y + (1 - d)V\phi \tag{18.11}$$

If the weighted constant d in Eq. (18.11) equals 0, this control system becomes a yaw velocity model following control system, and if $d = 1$, then it is a lateral acceleration following control system.

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GLOSSARY

Convergence Yaw stability of the vehicle when driver inputs a rapid steering wheel movement and releases the steering wheel.

Kickback Steering torque and angle arising from inverse input through the tires from uneven concrete road or other surface.

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Makoto Sato, executive chief engineer of Honda R&D Co., Ltd., was graduated from Nagoya Institute of Technology in 1960 and began his career with Kokusan Electric Co. He moved to Honda R&D Co., Ltd. in 1968 and began working on vehicle dynamics control systems with electronic technologies. He is now researching new safety technologies with intelligent electronics, especially in the area of crash avoidance.

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US PATENT NO. 6,557,540

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CHAPTER 19
LIGHTING, WIPERS, AIR CONDITIONING/HEATING

Richard Valentine
Motorola Inc.

19.1 LIGHTING CONTROLS

Controlling lamps with power electronics offers many advantages and a few disadvantages over conventional switches or relays. An important advantage includes easier diagnostics compared to the classical mechanical switch or relay approach, while a significant disadvantage includes the higher cost of the electronics. Because some lights, such as headlamps, turn signals, and brake lamps are safety related, a method to test the integrity of these lamps is an advantageous feature. The power electronic design can not only turn lights on or off, but it can vary the light's intensity and detect abnormal conditions such as open or shorted lamps. The cost tradeoff issue becomes more interesting when the power electronics load control is connected onto a data bus or multiplexed network (see Chap. 26).

The typical automotive lamp can range from small-wattage panel lamps to large 60-W or higher headlamps (refer to SAE ref HS-34/93 for auto lamp standards). Tungsten and halogen types prevail for many vehicular lamp designs. The prime difference between standard tungsten nonhalogen and halogen lamps is in light efficiency. Halogen types produce about 20 percent more light for each watt of energy consumed. Other types of lamps, such as light-emitting diodes (LEDs), fluorescents, or gas discharge, are somewhat adaptable for vehicular applications. The power electronics operating requirements vary for each lamp technology category.

19.1.1 Incandescent/Halogen

The single most important rule when switching normal incandescent lamps is to select a power transistor that can sustain a 10 to 15 times inrush current level. For example, a 2-A-rated stop lamp requires a 20- to 30-A-rated power transistor for an electronic switch design. When designing a lamp-driven circuit, other considerations besides inrush currents must be taken into account: maximum power transistor operating temperatures, shorted load protection, voltage supply transients, reverse voltage supply condition, open load detection, high or low side design topology, MCU interface, switching edge speed limits, and variable brightness control. In addition, the electronic lamp switch should not drop more than 0.3 V to ensure that the lamp's intensity is not degraded. Incandescent lamp light output is very sensitive to the lamp's operating voltage. A 20 percent drop in an incandescent lamp's nominal operating voltage can reduce its light output by 50 percent.

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Inrush Current Effects. Lamp loads behave like high positive temperature coefficient resistors. When the lamp is cold, the resistance value of its filament is about 1/10 of its normal operating resistance. Lab tests show a definite relationship between the lamp's peak inrush current level and light output intensity with respect to time. If the lamp current is limited to its nominal value, it takes longer—one-half second or more—for the filament to reach its nominal temperature, but if the current supply is unlimited, the lamp heats up very quickly. The lamp response time and the available supply current must be considered for applications when an instantaneous light output is desired, such as with turn signals or brake lights in a vehicle. A 1-s delay at 89 km/h (55m/h) equals 24.6 m (81 ft). Four seconds of time is about how long some vehicles take to stop at a vehicle speed of 89 km/h. If the driver has exerted maximum braking, and his stop lamp waits for 1 s to convey this action to trailing vehicles, the driver will almost be halfway stopped before the trailing vehicle has a stop light indication. This is not acceptable, and the delay time in stop or possibly in turn signal indicators must be minimized to benefit the reaction time for other drivers.

Limiting the lamp's inrush current will probably increase the lamp's longevity in high duty cycle applications, such as a flasher, because the mechanical stress on the filament is reduced. A limited inrush current design should decrease the chances of the lamp failing during the initial turn on. In vehicular applications when external induced filament vibration is high, the lamp reliability may not be improved enough to justify a current limiting design. A current limiting design does increase the power transistor's peak heat dissipation, and when operating with an energy source that is capable of supplying high peak currents, such as an automotive battery, the rationale to include current limiting would be to use a smaller, less expensive transistor or to minimize voltage sags in the wire bus when several lamps are switched on at the same time. A current limiting design may include a network to control the switching transition times from OFF to ON to OFF.

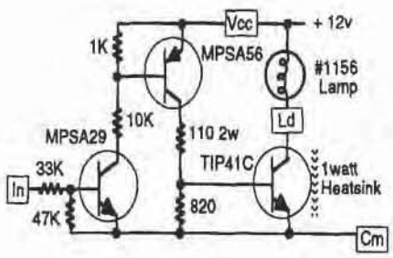
Setting the switching times to over 0.001 s would help minimize EMI effects. Most power transistors can switch in less than 1 μs, with some power FETs able to switch in 50 ns. These RF switching speeds are undesirable and will be detectable by other electronic systems in the vehicle, such as the AM radio.

Lamp Driver Design. The power electronics can drive the lamp in one of two simple ways. One is *low side switching* (Fig. 19.1) with the lamp connected between the power supply and the transistor, or *high side switching* (Fig. 19.2) with the lamp connected between the transistor and common. The main drawback to low side switching is that one side of the lamp is always connected directly to the battery supply, and if this lead is shorted to common, very high currents will occur. A high side switch disconnects the power supply from the load, so if the switch is off, the chances of an accidental power supply short when replacing or troubleshooting the lamp are minimal. On the other hand, in the high side switch design, a switch ON condition will cause very high current levels to flow through the power transistor in the event of a short. The selection between high or low side switching is affected by the lamp's environment.

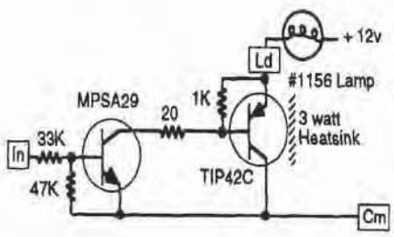
In a low side design, leakage currents that may flow from the lamp's socket or connectors can be a concern. This is especially true if the lamp is operated in a salt-laden, high-moisture atmosphere. This problem occurs because of the constant voltage potential at the socket and is one reason why most exterior vehicular lighting systems use high side switching. Other applications, such as instrument backlighting or signal indicators that are usually located inside the vehicle, can be a low side design to take advantage of lower-cost NPN transistors or N-channel power FETs. N-channel power FETs can be used for high side switching by adding a charge pump as shown in Fig. 19.2*a*. Special power linear devices are also available that incorporate a charge pump for high side switching of small lamps. These power linear IC devices (MC3399, MC33091, or equivalent) will find more usage as their pricing and reliability become more attractive for automotive application.

