Electronic Transmission Control

In complicated traffic situations, unfamiliar surroundings, or poor weather conditions (e.g., heavy rain, snow, or fog), manual gear changing can distract car drivers to such an extent as to create situations that are difficult to control. This also applies to the annoying, incessant process of engaging and disengaging the clutch when driving in stop-and-go traffic. Automatic transmissions with electronic control assist drivers in these and other traffic situations so that they can concentrate fully on the road conditions and what is happening around them.

Drivetrain Management

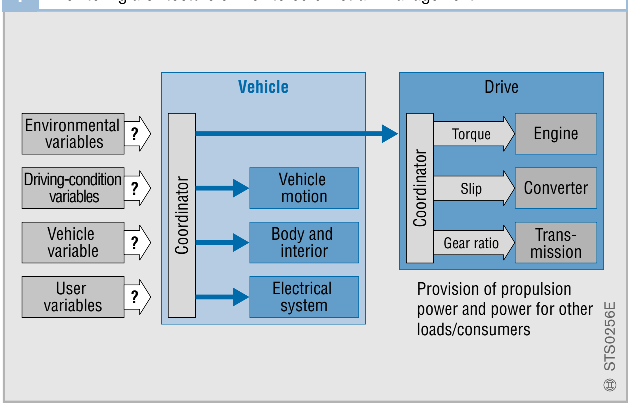
As the number of electronic systems in the vehicle increases, so too does the complexity of the overall network of the various ECUs. Controlling such networked structures requires hierarchical order concepts, such as, for example, the "Cartronic" system from Bosch. Coordinated drivetrain management is integrated as a substructure in the "Cartronic" system. It facilitates optimally matched management of the engine and transmission in the various vehicle operating conditions.

The engine is operated as often as possible in the fuel-saving ranges of its program map. If the driver adopts a sporty driving style, however, the high, less economical speed ranges are increasingly utilized. Such a situation-dependent mode of operation presupposes on the one hand that the driver command is recognized and on the other hand that its implementation is left to the electronic drivetrain-management system and a higher-level driving strategy. When the driver presses the accelerator pedal, the system interprets this action as an acceleration request. From this request, the drivetrain-management system calculates the conversion into torque and engine speed and implements them. In order for such a strategy to be implemented, it is essential for the system to be equipped with an electrically actuated throttle valve (drive by wire).

Figure 1 shows the organizational structure of drivetrain management as part of the overall vehicle structure. The vehicle coordinator forwards the requested propulsion movement to the drivetrain coordinator while taking into account the power requirements of other vehicle subsystems (e.g., body or electrical-system electronics). The drivetrain coordinator distributes the power demand to the engine, converter, and transmission. In the process, the various coordinators may also have to solve conflicts of interest that arise in accordance with defined priority criteria. A whole range of different external influencing factors (such as environment, traffic situation, vehicle operating status, and driver type) plays a role here.

The Cartronic concept is based on an object-oriented software structure with physical interfaces, e.g., torque as an interface parameter of drivetrain management.

1 Monitoring architecture of monitored drivetrain management



K. Reif (Ed.), *Automotive Mechatronics,* Bosch Professional Automotive Information,
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Market Trends

The statutory requirements relating to fuel consumption and exhaust-gas emissions will play a significant role in transmission development over the next few years. There follows a brief comparison of the requirements of the main markets for this purpose.

ACEA, JAMA and KAMA

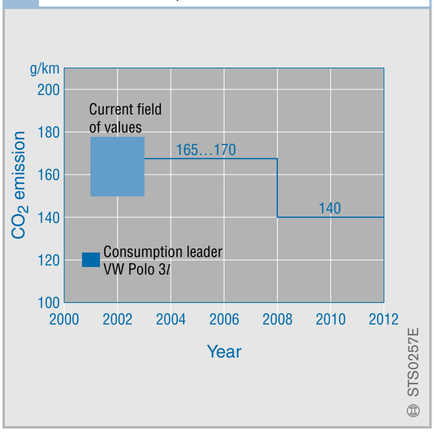
The ACEA (Association des Constructeurs Européens d'Automobiles, i.e. Association of European Automobile Manufacturers) has agreed to reduce the corporate average in CO_2 emissions in the period from 2002 to 2008 from 170 mg CO_2 to 140 mg CO_2 (Figure 1).

The Japanese and Korean manufacturers' associations (JAMA and KAMA) have adopted the same limits for the year 2009. In order for this target to be achieved, the next few years will see an increased acceptance of transmission types such as the 6-speed transmission CVT (continuously variable transmission) and AST (automated shift transmission).

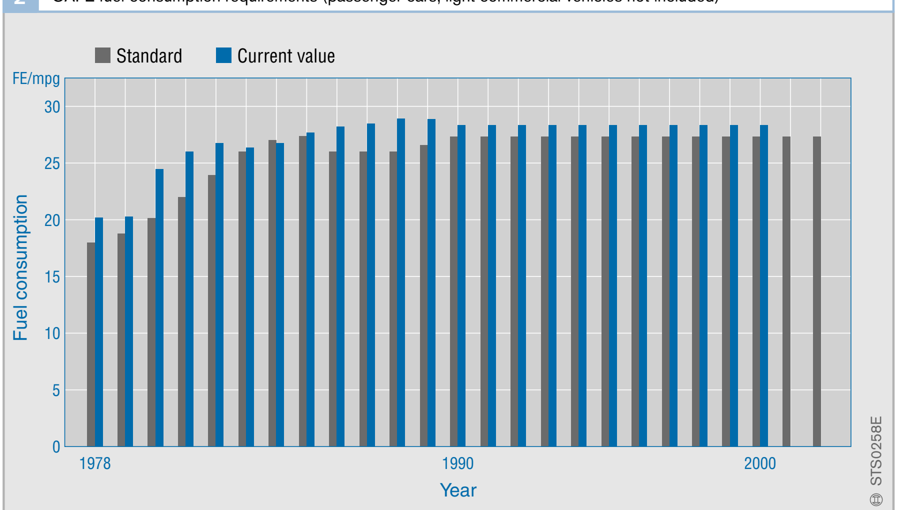
CAFE Requirements

In contrast to Europe, the USA, the most important market for automatic transmissions, has seen no change in the CAFE fuel-consumption requirements (corporate average fuel efficiency) since 1990 (Figure 2). All attempts to bring about a tightening of these requirements have proven unsuccessful.

1 CO_2-emission requirements



2 CAFE fuel-consumption requirements (passenger cars, light commercial vehicles not included)



Control of Automated Shift Transmission AST

Requirements

Current market developments reveal a marked trend towards an increase in in-car safety and operating comfort and convenience. This is accompanied by an increase in vehicle mass and in the final analysis an increase in fuel consumption. The emission guidelines laid down by legislators (140 g/km CO_2 by 2008) only intensify the situation.

The automated shift transmission (AST) combines the advantages of a manually shifted transmission with the functions of an automatic transmission. The automated version of the classic manual transmission is characterized by its high efficiency. Slip losses do not occur as in conventional torque-converter transmissions.

Specific fuel consumption in the automatic mode is below the low level of the manual transmission.

AST development is founded on the knowledge and findings gained in connection with electric-motor clutch management (ECM).

Electric-Motor Clutch Management (ECM)

Application

Following initial experiences with hydraulic clutch management, users are now concentrating their efforts on the use of electric motors as clutch actuators in the small-car segment. This new approach allows for savings to be made on costs and weight, as well as providing a higher degree of integration. Corresponding ECM systems are used in the Mercedes A-Class, the Fiat Seicento, and the Hyundai Atoz.

Design and Operating Concept

The most important step in minimizing costs was switching from a hydraulic actuator to an electric-motor actuator. This step removed the need for a pump, an accumulator, and valves as well as the need for a travel sensor in the clutch-release system. Instead the clutch-travel sensor is integrated in the electric-motor actuator.

The small electric motor offers a low power density in comparison with a hydraulic pump and accumulator. The electric-motor actuator is therefore only suitable if it can achieve sufficiently short declutching times for rapid gearshifts.

Shifting without Torque Correction

In the conventional system without torque correction (Figure 1), the clutch torque is far in excess of the engine torque. The reason behind this is that the dry clutch, which has to transfer at least the engine torque under all extreme conditions, normally offers a reserve of 50 to 150%. The engine torque drops when the driver wishes to change gear and at the same time takes his foot off the accelerator pedal. Operating the shift lever initiates the intention to change gear, and the clutch must now be moved from the fully closed to the fully open position. This defines the declutching time.

If the declutching time is too long, the clutch will still transfer torque while the next gear is being synchronized, a process which may result in rattling or damage to the transmission.

1 Shifting without torque correction

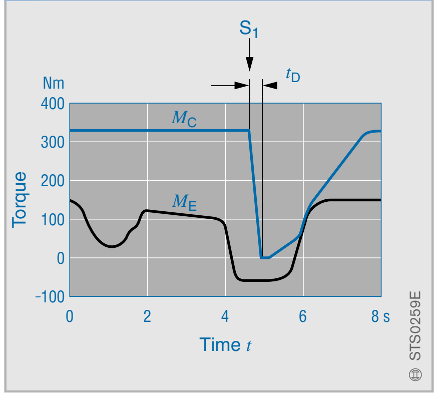


Fig. 1
M_C Clutch torque
M_E Engine torque
t_D Declutching time
S_1 Signal for gearshift command

Shifting with Torque Correction
The most technically sophisticated solution for avoiding transmission damage is to combine a reduced-force clutch with torque correction.

Figure 2 shows the shifting operation with torque correction, in which the clutch torque is only marginally above the engine torque. When the driver takes his foot off the accelerator pedal to change gear, the clutch torque drops as well as the engine torque. When the intention to change gear is initiated, the clutch is thus already almost open and the remaining declutching operation follows very rapidly.

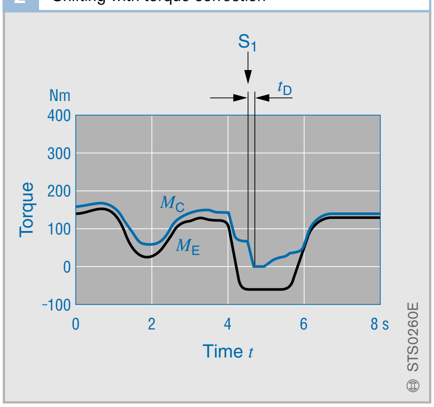
Figure 3 features a schematic representation of **e**lectric-motor **c**lutch **m**anagement (ECM) as a partial-automation solution, and the automated shift transmission (AST) as the complete automation of the manual transmission, both as add-on systems.

Electric-Motor Automated Shift Transmission AST

Application

Today the AST is used primarily in the lower torque classes (e.g., VW Lupo, MCC Smart, Opel Corsa Easytronic, see also chapter entitled "Transmission Types"), where, in comparison to the fully automatic transmission, the cost benefit makes up for the downside of the interruption in tractive force.

2 Shifting with torque correction



Design and Operating Concept
The electric-motor AST features the automated clutch operation of the ECM system. With an additional electric-motor actuator for the transmission, the driver is able to change gear without there being a mechanical connection between the selector lever and the transmission (shift by wire).

In the case of the AST, all modifications to the transmission are to be avoided. This will enable the transmission manufacturer to mount either manual transmissions or ASTs on the production line. As the hardware for this system (e.g., for the Opel Corsa Easytronic), Bosch supplies the electric motors for clutch engagement, shifting, and selection (see chapter entitled "Transmission, Automated Shift Transmission"), and the ECU. Automated and cost-effective mass production is made possible by the use of standard components in all AST applications.