Power Control Device Selection. When the drive topology has been decided upon, a power device can be chosen. There are numerous tradeoffs between bipolar and power FET transistors. The N-channel power FET device requires minimal drive power for a low side switch and

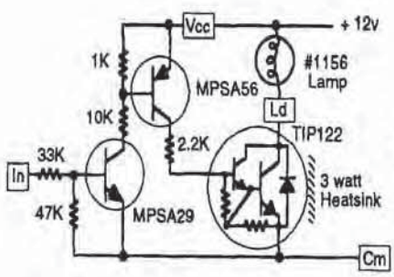
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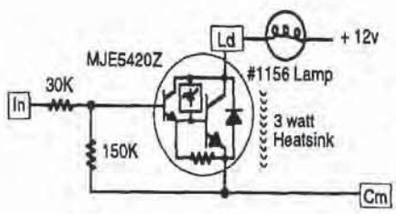
The NPN transistor runs in full saturation mode, but requires 1.5 watt base drive resistor.



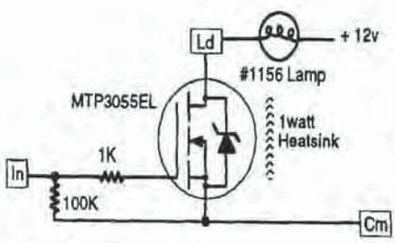
The PNP transistor dissipates about 3 watts since it does not operate in saturation mode.



NPN Darlington transistor dissipates about 3 watts since it does not operate in saturation mode.



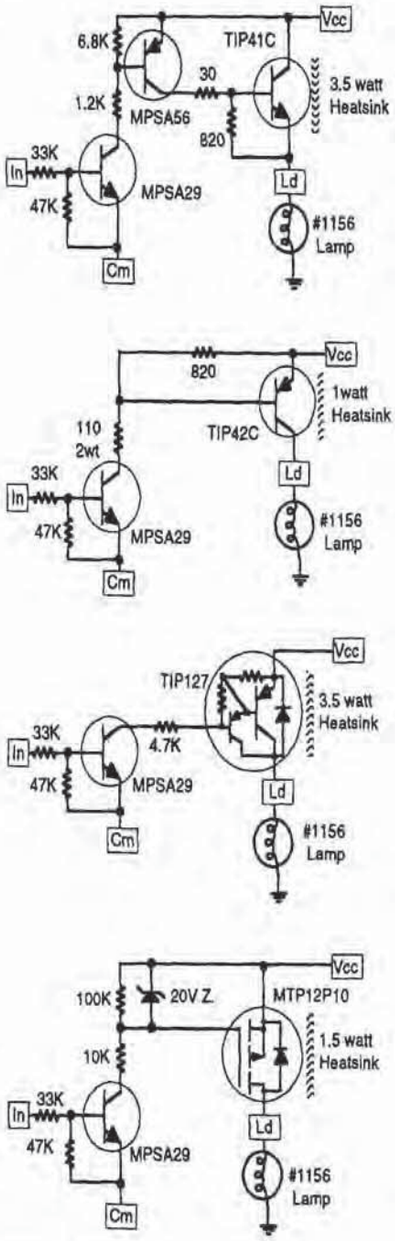
NPN Darlington trans stor has internal active zener element to clamp over voltage transients in forward SOA mode. Device dissipates about 3 watts since it does not operate in saturation mode.



N Channel MOSFET transistor dissipates about .8 watts, and can sustain low energy voltage spikes.

FIGURE 19.1 Low side switching.

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US PATENT NO. 6,557,540



The NPN transistor dissipates about 3 watts since it does not operate in saturation mode. Circuit requires two predrivers.

A PNP transistor can run in saturation mode, but dissipates about 1.5 watts in base drive resistor.

PNP Darlington requires small base current, but can't operate in saturation mode, and dissipates about 3 watts.

P Channel MOSFET offers best electrical performance, but the internal silicon die has to be twice as large as similar rated N-Channel MOSFET.

FIGURE 19.2 High side switching.

is cost-competitive with similar rated bipolars. Low side switching can easily use N-power FETs, whereas a high side switch would need more expensive P power FETs. Power FET transistors will usually be able to conduct the high peak inrush currents much more readily than bipolar transistors. This is because power FETs have higher transconductance than bipolars. Bipolars are usually biased for operation at the lamp's average load current and will therefore operate in a linear mode during the lamp's initial turn on, thereby offering some current-limiting at the expense of increased peak junction temperature. The power FET will usually con-

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US PATENT NO. 6,557,540

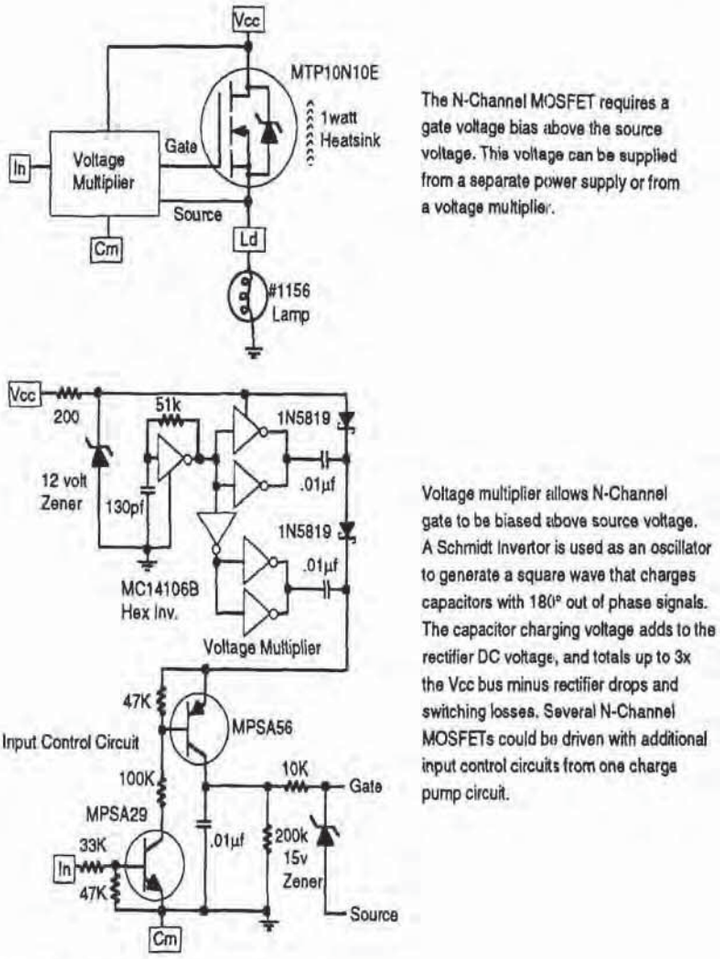


FIGURE 19.2*a* N FET high side design.

duct many times its average current without going into a linear mode, provided the overcurrent condition time period is fairly short, less than 100 ms.

Lamp driver long-term reliability is determined mainly by the power transistor's heat dissipation and somewhat by the solder joint totals. Selecting P-power FET devices for the high side configuration because of the low number of parts, ease of drive interface, and low power dissipation appears best, but P-channel FET devices are usually 50 percent more costly. The N-channel devices win both for cost and reliability in a low side switch design. Table 19.1 summarizes the various lamp driver designs.

Short-Circuit Protection. Another consideration is shorted circuits. The bipolar transistor is typically biased for minimum base drive power loss. During a shorted load condition, the bipolar will pull out of saturation, enter into a linear operation mode, and may fail due to excessive junction temperatures, ordinarily before a fuse will blow. A properly sized power FET may be able to safely conduct enough shorted load current to blow the fusing element. Power FET devices of 50 V are available that exhibit less than 0.0[illegible] ohm forward ON resistance in a fingernail-sized surface-mount package such as a MTB75N05HD or the equivalent.

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US PATENT NO. 6,557,540

TABLE 19.1 Lamp Switch Design Comparison

Device type	Driver stage loss, W	Output device loss, W	Pwr. eff. in/out, %	Relative system cost	Parts count total, pcs	Solder joints total, pts	Relative reliability
High side switch configurations for 2.1-A lamp load							
NPN	0.2	2.7	89	high	9	25	low
PNP	1.31	0.8	92	medium	6	18	medium
Darlington PNP	0.03	2.7	89	high	5	13	low
P-channel MOSFET	0.01	0.9	96	high	7	19	medium
N-channel MOSFET	0.05	0.7	97	highest	19	54	low
Low side switch configurations for 2.1-A lamp load							
NPN	1.32	0.8	91	medium	9	23	medium
PNP	0.13	3.2	88	high	5	15	low
NPN Darlington	0.08	2.7	90	high	8	23	low
NPN Darlington w/Zener	0.0004	2.7	90	low	3	10	medium
N-channel MOSFET	0.0002	0.7	97	low	3	10	high

Power supply voltage = 12 V

This type of device could safely conduct 14 A continuously (assuming a 50 °C ambient temperature and 2 W constant power dissipation, $[I = (P/R)^{1/2}]$.

Another way of protecting a transistor against excessive current levels is to monitor the current value by the insertion of a low-value power resistor in series with the load. A comparator is used to detect the current sense resistor's voltage drop and will toggle if this voltage drop exceeds the comparator's reference voltage. The comparator output is tied to an R-S flip-flop logic network that latches off the input signal. Some power FET devices are now manufactured with an internal current mirror element and eliminate the need for an external large power resistor. A basic circuit is shown in Fig. 19.3 for an internal current-sensing power

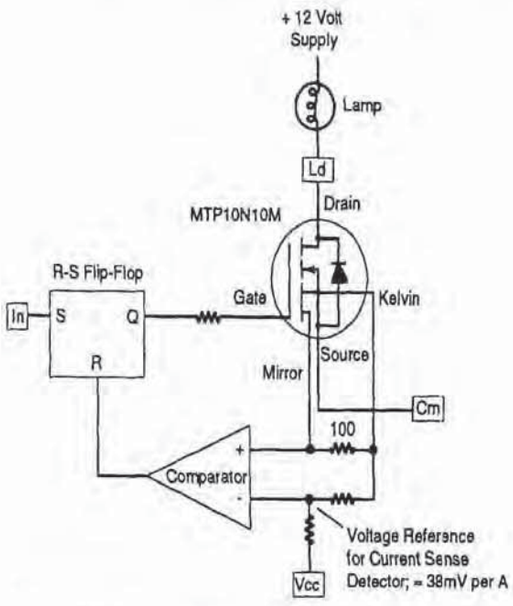


FIGURE 19.3 Internal current-sensing FET lamp drive.

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FET lamp driver. This circuit uses an RS flip-flop that will shut off the gate voltage. When the RS flip-flop has been toggled by the comparator, the RS input (set line) has to be retoggled to turn on the current-sensing power FET again. When an overcurrent condition occurs, the current-sensing power FET is latched off until the input is retoggled.

The transistor's voltage rating will be determined by the maximum voltage transients on the power supply line. In vehicular battery-charging systems, the normal 12-V supply can reach over 85 V due to intermittent battery connections when the alternator is operating. If the transistor is not avalanche-rated and enters a second breakdown due to excessive power supply voltages, it will fail shorted. Higher-voltage systems require much higher-voltage-rated devices.

MCU-based Lamp Driver. The current-sensing power FET can be directly interfaced to a single-chip microcomputer. The use of a single-chip microcomputer can allow the switch control circuit to perform many tasks: open load detection, shorted load protection, variable intensity, and diagnostics. The MCU concept is especially attractive when several lamps are clustered together and driven from the same module. Figure 19.4 shows the use of a single-

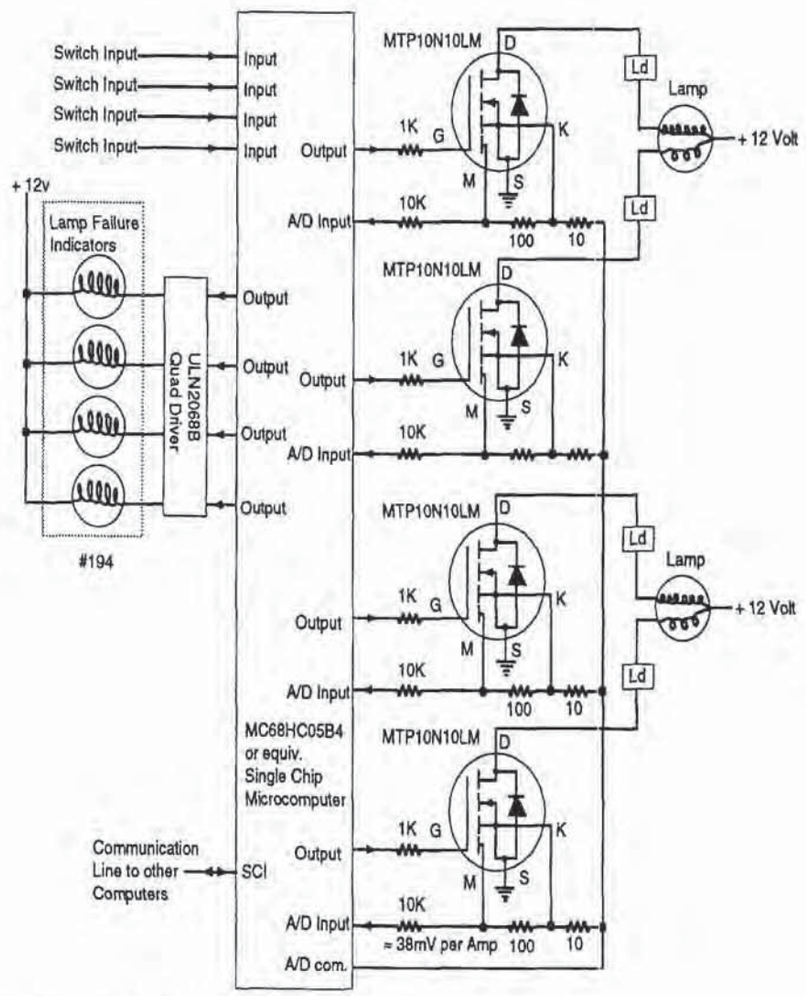


FIGURE 19.4 MCU lamp control.