3 ECM and AST as add-on systems

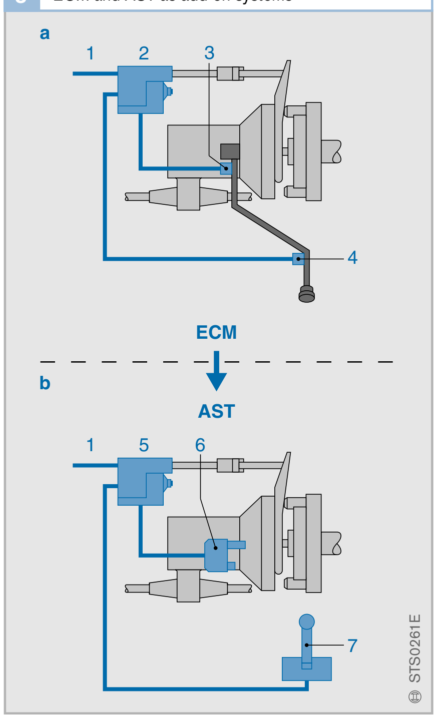


Fig. 2
M_C Clutch torque
M_E Engine torque
t_D Declutching time
S_1 Signal for gearshift command

Fig. 3
a ECM
b AST

1 Available signals
2 Clutch actuator with integrated ECM ECU
3 Gear recognition
4 Shift-intention recognition on shift lever
5 Clutch actuator with integrated AST ECU
6 Transmission actuator
7 Selector lever

Software Sharing

The vehicle manufacturer (OEM), the supplier and if necessary a system integrator share the AST software tasks. The operating system, signal conditioning, and the hardware-specific routines for activating the actuators are provided by Bosch. Bosch's extensive knowledge and experience in the field of automatic transmissions is also applied in establishing the AST target gears. This includes, among others, driver recognition, uphill/downhill recognition, cornering recognition, and other adaptive functions (see also chapter entitled "Adaptive Transmission Control, ATC").

The tasks of activating the transmission and coordinating the gearshift sequence (clutch, engine, transmission) are the responsibility of the OEM or the system integrator.

This also applies to clutch control, significant parts of which can be taken over from the ECM system. Each vehicle manufacturer brings its marque-specific philosophy regarding shifting time, shifting points, and shifting sequences to bear.

Shifting Operation and Interruption of Tractive Force

The basic problem associated with the AST is the interruption of tractive force. This is represented in Figure 4 by the "hollow" of the vehicle acceleration between the two shifted gears. In terms of what is required of the actuators, these phases can be divided into two blocks:

- phases which have an effect on the vehicle acceleration,
- phases which represent pure response times.

In the phases which have an effect on the vehicle acceleration, it transpires that a throttle action is needed because excessively quick changes in vehicle acceleration are felt to be unpleasant. The optimum interaction of engine, clutch, and transmission intervention results in the best possible performance.

The synchromesh can be supported for example by double-declutching. In the response times, however, the maximum speed of the actuators is demanded. It is important here that the synchromesh does not experience too hard a shock after the gear has been disengaged and the following rapid phase.

4 Phases of AST shifting operation

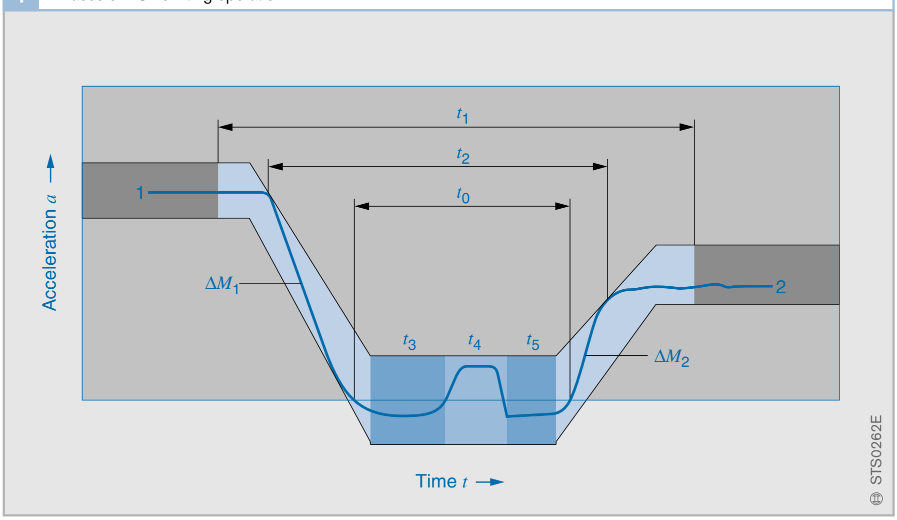


Fig. 4
1 Current gear
2 Next gear

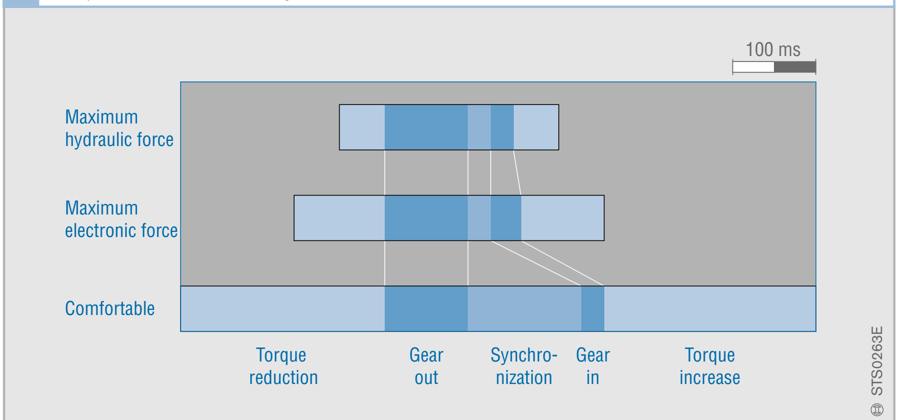
ΔM_1 Torque reduction
ΔM_2 Torque increase
t_0 Tractive-force interruption
t_1 Shifting operation
t_2 Acceleration
t_3 Disengage and select gear
t_4 Synchronization
t_5 Shift through gear

Figure 5 shows a comparison of the shifting times that can be achieved with a hydraulic system and an electric-motor system and the shifting time necessary for a comfortable shifting operation. The bar lengths equate to the times required for the individual phases and the same shading schemes are used.

When the capacity of the actuators is exploited to maximum effect, the electric-motor system only demonstrates a time disadvantage compared with its hydraulic counterpart in the clutch-operation phase. This could be reduced in particular in the torque-reduction phase by an intelligent control strategy such as torque correction or by the interaction of the engine and the clutch.

It is important to highlight here that there is hardly any difference between the two systems in the phases for response time, gear disengagement, and gear engagement. The response times are not extended practically, even in the case of comfortable shifting. However, the phases relevant to acceleration must be two to four times as long as in the extreme case, both for the hydraulic and the electric-motor actuator systems.

5 Comparison of achievable shifting times



6 Automated shift transmission (AST) in a diesel vehicle (example: system diagram)

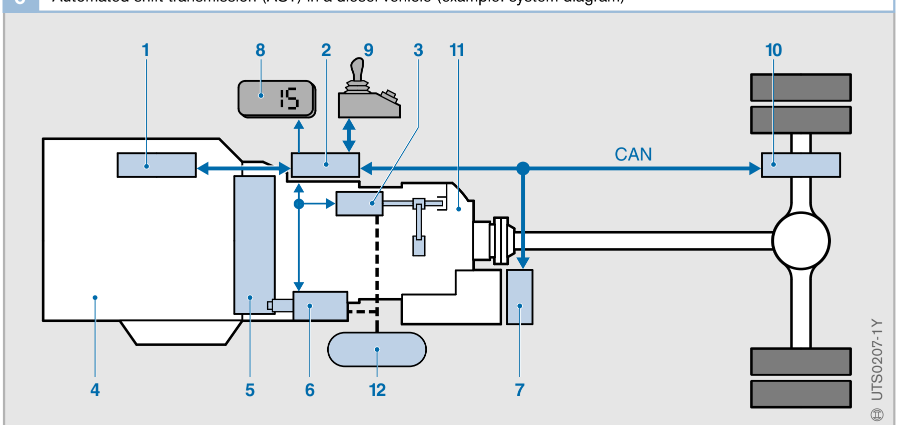


Fig. 6
1 Engine electronics (EDC)
2 Transmission electronics
3 Transmission actuator
4 Diesel engine
5 Dry interrupting clutch
6 Clutch servo unit
7 Intarder electronics
8 Display
9 Driving switch (selector lever)
10 ABS/TCS
11 Transmission
12 Air supply

—— Electrics
---- Pneumatics
—— CAN communication

Control of Automatic Transmissions

Requirements

The control system for an automatic transmission must fulfill the following essential requirements or functions:

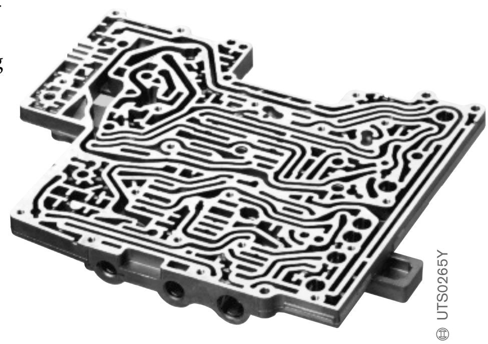
- always shifting the correct gear or setting the correct gear ratio as a function of assorted influencing variables,
- executing the shifting operation through adapted pressure characteristics as comfortably as possible,
- implementing additional manual interventions on the part of the driver,
- detecting maloperations, e.g. by preventing non-permitted gearshifts,
- providing ATF oil for cooling, lubrication and for the converter.

Current control systems are exclusively electrohydraulic in nature.

Hydraulic Control

The main function of the hydraulic-control system (Figures 1 and 2) is to regulate, boost, and distribute hydraulic pressures and volumetric flows. This includes generating the clutch pressures, supplying the converter, and providing the lubricating pressure. The housings of the hydraulic-control system are made from diecast aluminum and contain several precision-machined slide valves and electrohydraulic actuators.

2 Main control with hydraulic valves



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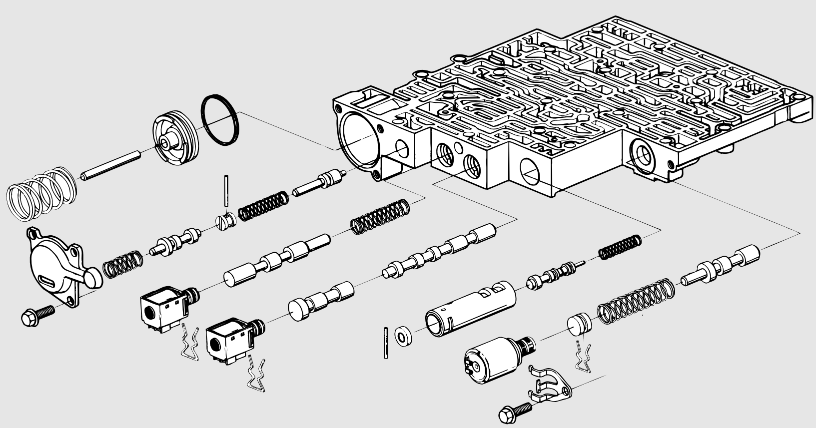
1 Exploded view of a hydraulic-control system (example: GM HYDRA-MATIC 4L60-E)



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Electrohydraulic Control
Due to their extensive range of functions, all modern automatic transmissions with four to six gears and continuously variable transmissions are exclusively controlled by electrohydraulic means. In contrast to earlier, purely hydraulic control systems with mechanical regulators, the clutches are activated individually by pressure regulators, which facilitate precise modulation and regulated overlapping gearshifts (without a one-way clutch).