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chip MCU for controlling four lamp filaments. Many more lamps could easily be added to this design. Low side switching is used to accommodate the use of logic level N-channel current-sensing power FETs. The MCU's program is set to sustain a shorted load condition for less than 50 ms; this allows a cold lamp turn-on condition to be tolerated. If a shorted load occurs after the initial turn on, the program can shut off the load in less than 1 ms, wait for 1 s, and then try to renenergize the lamp. A diagnostic fault lamp is energized to alert the user that a fault has occurred. An open load can be detected when the lamp is energized by having the program monitor the A/D inputs for a zero value. An 8-bit A/D converter will resolve to about 0.02 V when referenced from a 5-V supply. Therefore, a normal load current of 2 A will generate about a 0.084-V level, which will be read as a 04-hex number by the program. A value of 00 to 01 H would be considered an open, and a value of greater than 06 H would constitute a shorted load. A distinction could be made in the lamp failure indicator between an open and shorted load by turning the failure indicator full on for an open, and then flashing it at a 1-Hz rate for a shorted condition. The failure indicators are 3-W lamps for bright visibility, such as is required in vehicular dashboards. High-efficiency LEDs could be used when the 3-W lamp's brilliance is not required, and the LEDs may be directly driven from the MCU's output ports when connected in a low side switch configuration.

Variable light intensity can be obtained by programming a 100-Hz frequency with 0 to 100 percent pulse-width modulation. The 8-bit MCU can vary the pulse width by 256 steps, which would appear as an infinite intensity control to the human observer. One difficulty with PWM is when its duty cycle is producing very narrow pulses, such as at 1 percent or 99 percent. These narrow pulses may not allow the power stage to completely switch on or off. One method to minimize narrow PWM pulses is to force the PWM to jump from 0 percent to about 5 percent, and from 95 to 100 percent. This can easily be accomplished in the software design.

An external communication link can be established via the MCU's serial data port. This would allow the light control functions to be remotely accessible by another computer.

In summary, standard discrete power FET or bipolar transistors can effectively control incandescent lamp loads. Power FETs are preferred because of their low drive power requirement and high transconductance. The addition of an MCU enables the lamp control design to include many desirable features, such as shorted load protection, and minimizes the need for more advanced lamp driver semiconductors such as smartpower or power IC devices. MCUs are available with a built-in MUX port to communicate with other load control modules for remote operation of several lamps or other loads.

19.1.2 Fluorescent Lamps

A fluorescent lamp is commonly defined as a light source that produces light by the interaction of an arc with the gas mixture inside a glass vacuum tube. When this gas mixture is ionized by means of a high-voltage pulse from the tube's end terminals, its internal resistance decreases from megohms to several hundreds of ohms. Fluorescent lamps are typically 2 to 3 times more power efficient than standard incandescent lamps. The automotive lamp applications for fluorescents may include inside lighting, instrument back panel lighting, and other areas where an unfocused and discontinuous light spectrum, but highly efficient light source, can be utilized. The main drawback to fluorescent lamps is that they require a fairly high voltage to initiate the arc and a means to limit the lamp current once the arc is established. In other words, the fluorescent design will cost much more than a similar incandescent-sized light. This is an important consideration when operating the fluorescent lamp from a 12-V supply.

A 12-V powered fluorescent driver design requires an 8- to 16-V input to about 300-V output convertor with current limiting. These types of circuits usually operate at high frequencies (20 to 100 kHz) to minimize the size of passive components, such as transformers and capacitors, and to minimize audible noise. This type of circuit can generate significant EMI and RFI if not properly shielded and filtered. Figure 19.5 shows a block diagram of a 12-V fluorescent driver.

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US PATENT NO. 6,557,540

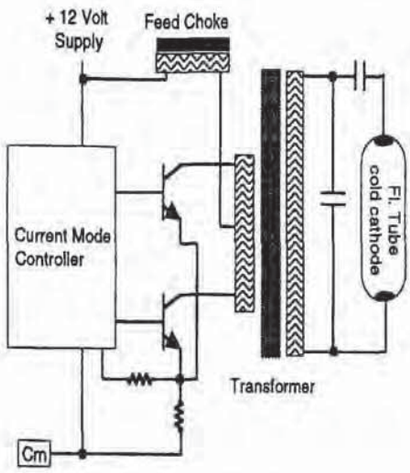


FIGURE 19.5 12-V fluorescent driver.

19.1.3 High-Intensity Gas Discharge Lamps

This class of lights includes mercury vapor, metal halide, and low- or high-pressure sodium. Some progress has been made to incorporate HID (high-intensity discharge) lamps in automotive applications. The HID metal halide type lamp offers high efficiency, up to four times more lumens per watt than incandescent, and high light output in a small package that can be utilized for headlamps. Again, the main drawback is the cost of the necessary power driver electronics. HID lamps require a circuit design similar to fluorescent drivers but at much higher power levels. The cost of the HID driver electronics limits their widespread use for automotive usage.

19.1.4 Electroluminescent (EL)

These low-intensity blue-green color light sources are used for decorative night lighting purposes and as backlighting for liquid crystal displays (LCD). EL lighting generally requires a low-current 100-V ac source. A 12-V dc to 100-V ac invertor is required for vehicular systems.

19.1.5 Light Emitting Diodes (LEDs)

Red LEDs may be adapted for stop light indicators. The LED produces a true red and does not require colored lenses. By combining several series LED strings together in one assembly, a stop light indicator is possible. A constant current source can be utilized to maintain a constant LED illumination level as shown in Fig. 19.6.

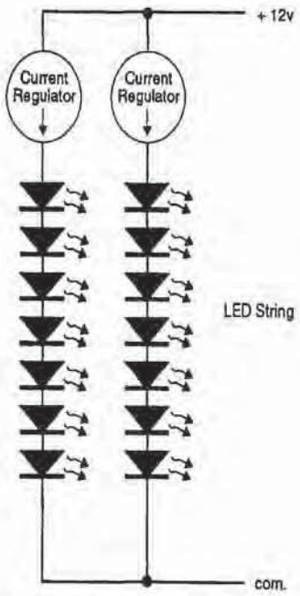


FIGURE 19.6 Constant LED illumination level design.