Clutch Control
Clutch control is always performed with either pilot-controlled or directly controlled pressure.

Pilot Control
With pilot control, the required pressure and throughflow for rapid clutch filling are provided via a slide valve in the control housing. Pressure regulation is effected by pilot pressure acting on the sensing surface of the slide valve. An actuator generates this pilot pressure (Figure 3).

This results in greater degrees of freedom in the packaging and the use of standardized actuators, high dynamics, and small electromagnets.

Direct Control
With direct control, the required pressure and throughflow for rapid clutch filling are provided directly by the actuator (Figure 4).

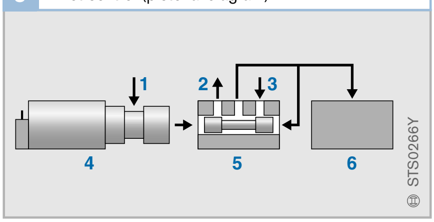
This results in a compact clutch-control system with reduced hydraulic sophistication.

Shifting-Sequence Control
Conventional Shifting-Sequence control
The following two shifting scenarios are examples of conventional control of a simple 4-speed automatic transmission with one-way clutches (Figure 5).

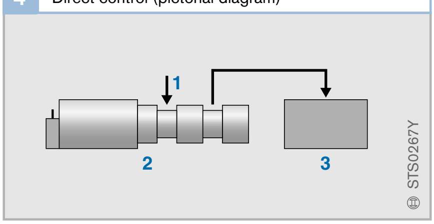
Upshift Under Load

Unlike in a manual transmission, throttling upshifts in an automatic transmission take place without an interruption of tractive force. The graphic in Figure 6 shows the time curve of the characteristic variables during an upshift into the direct gear (ratio 1). Shifting begins at time t_0: The clutch is filled with fluid and the friction elements are pressed against each other. The clutch transfers a torque from time t_1 onwards. As the clutch torque increases, so the torque supported at the one-way clutch decreases. The one-way clutch is released at time t_2. Now the engine speed begins to change. The clutch torque increases to t_3. The clutch slips up to t_4, after

3 Pilot control (pictorial diagram)



4 Direct control (pictorial diagram)



5 Load transfer with one-way clutch US

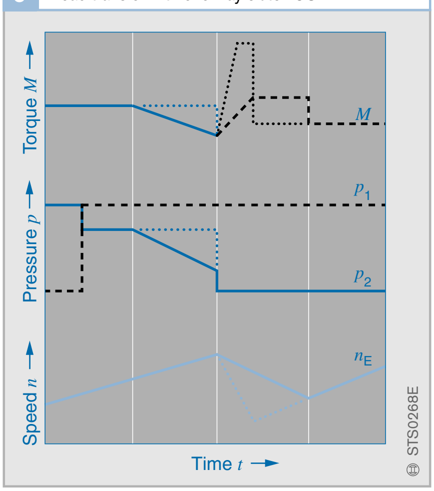


Fig. 3
1 Supply to actuator
2 Oil pan
3 Supply to slide valve
4 Actuator
5 Slide valve in control housing
6 Clutch

Fig. 4
1 Supply to actuator
2 Actuator
3 Clutch

Fig. 5
p_1 Pressure, cutting-in clutch
p_2 Pressure, cutting-out clutch
n_E Engine speed
M Torque

which it sticks. After the end of the shifting operation, the clutch pressure is controlled upward to a level of safety.

The speed difference between the engine and the transmission output speed remaining after t_4 is caused by the converter, which always operates with slip when locked up.

The progression of the output torque in the phase $t_1 ... t_4$ determines the gear-shift sophistication (ease of shifting). To ensure good shift quality, the clutch pressure must be set so that the output torque is between the level at $t < t_1$ and the level at $t > t_4$. The torque jump at t_4 should also be as low as possible.

The load on the friction elements is determined by the clutch torque and the slip time $(t_4 - t_1)$. It is clear here that controlling the shifting sequence always involves a compromise.

Downshift Under Load

In contrast to upshifts, downshifts take place with an interruption of tractive force. Figure 7 shows the time progression.

At t_0 shifting begins with the clutch being drained. Engine torque is no longer transferred from t_1 onwards and the engine revs up. At t_2 the synchronization speed of the new gear is reached and the one-way clutch is engaged; up to t_3 the converter slip is set according to the engine torque. The shifting operation is completed at t_3. The gear-shift sophistication (ease of shifting) is determined by the torque drop in the phase $t_0 ... t_1$ and depends quite significantly on the torque increase between t_2 and t_2.

All the shifting scenarios that occur are primarily controlled by the electronic system; the hydraulic system is left above all with the function of clutch power control.

In all newer transmissions (5-speed and 6-speed types), one-way clutches are replaced by regular clutches for weight reasons. However, during the shifting operations, they require overlap control

Fig. 6
p_C Clutch pressure
p_F Filling pressure
p_S Safety pressure
t_0 Start of shift
t_1 Start of torque transfer, clutch torque rises, one-way clutch torque drops
t_2 One-way clutch released
t_3 Clutch slips, clutch torque remains constant
t_4 Clutch sticks, clutch torque decreases, converter operates with slip

1 Output
2 One-way clutch
3 Clutch
4 Engine
5 Transmission output

6 Pressure, torque and speed curve during an upshift

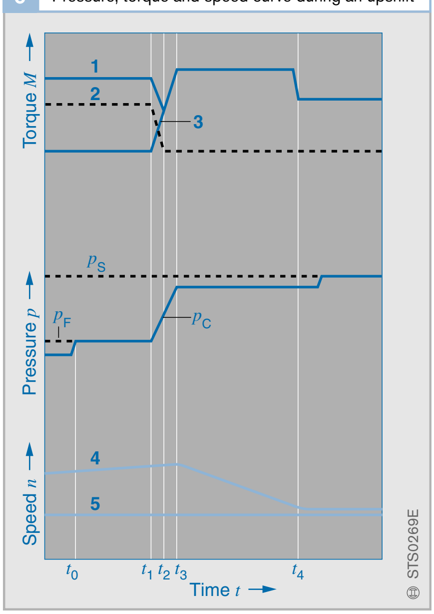
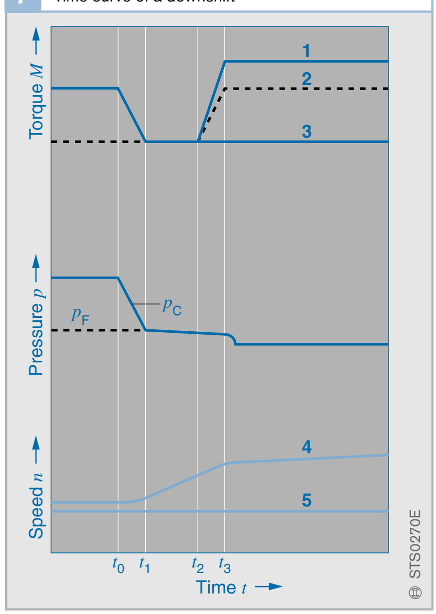


Fig. 7
p_C Clutch pressure
p_F Filling pressure
t_0 Start of shift, clutch drains
t_1 End of torque transfer, engine revs up
t_2 Synchronization speed of new gear reached, one-way clutch engaged, converter operates with slip
t_3 Shifting operation completed

1 Output
2 One-way clutch
3 Clutch
4 Engine
5 Transmission output

7 Time curve of a downshift



for the clutches (Figure 8). This means that while clutch 1 opens for gear x, clutch 2 must close for gear y. Since this type of control is very elaborate and time-critical, it is necessary to provide considerably higher computing power in the ECU than for the simple shifting sequences with one-way clutch shifts (see also chapter entitled "ECUs").

The most important features of overlap control are:

- low mechanical complexity,
- minimal space requirement,
- multiple use for different gear steps possible,
- high control precision for load transfer required,
- high software complexity for torque control,
- in event of incorrect control: excessive speed (engine races) or onset of a braking torque (extreme case: transmission blocking).

Adaptive Pressure Control

The function of adaptive pressure control is to achieve a consistently good shift quality over the entire working life of the transmission and the accompanying changes in the friction coefficients at the clutch surfaces. It also compensates for any potential deviation of the calculated torque or the torque transferred by the engine which can occur on account of changes to the engine or manufacturing tolerances.

In this case, an important role is played by pressure adaptation with the aid of the shifting times applied by the manufacturer. To this end, the applied shifting times are compared with the real shifting times that occur. If the measurements are repeatedly outside a prespecified tolerance range, the pressure parameters pertaining to the shifting operation are incrementally adapted. A distinction is made here between the fill time and the slip time of the clutch.

8 Overlap control US

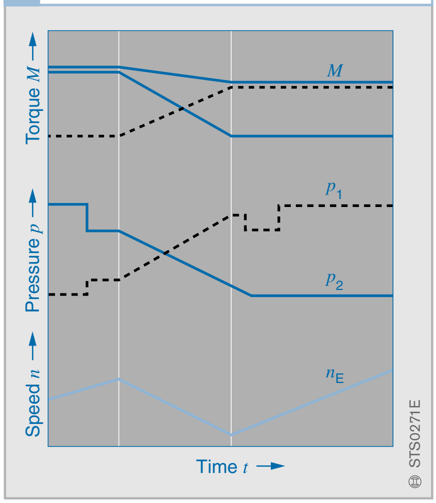
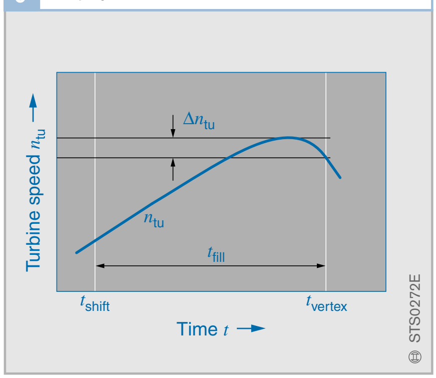


Fig. 8
p_1 Pressure, cutting-in clutch
p_2 Pressure, cutting-out clutch
n_E Engine speed
M Torque

9 Fill time

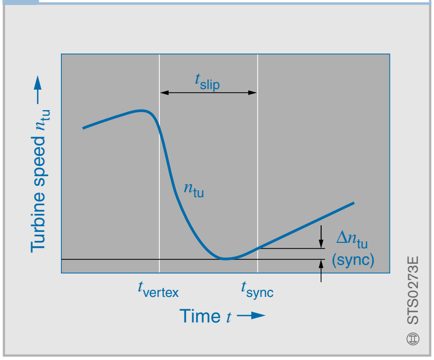


Fill-Time Measurement
The fill time t_{fill} (Figure 9) is the time from the start of the gearshift t_{shift} to the start of synchronization (a drop in speed is identified during the upshift [US]):

$$t_{fill} = t_{vertex} - t_{shift}$$

Slip-Time Measurement
The slip time t_{slip} (Figure 10) of the clutch is the time from recognition of the speed vertex (start of synchronization) to complete synchronization of the speed in the new gear.

10 Slip time



$t_{slip} = t_{sync} - t_{vertex}$

The speed thresholds used for measuring the slip time t_{slip} (Figure 10) are calculated in advance of the start of shifting, where the following relationship applies to upshifts: Start of fill-time measurement = start of shifting

Vertex: A decrease in the turbine speed n_{tu} by at least n_{tu} (vertex) revolutions is detected.

$n_{tu}\,(t-1) - n_{tu}\,(t) > n_{tu}\,(\text{diff})$

Synchronization speed: An increase in the turbine speed n_{tu} by at least n_{tu} (sync) revolutions is detected.

$n_{tu}\,(t) - n_{tu}\,(t-1) > \Delta n_{tu}\,(\text{sync})$

Pressure Correction

Pressure adaptation is only permitted within specific limits on account of operational reliability. The typical adaptation width lies in the range of ±10% of the modulation pressure calculated for the shift. The correction values are also still distinguished according to speed bands.

The adaptation values are stored in a non-volatile memory so that the optimum modulation pressure can be reapplied when the vehicle is restarted. The overall pattern of pressure adaptation can also be evaluated as a sign of changes in the transmission.

Shifting-Point Selection

Conventional Shifting-Point Selection

In the majority of automatic transmissions currently available, the driving program is selected using a selector switch or a button. The following driving programs are generally available in this respect:

- Economy (very economical),
- Sport, or
- Winter.

The individual programs differ in the position of the shifting points in relation to the position of the accelerator pedal and the driving speed. The Economy and Sport shifting maps of a 5-speed transmission are used here as examples (Figure 11).

If the current driving speed or the accelerator-pedal position corresponding to the driver command (accelerator-pedal value) intersects the shift curve, a gearshift is triggered. A requested gearshift can be either canceled or converted into a double shift within a specific period of time (which depends on the hydraulic system of the automatic transmission)

For example, the driver is driving in fifth gear on an interstate highway and would like to overtake. To do so, he presses the accelerator pedal to the floor, whereupon a downshift is requested.

11 US and DS characteristics in Economy mode (XE) and Sport mode (XS)

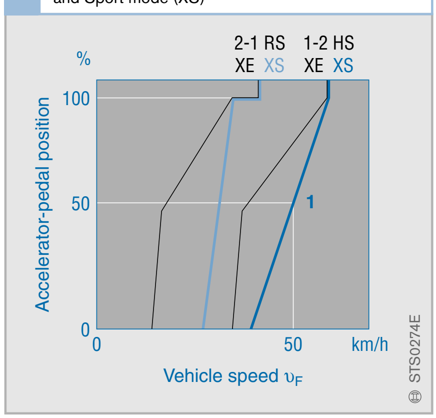


Fig. 11
1 Upshift
XE Economy mode
XS Sport mode

When the accelerator pedal is firmly pressed to the floor, the 4-3 DS shift curve is intersected directly after the 5-4 DS shift curve, and a 5-3 double downshift is performed instead of a sequential downshift. Special shifting points for kickdown (forced downshift) allow the maximum possible engine power to be utilized at this point.

Adaptive Transmission Control (ATC)

All newer transmission-control systems have – instead of active driving-program selection by the driver – software which enables the driver to adapt to the special ambient conditions while driving. This includes first and foremost driver-type recognition and driving-situation recognition. Examples which are currently in use are **a**daptive **t**ransmission **c**ontrol (ATC) from BMW and the **d**ynamic **s**hift **p**rogram (DSP) from Audi.

Driver-Type Recognition
A driver type can be identified by means of an evaluation of the actions he or she performs. This includes:
- kickdown operation,
- brake operation, and
- restriction via selector lever.

For example, the kickdown evaluator counts the number of times the driver engages kickdown during a presettable period of time. If the counter exceeds a specific threshold, the driver-type recognition facility selects the next, more sporty driving program. It automatically switches back to a more economical driving program once this time has elapsed.

Driving-Situation Recognition
For driving-situation recognition, different transmission-control input variables are linked to conclusions about the present driving condition. The following situations can generally be recognized:
- uphill driving,
- cornering,
- winter operation, and
- ASC operation.

Uphill Driving
Recognition of uphill driving by comparing the current acceleration with the requested acceleration by way of the engine torque, results in upshifts and downshifts at higher engine speeds and thus prevents gearshift hunting.

Cornering
This facility uses the difference in wheel speeds to calculate whether the vehicle is in a curve or bend. With active cornering recognition, requested shifts are delayed or prohibited in order to increase vehicle stability.

Winter Recognition
Winter operation is recognized on the basis of slip detection from analysis of the wheel speeds. This serves primarily to
- prevent the wheels from spinning and
- select a higher gear during starting so that less torque is transferred to the drive wheels, thereby preventing premature wheel spin.

ASC Operation
If the system detects while driving that the ASC ECU (**a**nti-**s**lipping **c**ontrol or **t**raction **c**ontrol **s**ystem, TCS) is in control mode, requested gearshifts are suppressed in order to support the ASC function.

Engine Intervention

Application

A precisely controlled time characteristic of engine torque during the shifting operations of an automatic transmission offers the possibility of optimizing transmission control with regard to gear-shift sophistication (convenience), clutch service life, and transferrable power. The engine management system implements the torque command (reduction) of the transmission control by retarding the moment of ignition.

The theoretical principles, processes, and measurement results are presented using the example of engine intervention in ignition.

Symbols and Abbreviations

C	Spring stiffness of drivetrain
i	Gear ratio
J	Mass moment of inertia
k	Constant
M	Engine torque
n	Rotational speed
q	Specific lost work
Q	Lost work
t	Time
W	Running resistance
x	Spatial coordinate
δ	Temperature
ω	Angular velocity
Φ	Angle of rotation
φ	Angle of rotation, linearized

Indices

O	Output
V	Vehicle
limit	Permitted limit value
C	Clutch (friction element)
kin	Kinetic share
E	Engine (transmission input)
red	Reduced value
s	Slip time
com	Share of combustion energy (engine torque)
•	Reference variable
1	Clutch drive side
2	Clutch output side

Requirements

The ever-increasing demand for more economical fuel consumption in motor vehicles dictates to a significant degree the development objectives in the field of automatic transmissions as well. In addition to measures for improving the efficiency of the transmission itself (such as, for instance, the torque converter lockup clutch), these objectives include introducing transmissions with more gears. However, additional gear steps inevitably call for increased shift frequency. This in turn results in increased demands placed on gear-shift sophistication (convenience) and the load capacity of the friction elements.

Engine intervention takes into account both requirements and institutes an additional degree of freedom for controlling an automatic transmission. “Engine intervention” covers all those measures which allow the engine torque generated by the combustion process during the shifting operation in the transmission to be specifically influenced and in particular reduced. Engine intervention can be used in both upshifts and downshifts.

The primary aim of engine intervention in upshifts is to reduce the lost energy that occurs in the friction elements during the shifting operation. This is done by reducing the engine torque during synchronization without interrupting the tractive force. The margin acquired in this process can be used to:

- Increase the service life by shortening the slip time (if all other operating parameters in the transmission, such as clutch pressure and number of plates, remain unchanged).
- Improve the convenience by reducing the clutch torque, brought about by lowering the clutch pressure during the slip phase.
- Transmit higher power, provided the mechanical strength of the transmission permits this; in most cases, however, the power loss in the clutches is the limiting factor.

Naturally, it is also possible to adopt a sensible combination of these measures within the framework of the specified margin.

The aim of engine intervention in downshifts is to reduce the jolt which occurs when the one-way clutch or a friction element engages at the end of the synchronization processes. This results in

- improved convenience and
- supported and improved synchronization in transmissions without one-way clutches.

Interventions in the Mechanical Shifting Sequence

The following explanations illustrate which possibilities present themselves for intervention in the mechanical shifting sequence. The individual phases of upshifts and downshifts are described in the section entitled "Shifting-Sequence Control".

Upshifts

Engine intervention is discussed using the example of an upshift from the direct gear ($i = 1$) to overdrive ($i < 1$). The following simplifications serve to illustrate the physical relationships more clearly:

- The influence of the torque converter is disregarded.
- There is no overlap of friction elements, i.e., only *one* friction element participates in the gearshift.
- The engine torque remains constant during the gearshift, thereby providing linear speed characteristics.
- The vehicle speed during the gearshift is taken to be constant.
- The heating of the friction linings by briefly successive shifting operations is disregarded.

Upshifts take place without an interruption of the tractive force. Synchronization of engine and transmission takes place via a friction element in slipping-intervention operation. The following relative speed ensues between the drive and output side of the clutch:

$$\Delta\omega = \omega_1 - \omega_2 \qquad (1)$$

In relation to the lost energy which must be absorbed or forwarded by the friction elements during the shifting operation, the following equation applies:

$$Q = \int_0^{t_S} M_C\,(t) \cdot \Delta\omega(t) \cdot \mathrm{d}t \qquad (2)$$

Furthermore, the angular-momentum principle applies to the drive and output sides of the clutch. For the rotational masses of the drive side:

$$\Delta\omega = \omega_O \cdot \frac{1-i}{i} + \frac{M_E - M_C}{J_O} \cdot t \qquad (3)$$

Under the above-mentioned preconditions, this produces:

$$\Delta\omega = |\omega_E - \omega_O| = \omega_E\,(t{=}0) - \omega_O + \omega_E \cdot t$$

or

$$Q = M_C \left[\omega_O \cdot \frac{1-i}{i} \cdot t_s + \frac{M_E - M_C}{J_E} \cdot \frac{t_S^2}{2} \right]$$

From (1), (2), and (3), this produces for a time-constant clutch torque the lost energy as a function of the shifting-sequence parameters.

$$J_1 \cdot \dot{\omega}_1 = M_E - M_C$$

The slip time itself is dependent on the clutch and engine parameters, where

$$t_s = \frac{\omega_O}{|\dot{\omega}_E|} \cdot \frac{1-i}{i} = \omega_O \cdot \frac{1-i}{i} \cdot \frac{J_E}{|M_E - M_C|} \qquad (4)$$

This produces the lost energy to be absorbed by the friction element

$$Q = \frac{1}{2} \cdot \frac{M_C \cdot J_E}{M_E - M_C} \cdot \omega_O^2 \left(\frac{1-i}{i} \right)^2 \qquad (5)$$

i.e., the lost energy is dependent only on the clutch and engine torques, the driving speed, and the gear ratios.

When the clutch torque determined by (4) is applied in (5), this produces the lost energy as the sum of a share of the kinetic energy which is released when the rotational masses are braked to the synchronization speed and a share of the engine combustion energy:

$$Q = Q_{\mathrm{kin}} + Q_{\mathrm{com}} = J_{\mathrm{E}} \cdot \frac{\omega_0^2}{2} \cdot \left(\frac{1-i}{i}\right)^2 + M_{\mathrm{E}} \cdot t_{\mathrm{S}} \cdot \frac{\omega_0}{2} \cdot \frac{1-i}{i} \qquad (6)$$

Both these shares are roughly of the same order of magnitude. At speeds of $n = 3000$ rpm and typical values for the gear step and the engine-drag torque ($i = 0.8$, $J_{\mathrm{E}} = 0.3\ \mathrm{kg \cdot m^2}$, $M_{\mathrm{E}} = 100$ Nm, $t_{\mathrm{S}} = 500$ ms), this produces:

$Q_{\mathrm{com}}/Q_{\mathrm{kin}} \approx 1...4$

This clearly shows the possibilities of engine intervention for reducing the power loss in the friction elements.