19.2 WINDSHIELD WIPER CONTROL

Windshield wiper (WW) systems are required by government regulations and therefore should be designed for high reliability. The motor control usually allows for low- and high-speed operation, or variable speed. The mechanical wiping motion is accomplished by a worm-gear design which gives torque multiplication and connects to the windshield wiper mechanical arm assembly. The key design specification is that the motor must sustain a "stalled" wiper condition without damage. The motor must also provide sufficient torque to run the wiper

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US PATENT NO. 6,557,540

mechanism under worst-case conditions, such as blades frozen to the windshield. Electronic controls are used with WW systems, usually for intermittent operation plus variable speed. The driver adjusts a time interval control potentiometer for a one- to several-seconds wiper rate. The electronic circuit activates the motor with the appropriate time interval. A travel limit switch is normally used to insure that the wipe cycle is completed, and that the blades are returned to their nominal position.

Automatic windshield wiper designs are possible with today's technology that utilize a moisture detector to determine the degree of WW action required.

19.2.1 PM Motor Speed Control

The auto industry uses dc PM motors because they are economical to produce and ensure good performance. The motor's speed can be varied by either a simple voltage dropping passive resistor or an active linear voltage regulator. This simplistic method is widely used for vehicular motors requiring variable speed control. It does have a serious drawback in terms of power efficiency. For example, to control the speed of a motor that draws 20 A at full speed requires about 10 A at half speed. At full speed, the overall motor control system's efficiency will be around 80 percent. If the speed is reduced to half, the system's efficiency drops to 40 percent. This is because there is a heat loss of 70 W in the series voltage dropping element, in addition to the 14 W lost in the motor. A more efficient speed control system is therefore a desirable goal and can be accomplished by interrupting the motor's voltage at a variable duty cycle or by using a switching power supply. A switching power supply to control up to 300 W would cost more than the motor because of its high-frequency transformers and other components.

PWM Speed Control Design. Electronic permanent magnet motor speed designs can range from a single power transistor to turn the motor on or off to an "H" bridge microcontroller-based system with closed-loop speed control. The design of a unidirectional 280-W 20-A motor speed control will be examined that uses power FET semiconductor technologies.

Since the PM motor is a dynamic machine with the armature acting as a flywheel, the voltage interruption or chopping rate could be 1000 Hz or slower, before the motor's speed actually pulsates. A problem at 1000 Hz or other audible rates is noise generated from within the motor. This audible motor sound may not be tolerable for drivers or passengers. At higher frequencies—16 kHz or greater—the audible noise is minimized. Another noise issue is the significant electronic radio frequency interference (RFI) that radiates into the electrical system, including the vehicle's radio equipment. This RFI is generated by the fast switching edges of the PWM signal. The drawing in Fig. 19.7 illustrates slow and fast switching speeds and the resultant power transistor dissipation relationships. Slowing down the switching edges minimizes the radio frequency interference (RFI), but the switching speed parameters also play a crucial role in the overall controller system. High transitional speeds are desirable to minimize switching losses and to improve reliability, but very fast switching speeds become impractical beyond a certain point due to the inherent inductance in the wire and component leads. A compromise has to be made between the edge speeds and power device heat loss.

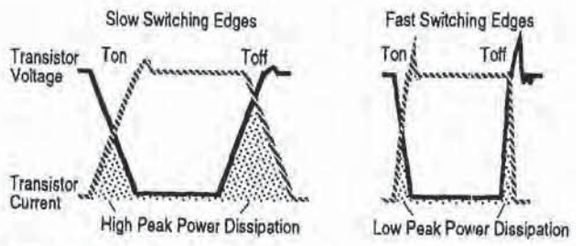


FIGURE 19.7 Slow vs. fast power loss.

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Switching Speed Effects. Figure 19.8 graphs the switching edge speed effects upon the power transistor's power dissipation when switching a 20-A motor load from a 14.4-V power supply. Three different operating frequencies are plotted, one at 20 kHz, one at 16 kHz, and the last at 2 kHz. Note how the 2-kHz switching times are less critical than the higher 16-kHz and 20-kHz frequencies. In order to minimize audible noise, the 16- or 20-kHz frequency can be used, but, unfortunately, the 200- to 500-ns switching speeds required to minimize switching losses cause the intrinsic inductance in the hookup wiring, components, and motor to become critical. These faster edge speeds will generate significant voltage spikes across the hookup wiring and may lead to reliability problems unless precautions are made to minimize stray inductance in this wiring.

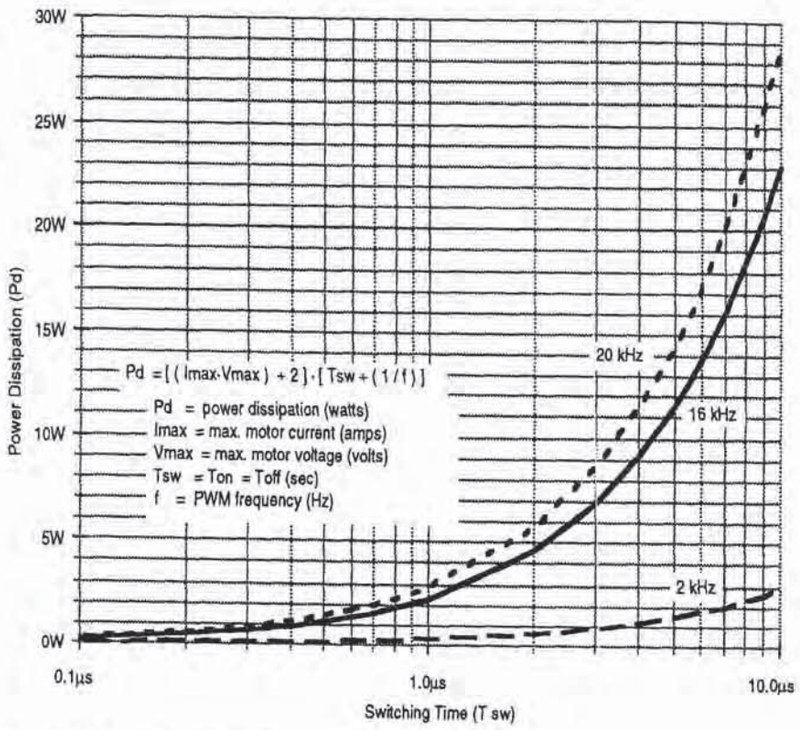


FIGURE 19.8 Power loss vs. switching speeds.