A further significant aspect is derived from (6): Only the share of lost energy stemming from the combustion energy is dependent on the slip time t_{S}. The decisive factor is the product of the engine torque and the slip time. This means, however, that the slip time can be extended accordingly when the engine torque is reduced without an increase in the total lost energy. In actual fact, the wear of the friction elements even decreases with constant total lost energy when the slip time is extended. The temperature of the friction linings corresponds to the load on the friction elements.

Figure 12a shows the lost energy absorbed by the friction element as a function of the engine torque and the slip time. The maximum permitted lost energy Q_{limit} and the engine torque to be transferred during this gearshift determine the maximum slip time, for instance in accordance with point S. The maximum permitted energy Q_{limit} corresponds in accordance with (5) to the clutch torque determined by the slip time M_{Climit} (point 1 in Figure 12b).

To reduce the lost energy, the clutch torque in relation to point S would have to be increased and thereby the slip time shortened. However, this would lead in equal measure to

12 Lost energy Q_{com} (a) and clutch torque M_{C} (b) of a clutch as a function of slip time t_{S} and engine torque M_{E}

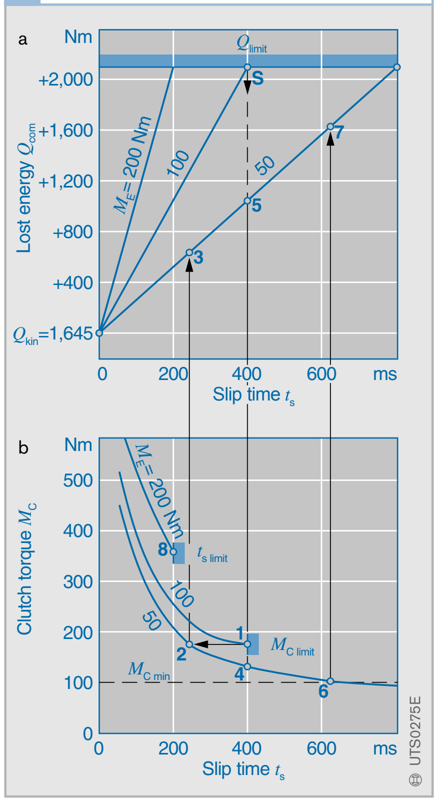


Fig. 12
a Lost energy Q_{com}
b Clutch torque M_{C}

M_{Climit} Maximum clutch torque
M_{Cmin} Minimum clutch torque
M_{E} Engine torque
Q_{limit} Maximum permitted lost energy

a reduction in gear-shift sophistication (convenience). A reduction of the clutch pressure is not permitted in this case, as otherwise Q_{limit} will be exceeded.

It is now easy to tell from Figure 12 which possibilities are offered by engine intervention. It is taken that the engine torque to be transferred $M_{\mathrm{E}} = 100$ Nm can be reduced during the slip phase to an average of 50%. When first the case of constant clutch torque (shift quality) is considered, reducing the engine torque to 50 Nm results in a shortening of the slip time from 400 ms to 245 ms (point 1 → point 2) with a simultaneous reduction in lost energy to 61% (point 3). If, on the other hand, the slip time is kept constant, the clutch torque can be reduced from 179 Nm to 128.5 Nm (point 1 → point 4) with a reduction in lost energy to 72% (point 5).

The maximum sensible slip time is then obtained if the minimum clutch torque M_{Cmin} during the gearshift does not drop below the value after the end of the shift. On the one hand, a smaller engine torque as a result of engine intervention would result in a deterioration in convenience; on the other hand, the clutch torque for safety reasons should at any rate be so large that the non-reduced engine torque can be transferred by the friction element after the end of the shift.

In this example, it is taken that the torque to be transferred at least by the clutch is 100 Nm in accordance with the engine torque (direct gear). This means that the slip time can be stretched from 400 ms to max. 625 ms (point 6), again with a simultaneous reduction in lost energy to 88% (point 7).

Finally, it can be gleaned from Figure 12 that even an engine torque of 200 Nm, which without intervention would require a maximum slip time of 200 ms with a minimum clutch torque of 360 Nm (point 8), can be traced back to the example with a torque of 100 Nm (point 1).

The results of this analysis are on the safe side in this respect because extending the slip time with constant lost energy results in reduced friction-lining temperature and thus protects the friction linings. Table 1 lists the numerical values for these examples.

Downshifts
In contrast to upshifts, downshifts take place in the throttling mode with an interruption of load. The engine is decoupled from the drivetrain and runs as a result of the torque it has generated up to the synchronization speed. Only after the one-way clutch or the friction element has engaged is the frictional connection re-established. The torque ratios when the synchronization speed is reached

1 Numerical values for text examples and Figure 12

M_E Nm	M_{redE} Nm	M_C Nm	M_E Nm	t_0 ms	Q/Q_{100} %
100	100	179	400	3740	100
100	50	179	245	2285	61
100	50	128.5	400	2693	72
100	50	100	628	3290	88
200	200	360	200	3740	100
200	100	179	400	3740	100

Table 1

determine the gear-shift sophistication (convenience) to a substantial extent.

For a better understanding of the characteristic relationships, damping in the drivetrain is disregarded in the following analysis. It also applies on the assumption that the overall vehicle dynamics can be reduced to engine mass, input-shaft rigidity, and vehicle inertia.

In the case of all the moments of inertia relating to the transmission output, the engine and the vehicle are governed by the following:

$$J_E \cdot \ddot{\Phi}_E = M_E,\ J_V \cdot \ddot{\Phi}_V = -W \qquad (9)$$

At the point when the one-way clutch engages, the drivetrain resembles a torsion damper (Figure 13, next page), and the motion equations are as follows:

$$J_E \cdot \ddot{\Phi}_E = c(\ddot{\Phi}_V - \ddot{\Phi}_E) + M_E \qquad (10a)$$

$$J_V \cdot \ddot{\Phi}_V = c \cdot (\ddot{\Phi}_V - \ddot{\Phi}_E) - W \qquad (10b)$$

Since in this case it is not the absolute angle of rotation but rather only the deviations from the basic rotation (i.e., the rotation of the input shaft) which are significant, these two equations can combined. If the driving speed v for the short time segments to be considered is taken as constant, this produces with

$$\Phi_E = \Phi_{Eo} + \varphi_E,\ \Phi_V = \Phi_{Vo} + \varphi_F,$$
$$\dot{\Phi}_{Eo} = \dot{\Phi}_{Vo} = v = \text{const.}$$

and

$$\psi = \varphi_V - \varphi_E,$$

the motion equation

$$\ddot{\psi} + c \cdot \left(\frac{1}{J_E} + \frac{1}{J_V}\right) \cdot \psi = -M_E \qquad (11)$$

The natural frequency ω_0 for this system as follows

$$\omega_0 = \sqrt{c \cdot \left(\frac{1}{J_E} + \frac{1}{J_V}\right)}$$

The general solution produces the acceleration acting on the driver:

$$\psi = A \cdot (\sin \omega_0 \cdot t) + B \cdot \cos(\omega_0 \cdot t) \qquad (12)$$

(9) produces as the end condition of the engine run-up phase:

$$\ddot{\psi}_E = \frac{M_E}{J_E} \text{ for } t < t_0 \qquad (13)$$

The energy to be applied here simply to accelerate the engine is transformed when the one-way clutch engages (at time t_0) abruptly into a torque, which causes the input shaft to rotate:

$$\varphi_{Eo} = \frac{M}{c}$$

13 Diagram of drivetrain during load interruption and establishment of frictional connection

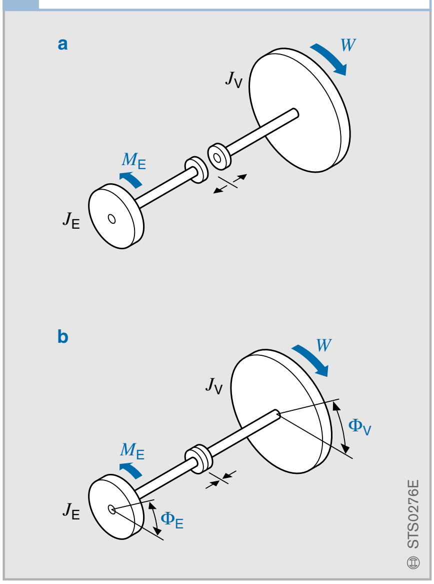


Fig. 13
a Load interruption
b Frictional connection

J_V Mass moment of inertia of vehicle drivetrain
J_E Mass moment of inertia of engine
M_E Engine torque
Φ_V Angle of rotation of vehicle drivetrain
Φ_E Angle of rotation of engine
W Running resistance

(12) produces the relative acceleration

$$\psi = \omega_0^2 \cdot \frac{M}{c} \cos(\omega_0 \cdot t)$$

and (10b) the vehicle acceleration

$$\ddot{\Phi}_V = \frac{M}{J_V} \cdot [1 + \cos(\omega_0 \cdot t)] \qquad (14)$$

This means that, at time t_0 when the one-way clutch engages, an acceleration jump takes place, namely

$$\text{from } \ddot{\Phi}_V = 0 \text{ for } t < t_0 \qquad (15)$$

$$\text{to } \ddot{\Phi}_V = 2 \cdot \frac{M}{J_V} \text{ for } t = t_0$$

followed by a drivetrain vibration damped in the real vehicle.

Similar conditions are present in a gearshift from friction element to friction element (overlapping gearshift), only this involves the additional problem of cutting in the friction element of the new gear exactly when the synchronization speed is reached. As the damping effects of the torque converter and the rest of the drivetrain have been disregarded in this analysis, the possibility which engine intervention offers is all the more clear:

According to (13), the initial acceleration acting on the driver at time t_0 is directly proportional to the engine torque and thus the engine acceleration during the run-up phase. With precisely timed control of the engine torque in the time segment $t \leq t_0$ to $t \gg t_0$, it is possible to create an almost continuous transition from the range of tractive-force interruption to the range of tractive-force transfer.

Implementation takes the form of a marked reduction in engine torque at time t_0 followed by renewed control-up in accordance with a time function. The convenience can be varied within broad limits with this control-up.

There is an equally clear possibility in gearshifts without a one-way clutch of influencing the engine acceleration by controlling the engine torque in the time range $t \leq t_0$ and thereby reducing the time demands on the cut-in precision of the friction element at the synchronization point.

Sequence Control

The process of reducing the engine torque is essentially very simple. However, effective control requires precise coordination as the entire process only lasts approximately 500 ms.

Pure time control of engine intervention is not practicable because different variables which determine the sequence (such as clutch fill times, plate friction coefficients, and similar) fluctuate within broad limits depending on the temperature and the service life.

As engine intervention is directly linked to the shifting sequence, a speed sequence control system suggests itself. The characteristic variable that characterizes the shifting sequence exactly is the transmission input sequence. The engine speed is also suited with limitations to transmissions with hydrodynamic converters as the controlled variable. This is therefore important because it requires a separate sensor to record the transmission input speed, and not every transmission is fitted with this sensor for cost reasons. In the interests of clarity, the following text describes the control system with the transmission input speed as the characteristic variable and reference is made, where necessary, to the limitations or changes when the engine speed is used.

Upshifts

The time curve of the characteristic variables for an upshift is depicted in Figure 14.

The ratio of the old gear is retained up to the overrunning point t_2; only then does only a slipping clutch intervene. For this reason, the engine torque cannot be reduced before the overrunning point is reached, otherwise this would entail an intensified dip in the output torque in the phase $t_1 ... t_2$.

The overrunning point is identified through continual monitoring of the transmission input speed in the time phase after t_0. For this purpose, the maximum speed is calculated in the time phase $t_0 ... t_3$ as well as the speed gradient.

The overrunning point is identified in the event of a reduction in the gradient by more than a pre-specified threshold value, and engine-torque control begins with control-down to a pre-specified value in accordance with a pre-specified time function.

In order to determine the speed n_3 at the start of control-up, the synchronization speed $n_4 = n_1 / i$ in the new gear is calculated from the maximum speed n_1 at the overrunning point and the ratio jump i of the gear change to be carried out. A speed-dependent share Δn is added to this synchronization speed in order to obtain a derivative action for control-up. When the speed $n_3 = n_4 + \Delta n$ is reached, torque control-up begins in accordance with a pre-specified time function. The end of the gearshift is identified as soon as the value of the non-corrected torque is achieved.

14 Sequence control, upshifts with engine intervention

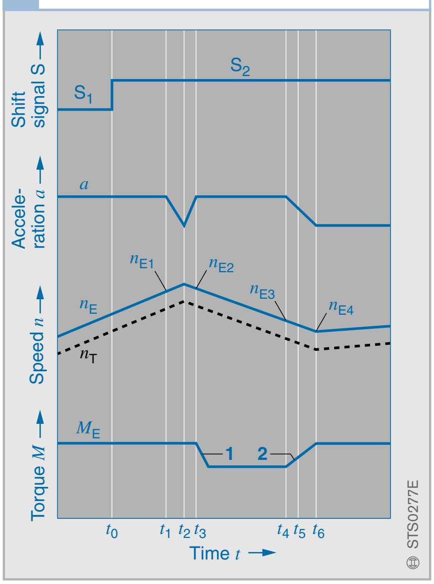


Fig. 14
1 Control-down phase
2 Control-up phase

a Acceleration
n_T Transmission input speed
n_E Engine speed
M_E Engine torque
S Shift signal

For upshifts in the upper load range (greater than half load), the engine speed is used as the controlled variable instead of the transmission input speed because here the shifting points are at such great engine speeds that the converter operates in the clutch range and is thus subject to roughly constant slip.

Gearshifts at part load on the other hand take place in the converter range. This means that the slip can change very substantially during a gearshift. Here the engine speed is no longer suitable for determining the synchronization speed. In this case, a superimposed form of time control, which terminates engine intervention after a prespecified time, is suitable for the part-load range $t_3 \dots t_4$.

Downshifts
The time curve of the characteristic variables for a downshift is depicted in Figure 15. Precise determination and recording of the synchronization speed are crucial to engine intervention in the case of downshifts because

- retarding the ignition angle too early extends the engine revving phase and thus the time of tractive-force interruption and
- engine intervention after the one-way clutch has engaged does not bring about any improvement in convenience, but rather a deterioration as this causes a dip in torque for the duration of the engine intervention.

The synchronization speed is calculated via the gear step from the transmission input speed at the start of the gearshift. The engine torque is abruptly reduced approximately 200 rpm before the synchronization speed is reached until this speed is reached or slightly exceeded. Then the engine torque is slowly controlled up again.

The synchronization speed cannot be calculated directly by means of the engine speed as a result of the slip at the hydrodynamic torque converter. A consideration of the converter program map with the requisite accuracy requires too much computation effort in the microcontroller.

It is possible however to calculate the synchronization speed from the transmission output speed by multiplying it by the corresponding gear step. It is now possible to identify by means of the engine speed when the synchronization point is reached, since the speed difference between the engine and the turbine is approximately zero when the engine revs up freely (interruption of tractive force) to the synchronization point.

The different possibilities of torque reduction are now discussed in the following text.

15 Sequence control, downshifts with engine intervention

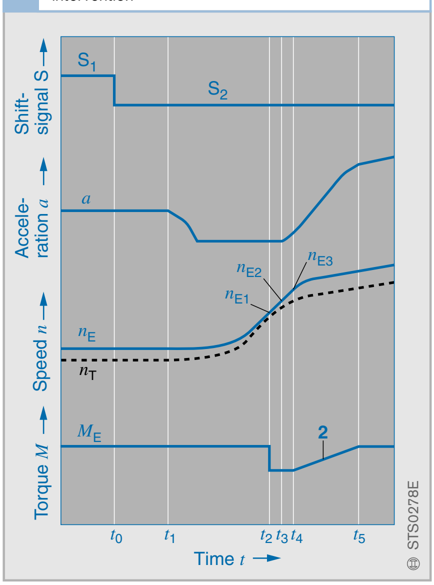


Fig. 15
2 Control-up phase
a Acceleration
n_T Transmission input speed
n_E Engine speed
M_E Engine torque
S Shift signal

Shifting the Ignition Angle

The oldest version of engine intervention involves intervention through shifting of the ignition angle. This form of intervention offers the following advantages:

- continuous regulation of the engine torque within broad limits,
- short response time, and
- availability in all vehicles with gasoline-engines.

Figure 16 shows in schematic form the dependence of the engine torque on the ignition angle for different load conditions and engine speeds. It is clear from this figure that adjusting a pre-specified engine torque generally requires an ignition map as a function of engine load and engine speed.

The response time τ between the initiation of engine intervention and the start of reduction in engine torque is specified by the ignition angle, therefore

$$\tau \approx \cdot \frac{1}{(z/2) \cdot n_E}$$

where z is the number of cylinders and n_E the engine speed. In the effective speed range $n \geq 2000$ rpm, the maximum delay for a 6-cylinder engine is 10 ms for initiation and 30 ms for a complete reduction in engine torque.

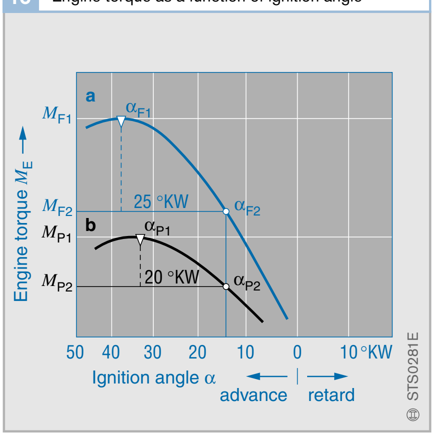
Engine-Torque Specification

In appropriately equipped vehicles with their CAN network of all the ECUs in the drivetrain (Figure 17), torque reduction is performed on the basis of a torque interface between engine management (TI-Motronic) and electronic transmission control (ETC). The torque reductions of the ABS and TCS ECUs must also be taken into consideration.

Figure 18 shows how a current transmission-control system calculates the desired engine torque intervention (ETI_Etc).

The next torque intervention is determined as a function of the available torque (actual torque). The torque M is the engine torque of the engine-management system without intervention by transmission control.

16 Engine torque as a function of ignition angle



17 ECUs in the drivetrain

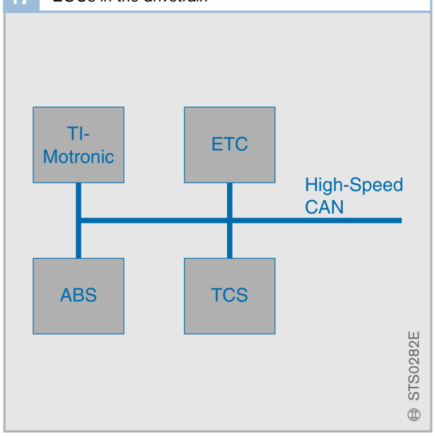


Fig. 16

a Full load (index F)

α_{F1}, M_{F1} Ignition angle or engine torque without engine intervention

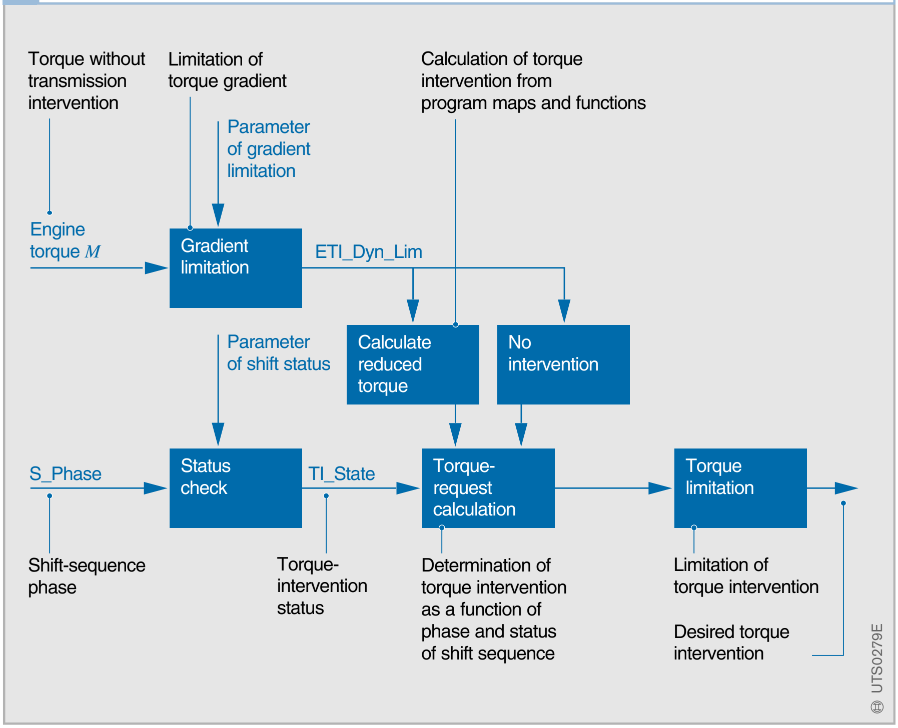
α_{F2}, M_{F2} Reduced ignition angle or engine torque with engine intervention

b Part load (index P)

α_{P1}, M_{P1} Ignition angle or engine torque without engine intervention

α_{P2}, M_{P2} Reduced ignition angle or engine torque with engine intervention

18 Calculation of engine-torque intervention (ETI) in current transmission control



Key to Figure 18:
ETI_Etc = f(ETI_Dyn_Lim, TI_State)

where

ETI_ETC:	Engine-torque request of transmission control
ETI_Dyn_Lim:	Dynamically limited engine torque (reduction of torque change in the engine to ensure convenience)
TI_State:	Current status of torque intervention

Torque Converter Lockup Clutch

Application and Operating Concept

The hydrodynamic converter is (necessitated by its operating principle) subject to a level of slip which is required particularly for convenience reasons during startup and in certain driving situations to increase torque. Since this slip also involves a simultaneous loss of power, the torque converter lockup clutch (TCLC) was developed with this in mind (see also section entitled "Torque Converter").

It is only advisable to lock up the converter from a specific speed because at low speeds the irregular rotation of the engine would cause uncomfortable vibrations in the drivetrain. The controlled torque converter lockup clutch (CTCC) was developed in order to be able to utilize these ranges for lockup as well.

Controlled Torque Converter Lockup Clutch

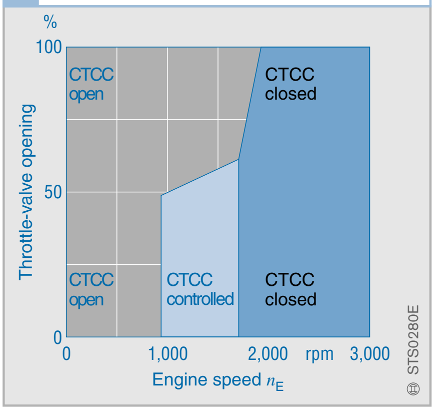
The controlled torque converter lockup clutch (CTCC) sets a very low level of slip (40...50 rpm) and thus an almost stationary state. In this way, it keeps unwanted vibrations away from the drivetrain. The converter lockup clutch therefore has three different states:

- open,
- controlled, and
- closed.