If the motor controller design has lead lengths that exceed 25 mm (1 in), their inductance values must be considered. An inductance value of 0.12 µH can be obtained in about 127 mm (5 in) of wire length with a #12 wire size, as shown in Fig. 19.9. This 0.12-µH value may not sound like much, but a plot (Fig. 19.10) of power line inductance versus switching transition times shows that just 0.1 µH of lead inductance can generate a 10-V voltage spike (at 20-A motor current) when switching edges are about 0.2 µs. Therefore, if fast switching edges are to be used, as with a 20-kHz operating frequency, a capacitor filter network is mandatory near the motor's power lead and controller. This filter network actually has to smooth out both the 20-kHz, and the much higher frequencies associated with the 200- to 1000-ns switching edges. To summarize, when fast switching edges are generated, filter networks are mandatory, or the hookup wiring will act as an antenna.

Motor Inductance Clamping. The kickback voltage from the motor's inductance also has to be contained, as the motor's inductance value will be dominant over the hookup wiring. This high-energy-laden voltage spike must be dealt with or the power transistor will enter an

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US PATENT NO. 6,557,540

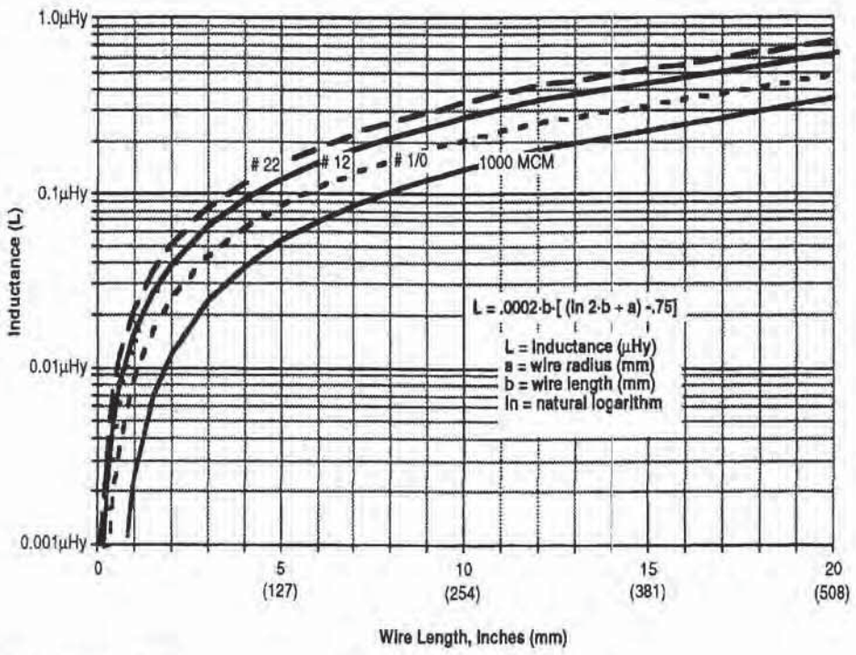


FIGURE 19.9 Line Z.

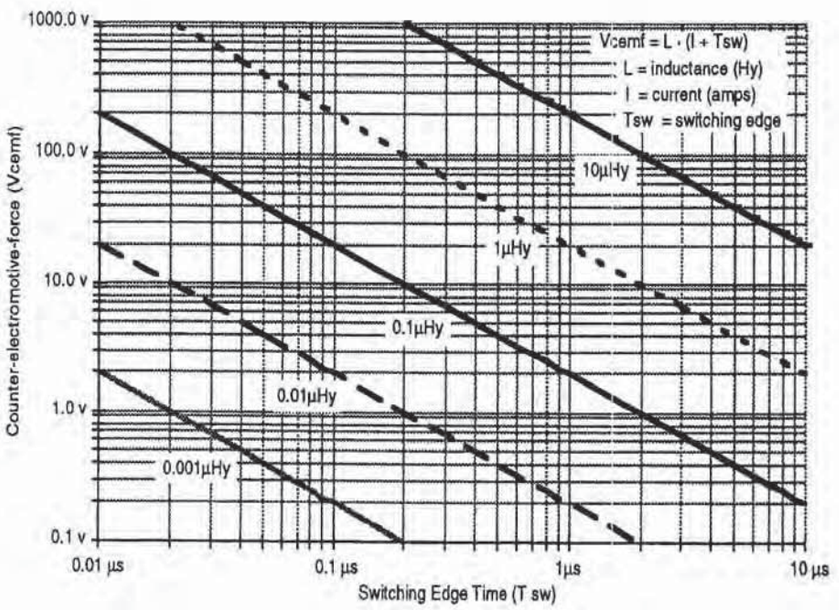


FIGURE 19.10 L di/dt effects.

avalanche or second breakdown mode, which is something to be avoided. The energy content of the motor's inductive kickback voltage is mainly determined by the motor's current, the motor's internal and external wire inductance value, and the rate at which the current is switched off. A high-current-rated freewheeling rectifier network can be connected across the controller's battery and motor terminals to clamp the motor's inductive kickback voltage spike. The reverse recovery times of these freewheeling rectifiers do affect the power transistor's switching performance.

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US PATENT NO. 6,557,540

When operating at a frequency of 20 kHz or a 50-μs repetition rate, the motor's inductance value is large enough to keep the freewheeling rectifier in a forward conduction mode until the next 50-μs cycle occurs. For example, if the PWM is set to a 25 percent duty cycle, the power transistor is on for the first 12.5 μs, and the freewheeling rectifier is in conduction for the remaining 37.5 μs. This means that when the next cycle occurs, the power transistor is switching on while the freewheeling rectifier is still in conduction. The result is that the power transistor and freewheeling rectifier conduct a high level "shoot-through" current spike. The magnitude and duration of this current spike are directly related to the rectifier's reverse recovery time rating, lead inductances, and the battery supply source impedance at the rectifier and power transistor location. Selecting freewheeling rectifiers with a "soft" reverse recovery characteristic will help minimize the shoot-through current problem.

The motor's back-emf, is produced from the rotating armature in a magnetic flux from the permanent magnets (PM). The PM motor's back-emf with a fixed mechanical load is normally lower than its power supply voltage and should therefore not exceed the voltage ratings of the power transistor.

Motor Current. When the motor's inductive generated transients are safely contained, one needs to deal with stalled and shorted motor conditions. Note that the motor's copper windings exhibit a positive temperature coefficient of 0.00393 per °C. Therefore, a 0.25-ohm motor resistance value at 25 °C room temperature would be about 0.18 ohm at –40 °C. Using the 20-A motor as the load, the maximum stalled or locked rotor current can be calculated as shown below.

$$I_{max} = E_{max} \div R_{mtr} = 14.4 \div 0.18 = 77 \text{ A}$$

where E_{max} = maximum power supply voltage
R_{mtr} = minimum motor resistance*

Power Transistor Specifications. When the maximum motor current has been established, the power transistor device specifications can be determined. In this case, the device needs an average current rating of at least 77 A, but the prime consideration for reliable power transistor operation is its worst-case heat dissipation.