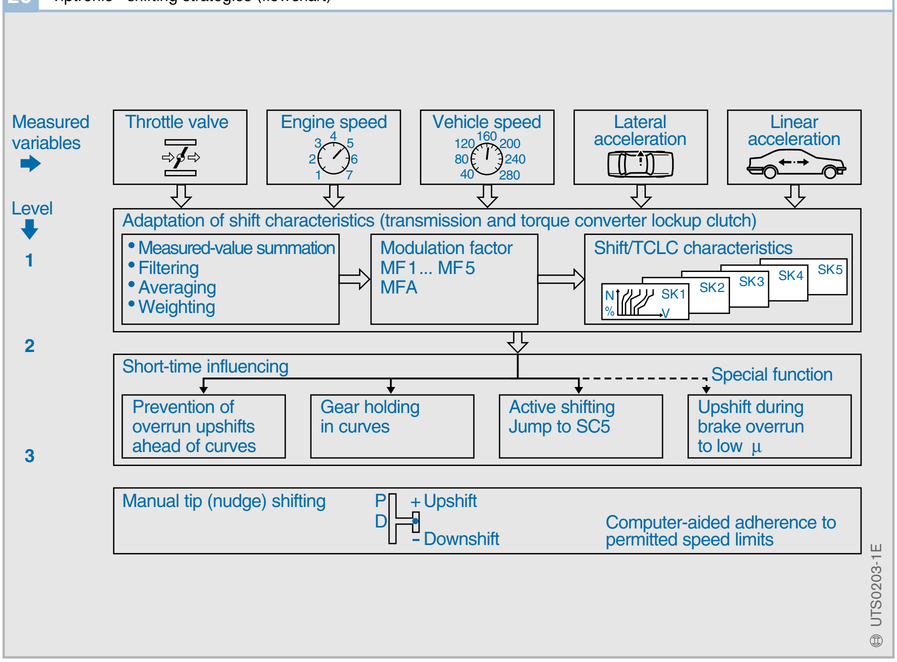
These states are defined by means of characteristic curves, which are plotted like shift curves for each gear against throttle-valve opening and driving speed (Figure 19). As is the case with shift curves, fuel consumption and tractive force are crucial criteria for the torque converter lockup clutch.

In slipping controlled operation, the speed differential between the converter impeller and turbine is constantly set to a low value. A closed control loop constantly compares the speed differential with a pre-specified setpoint value and corrects the pressure continually. Special functions perform changeovers between the individual states and provide for a comfortable shift performance.

19 Characteristic (pictorial diagram) of controlled torque converter lockup clutch (CTCC)



20 Tiptronic® shifting strategies (flowchart)



Control of Continuously Variable Transmission

Requirements

Continuously variable transmissions that operate according to the wrap principle have a whole variety of different equipment specifications (Table 1). The following equipment packages are widely used in compact and mid-range class vehicles:

- When the master/slave concept is used, the primary pulley (transmission input side) has double the surface of the secondary pulley (transmission output side). The pressure in the primary chamber can thus always be below the secondary pressure.
- The converter with torque converter lockup clutch as power take-up element offers very good starting convenience and facilitates a good starting response through torque increase so that the large ratio span of the CVT is completely beneficial to the overdrive range.
- Two wet clutches for the forward and reverse gears.
- Variable-capacity pump.
- Convenient fail-safe and limp-home strategies.

In the event of a control-electronics failure, the fail-safe and limp-home requirements partially determine the hydraulic concept. Engine overspeeding and associated high slip at the driven wheels must be avoided at all costs. An adjustment in the overdrive direction would satisfy this requirement, but starting from a standing stop would no longer be possible.

Open and Closed-Loop Control Functions

The aforementioned transmission equipment requires the following open and closed-loop control functions:

- contact-pressure control,
- ratio control,
- driving program,
- clutch activation,
- activation for converter and torque converter lockup clutch,
- pump activation,
- reverse-gear lock, and
- deactivation of limp-home function.

Contact-Pressure Control

The belt contact pressure is adjusted in accordance with the current load situation with the aid of the measured secondary pressure. To achieve a high level of efficiency, the secondary pressure is reduced to such an extent that the current engine torque can still be transferred to a specific degree of safety and reliability without the belt slipping.

Table 1

1 Variations of CVT based on wrap principle

Assembly, function	Variants		
Wrap element	Band	Chain	Belt
Variator principle	Master/slave	Partner principle	Partner principle
Converter:	Fitted	Not fitted	Not fitted
– Converter clutch	Yes	No	No
– Slip duration	Brief	Permanent	Permanent
Clutch:			
– Type	Friction surfaces	Magnetic-powder	Magnetic-powder
– Pressures	Low	High	High
– Slip duration	Brief	Permanent	Permanent
Pump adjustment	Constant	2-stage	Continuous
Limp home	Not possible	Limited (loss of convenience)	Unlimited (increased fuel consumption)
Vehicle:			
– Class	Small to mid-range class	Compact class	Mid-range and Luxury class
– Engine size	<3 *l*	<2 *l*	>2 *l*
– Type of drive	Front transversal	Front longitudinal	Rear

Ratio Control

The gear ratio can be changed by means of the primary pulley. The enclosed fluid volume determines the axial position of the moving part of the primary pulley and thus the radius on which the belt circulates on the pulley. The primary pressure adjusts itself in response to the secondary pressure.

The requirements of driveability determine the necessary adjustment speed. For example, in the case of kickdown, it is necessary to switch from overdrive to low within 1.5 s. On the other hand, the pump delivery limits the adjustment speed.

Driving Program

A driving program ascertains the desired gear ratio. In addition to different program maps for normal operation, in which there is the option of choosing between economical and sporty operation (see also section entitled "Adaptive Transmission Control, ATC"), it is also possible to implement special functions such as kickdown, downhill driving, etc.

It is also possible to simulate range transmissions, where any intermediate variations between copying a manually shifted transmission and an automatic transmission are realizable (see also chapter entitled "Transmission for Motor Vehicles").

Clutch Activation

The interrupting clutch between engine and drivetrain is designed as a function of the position of the selector lever (P-R-N-D), the engine speed, and the engine load.

Activation of Converter and Torque Converter Lockup Clutch

In order to achieve the greatest possible efficiency, it is essential for the converter to be locked up as early as possible. Depending on the power requirement, the torque increase is used up to different speeds for acceleration.

Pump Activation

A variable-capacity pump must be used to ensure high transmission efficiency. This pump enables the delivery flow to be limited at high speeds.

Suitable suction-throttled pumps which operate without additional activation have been in development for years, but have failed to make a breakthrough as yet. An initial step towards a "variable-capacity pump" has been taken in the form of a two-stage version, in which the more favorable delivery flow can be selected as a function of the current demand.

Further concepts are feasible with continuously variable pumps, in which secondary-pressure control and pump adjustment can be combined.

Reverse-Gear Lock

Engagement of the reverse gear is disabled during forward driving at speeds above a limit to be defined (e.g., 7 km/h).

Deactivation of Limp-Home Function

Limp-home is an emergency function which is shut down during normal control operation.

The fail-safe function remains permanently activated so that engine overrevving is avoided even in the event of a partial failure or late identification of partial failures.

ECUs for Electronic Transmission Control

Application

In implementing electronic transmission control, engineers can choose to locate the ECUs in different positions in the vehicle. There are, for example, separate, combined, mounted, or integrated ECUs (Figure 1).

The way in which these ECUs are distributed in the vehicle is essentially determined by

- the ratio of vehicles with automatic transmissions to vehicles with manually shifted transmissions and
- the demands made by the transmission on the control system (performance of the microcontroller used).

In Europe, the market is still dictated by the separate printed-circuit-board ECUs ME and TC (Figure 1a). In the USA, on the other hand, it is predominantly the combined drivetrain ECU (MEG) that is used for 4-speed and 5-speed automatic transmissions (Figure 1b), the reason for this being that automatic transmissions dominate the market in America with a market share of over 85 %.

The newer 6-speed or 7-speed transmission types and the increasing demands on engine-management systems (emission-control legislation, CARB requirements) have now started a trend in the USA moving away from combined ECUs in favor of separate ECUs. This trend is consolidated still further by the latest generation of 6-speed transmissions. These transmissions are already equipped with electronic modules with integrated electronic circuitry.

1 Possible distribution of ECUs in the drivetrain

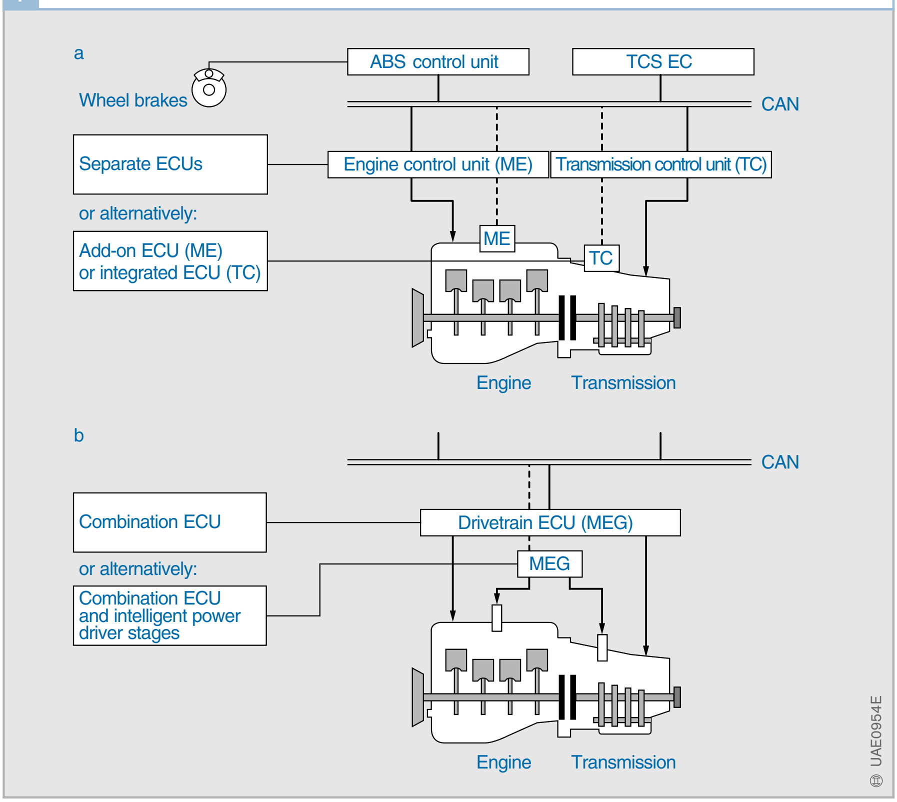


Fig. 1
a Layout with separate ME and TC printed circuit board ECUs
b Layout with combined drivetrain ECU

Design and Operating Concept

The various ECUs and their technical and functional details will be discussed below.

Printed Circuit Board ECUs

The most widely used ECUs currently are printed circuit board (PCB) units.