The worst case scenario would include maximum values for the supply voltage, ambient temperature, and motor current. A maximum junction temperature of 150 °C for the power transistors is used as a maximum point. It should be noted that the reliability will increase by about one order of magnitude for each 10 °C drop in the power transistor's junction temperature. The following equations calculate the power transistor's maximum allowable heat dissipation for use in an 85 °C environment using a 2.7 °C/W heatsink unit and a 1 °C/W junction to case power FET thermal resistance.

$$PD_{max} = (TJ_{max} - TA_{max}) \div (R\text{ø}JC + R\text{ø}CS + R\text{ø}SA)$$

$$= (150 - 85) \div (1 + .1 + 2.7)$$

$$= 17.1 \text{ W}$$

where TJ_{max} = maximum allowable junction temperature
TA_{max} = maximum ambient temperature
$R\text{ø}JC$ = junction to case thermal resistance
$R\text{ø}CS$ = case to heatsink interface thermal resistance
$R\text{ø}SA$ = heatsink to ambient thermal resistance

* Formula for calculating the motor's cold resistance value is $R_{mtrcold}$ = [(delta temp. × R_{coef}) R_{mtr}] + R_{mtr} = [(–65 × 0.00393) × 0.25] + 0.25 = 0.18.

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US PATENT NO. 6,557,540

After the maximum transistor power dissipation is known, the maximum forward voltage drop, VDS_{on}, and ON resistance can be calculated as shown:

$$VDS_{on} = PD_{max} \div I_{max}$$
$$= 17.1 \div 77$$
$$= 0.22 \text{ V at } 150 \text{ °C Tj}$$
$$RDS_{on} = VdS_{on} \div I_{max}$$
$$= 0.22 \div 77$$
$$= 0.003 \text{ ohm at } 150 \text{ °C Tj}$$

In order to obtain a 0.003-ohm power FET ON resistance value at 150 °C Tj, several power FET devices will need to be paralleled. Bipolar devices were not even considered because of their high-current base drive requirements. Power FETs exhibit about a 75 percent ON resistance increase from a 25 to 150 °C rise in junction temperature. Therefore, six MTB75N05HD or equivalent devices in parallel would be required to achieve 0.003-ohm total ON resistance value when operating in an 85 °C ambient with a 2.7 °C/W heatsink. Six devices may not be economically attractive, so a larger heatsink may be more desirable. If the heatsink's thermal resistance is lowered to 1 °C/W, an ON resistance of 0.005 ohm would be required, which is three MTB75N05HD or equivalent paralleled devices. This is an example of the cost tradeoff between heatsink size and power FET ON resistance. It also points out that the power FETs ON resistance must be considered at the actual worst-case operating temperatures.

Short Current Protection. Some form of overcurrent protection would minimize power transistor failures in the event of a shorted motor, as might happen if the motor leads touched each other. Assuming motor's internal resistance will always present at least 90 percent of its nominal value, the shorted motor leads is the worst possible overcurrent condition. If three 75-A-rated power FETs are used in parallel, they would be capable of conducting well over 200 A. This current level may clear a 30-A fuse or activate a circuit breaker before the power FETs start to seriously overheat. In some wiring systems a *fuse link* is used to protect against a catastrophic wiring harness fire. The fuse link is usually just a short piece of copper wire that is encased in flame resistant insulation and is one wire-size smaller than the motor harness wire. The fuse link would probably not clear before the failed power transistors had shorted and blown open. If a fuse is selected that is slightly larger than the motor's normal run current, the power transistors may be capable of sustaining the shorted load current levels long enough to blow out the fuse before the transistors fail. A more reliable shorted load protective concept may still be necessary.

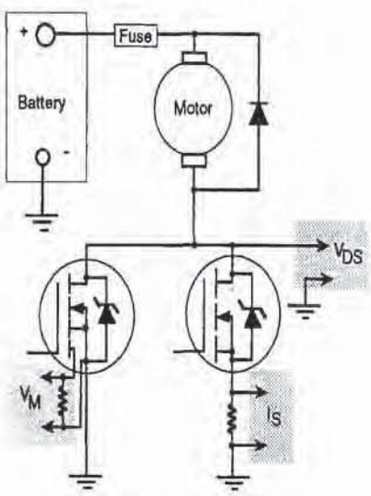


FIGURE 19.11 Load current sensing.

It may be desirable to latch off the power transistor during shorted load conditions. Load current sensing can be accomplished in three ways, as shown in Fig. 19.11. The voltage drop across the power transistor can be monitored, and, when a certain threshold is reached, the gate drive is latched off until the next operating cycle. Another method is to insert a series resistor in the source's path and monitor the voltage drop across the resistor. An internal current-sensing power FET device can also be substituted for one of the power FETs, with its current mirror voltage representing a portion of the total load current.

The method of measuring the transistor's forward ON voltage does have some inher-

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US PATENT NO. 6,557,540

ent drawbacks. The drain-to-source measurement circuit has to be synchronized with the gate drive voltage, and the forward voltage will change by at least 2 to 1 at a nominal current level over a –40 to +175 °C junction temperature range. The temperature drift of 2 to 1 means that, at 175 °C, a 60-A overcurrent would read the same as 120 A at –40 °C. Therefore, at high temperatures, the overcurrent detector may falsely shut down with a stalled motor, and, at cold temperatures, the overcurrent may not shut down at all. The most accurate current measurement method is to use a series resistor added in the high current path.

A single-chip microcontroller could greatly enhance the operation of this design by allowing more functionality. For example, a stalled motor condition could be detected and a preset shutdown mode invoked to protect the motor against burnout. The MCU would include internal A/D for current detection directly from the current-sensing resistor or from the internal current-sensing power FET. The PWM signal would also be generated from the MCU. Figure 19.12 shows an MCU-based speed control conceptual design that uses an 8-bit single-chip microcomputer.

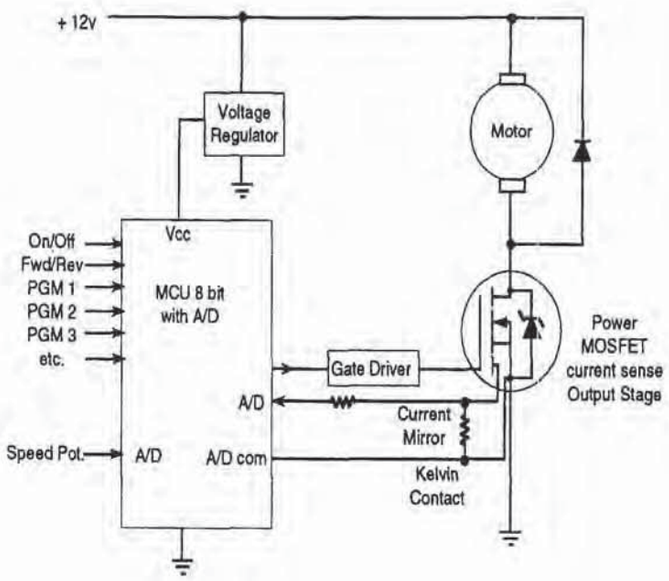


FIGURE 19.12 MCU motor control.

19.3 AIR CONDITIONER/HEATER CONTROL

Electronic controls allow an automatic climate control system to be designed for the comfort of the driver and passengers. An upscale automatic HVAC (heater and air conditioning) system, Fig. 19.13, may utilize an MCU to compute the most effective heat and ac flow rates. The heater and ac evaporator fan motor speed control designs can be similar to the WW variable speed design previously discussed if PM motors are selected. Brushless motors (BLMs) offer several features for this application. The BLM offers no brush-to-commutator noise. This feature alone is desirable in luxury vehicles, and, when one considers the increased mechanical reliability gained by the elimination of the brush-commutator assembly, the BLM cost-to-performance ratio seems more reasonable.

19.3.1 BLM System

If a motor-driven load requires closed-loop speed and direction control, the cost disadvantage of a BLM system becomes less significant. Figure 19.14 shows a typical permanent magnet brush motor compared with a BLM with both motors utilizing speed and directional control

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US PATENT NO. 6,557,540

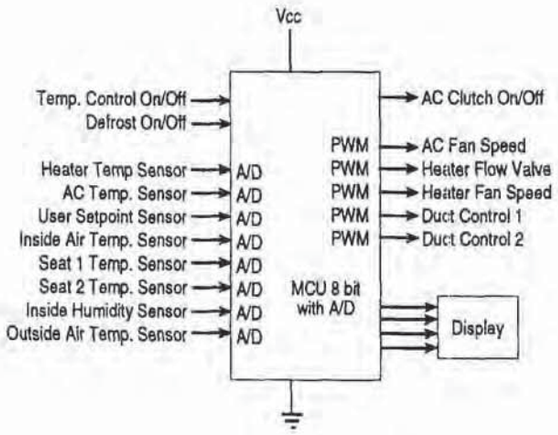


FIGURE 19.13 Climate control system.

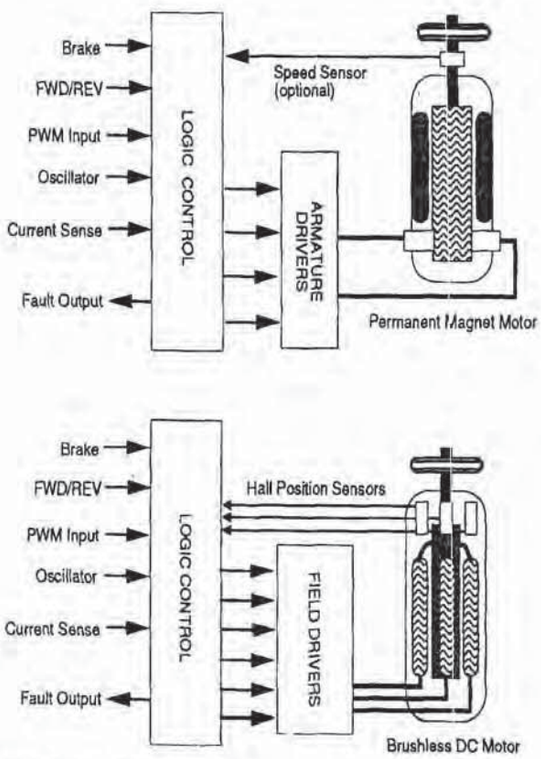


FIGURE 19.14 PM vs. BLM.

networks. Note that the BLM system uses only about one-third more drive electronics than the PM motor.

The BLM drive electronics may consist of a BLM linear bipolar integrated circuit, such as an MC33035 or similar device, that not only contains logic gates for commutation timing, but also includes internal driver stages capable of driving either power bipolar or power FET output transistors, or the MCU itself can be used to generate the necessary BLM control signals. One possible system may include the MC33035 linear bipolar IC and six power FETs. Three of the power FETs are internal current-sensing types. These current-sensing FET signals are conditioned and fed back to the MCU and to the linear IC as signals representing a motor

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US PATENT NO. 6,557,540

normal condition or a motor overload condition. When the MCU detects a motor overload condition, a predetermined course of action takes place. First, the overload is counted as an event. The MCU timer allows the overload to continue for a length of time depending on how much time has elapsed from the previous overload. This is done to allow for the thermal heat buildup that will occur when the motor is overloaded or stalled on a repetitive basis, enabling the motor to have a high starting torque when cold, but to protect the windings, bearings, and drive electronics when the motor's temperature has become dangerously high due to repeated starts with a faulty load. The MCU's EEROM is used to remember user settings such as temperature preferences, as well as for interfacing into a multiplex data network.

19.3.2 Clutch Coil Control

The HVAC system also controls the ac clutch coil, plus other small duct motors or solenoids. Care must be taken when switching these highly inductive loads with power transistors. Figure 19.15 shows several possible voltage clamping networks to prevent damage to the power module when the inductive loads are switched off. The following formula should always be on hand and used when designing power-switching circuits that drive coils or solenoids.

$$V_{pk} = L\ (Di/Dt)$$

where V_{pk} = maximum inductive generated voltage
Di = coil's peak current level when switched off
Dt = switching time of the coil's current level

A typical air-conditioner clutch coil may exhibit 15 mH of inductance and 3 ohms resistance (or 4.8 A of current), and, when operating from a 14.4-V battery, would generate about 720 V when switched off in 100 μs (V_{pk} = 0.015 × (4.8/.0001). Needless to say, this would severely stress or destroy a typical 50-V-rated power transistor. Some form of voltage clamping is required.

A fully automatic HVAC system can monitor the inside and outside air temperatures. The MCU program can monitor these inputs plus the user's temperature setpoint and then compute the heater and/or ac fan motor's speed plus heater core flow rate and/or ac clutch to achieve the desired inside temperature. The HVAC MCU software design can be implemented using fuzzy logic rules rather than standard control algorithms or massive lookup tables.

19.3.3 Fuzzy HVAC Control

Fuzzy logic design can be used to develop an HVAC control program. In general terms, a fuzzy logic design allows the software programming of an MCU-based controller to be simplified and is especially useful for complex nonlinear control dynamics. At this time, standard MCU devices can be used for fuzzy programming. Future MCU devices will probably be available that will enhance a fuzzy program. Programmers who can easily understand how to compose assembly language programs for converting various input signal conditions into the required output signals may not contemplate a fuzzy logic design, but, for many system designers, the fuzzy program method will shorten the software design and can allow the program design to be evaluated with the use of fuzzy development tools even before the hardware has been constructed.

A fuzzy logic implementation using an 8-, 16-, or 32-bit single-chip microcontroller (MC68HC11, MC68HC16, MC68332, etc.) requires the control system's behavior to be well defined from an intuitive sense. The designer defines the input conditions and desired output in a subjective manner. For example, if the auto's inside temperature is 25 °C, most people would probably find that value comfortable. In fuzzy terms, 25 would be 100 percent true for an input

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