Figure 2 shows an ECU with a 32-bit microcontroller (Motorola 683xx) for a ZF 5-speed transmission. This transmission has been in mass production at BMW for some years now. The figure depicts the essential layout and the data flow of the ECU in a block diagram. The ECU itself can be roughly divided into three sections:

1. Input side
The input side comprises the power supply (terminals 15 and 30), signal acquisition, and the communication interface.

The input signals include the signals for engine speed, turbine speed, output speed, and wheel speeds. The transmission control unit usually receives the engine-speed and wheel-speed signals via the CAN interface from the acquiring ECUs (engine and ABS control units).

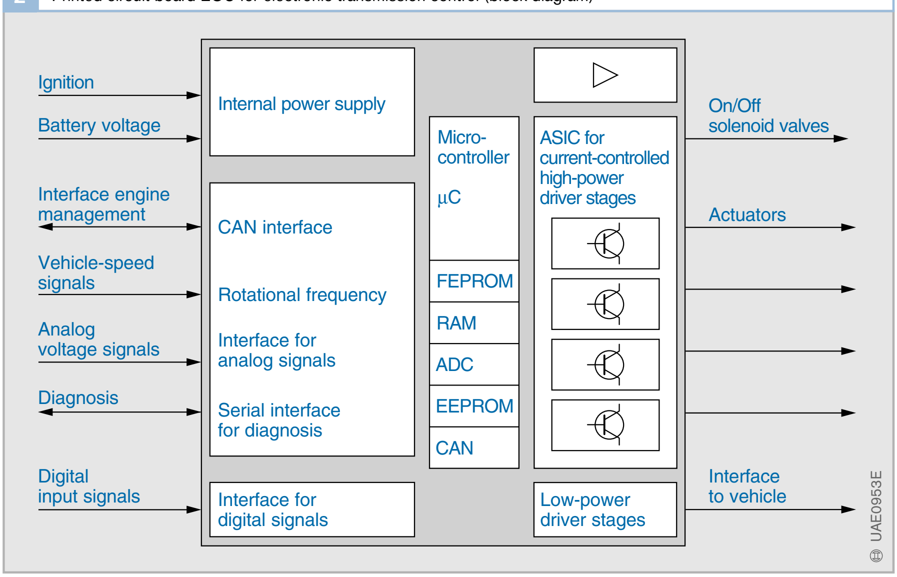
The ECU acquires the transmission-fluid temperature as an analog input signal because the properties of the fluid have a significant effect on shift quality, especially when the engine is cold. The ECU acquires the position of the selector lever in the form of a digital signal. The following information can also be obtained and evaluated via the CAN interface:

- accelerator-pedal position (driver command),
- kickdown switch,
- engine temperature, and
- engine torque.

2. Computer Core
The computer core comprises microcontroller, flash, RAM, EEPROM, analog-digital converter, and CAN bus system.

3. Output Side
The output side features the driver stages for the on/off valve, ASICs, current control (CG205), and low-level signal driver stages.

2 Printed circuit board ECU for electronic transmission control (block diagram)

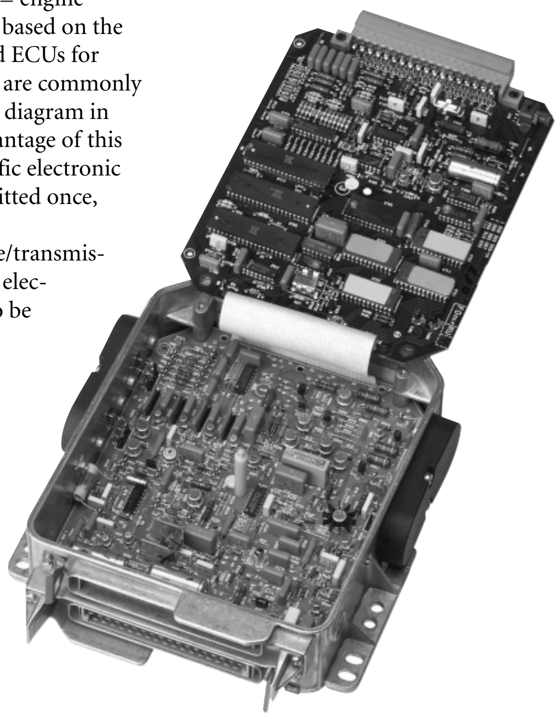


Drivetrain ECUs

MEG drivetrain ECUs (MEG = engine ETC/EGAS transmission) are based on the standard printed circuit board ECUs for engine and transmission, and are commonly used in the USA. As the block diagram in Figure 3 shows, the main advantage of this ECU lies in the fact that specific electronic components only have to be fitted once, thereby reducing costs.

An MG ECU (MG = engine/transmission) was the first example of electronic transmission control to be mass-produced. This ECU was developed back in 1983 for BMW for use with a ZF 4HP22 automatic transmission (Figure 4).

4 BMW drivetrain ECU from 1983



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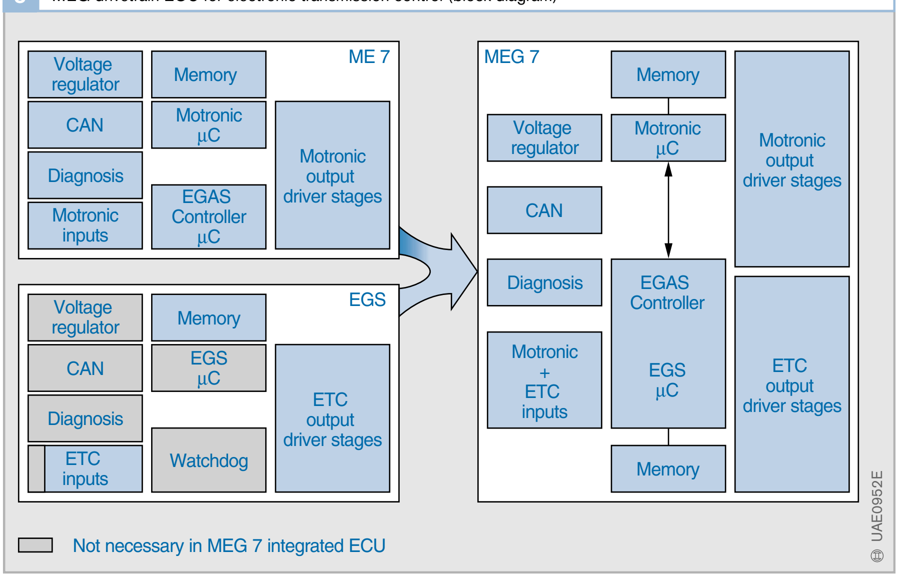
3 MEG drivetrain ECU for electronic transmission control (block diagram)



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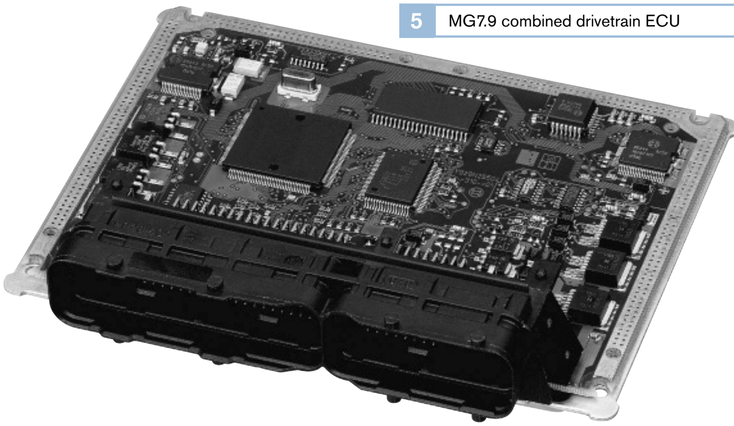


5 MG7.9 combined drivetrain ECU

Figure 5 shows the current configuration of a drivetrain ECU for the block diagram in Figure 3.

Since then, the demands on ECU computing power and memory capacity have changed dramatically (see Figure 6 and Table 1).

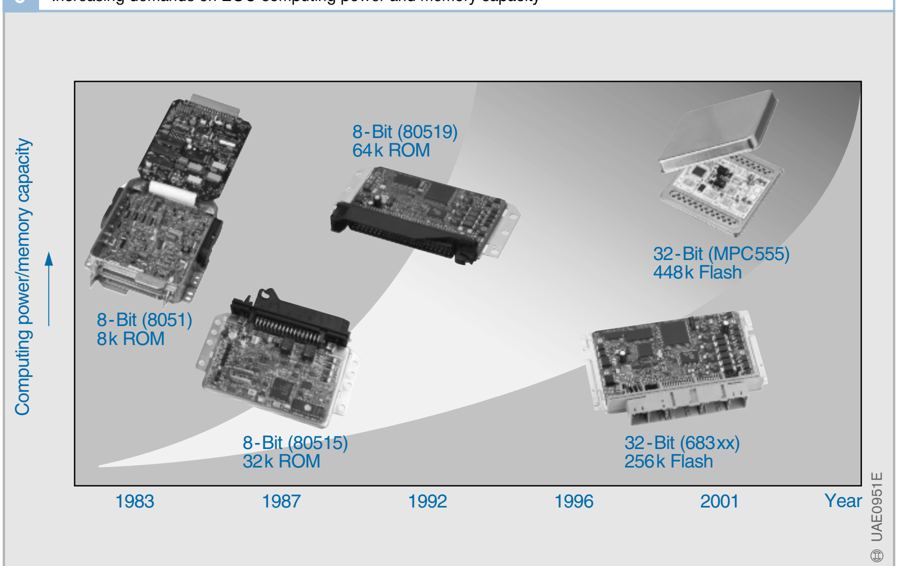
As the figures in Table 1 demonstrate, these demands are constantly rising and there is no end in sight to their development.

1 Development of computer performance

Year	Computer	Memory	RAM
Year	Computer	Memory	RAM
1983	Cosmac	8 k ROM	128 byte
1988	80515	32 k ROM	256 byte
1992	80517	64 k ROM	256 byte
1996	80509	128 k flash	2 k
199x	C167	256 k flash	4 k
1996	683xx	256 k flash	8 k
2001	MPC555	448 k flash	28 k
2003	MPC555	1 MB flash	28 k
2005	?	1.5 MB flash	66 k

Table 1

6 Increasing demands on ECU computing power and memory capacity

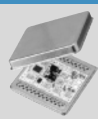


Microhybrid ECUs

The introduction of new transmissions (such as the ZF 6HP26) has seen a transformation in the type of ECU from the PCB to the microhybrid. This development has been influenced by the changing demands, mainly because of the environmental conditions under which the ECU is used (Table 2).

The microhybrid ECU basically contains the same circuitry as the PCB, but now with unencapsulated semiconductor components being used, i.e. as "bare" silicon chips. Electrical contacting is provided by wire bonding (in the PCB ECU with soldering).

2 Technical limits for PCB and microhybrid

Type	PCB	Microhybrid
		
Installation location	Interior or engine compartment	In transmission
Temperature	−40...+85/+105°C	−40...+140°C
Vibration	...5 *g*	...30 *g*
Protection	IP 40 / IP 69	IP6K9K in ATF

Passive components are electrically contacted by means of conductive bonding agents.

In contrast to the currently mass-produced circuits with LTCC (low-temperature cofired ceramics), finer layout structures are used in the new systems with 32-bit processors. This relates in particular to the via density and the bondland size.

The previous bondland grid of 450 µm would require four bond rows and at least three wiring layers to route the computer core. With the via grid of 260 µm used, two bond rows are sufficient, and this even with a reduced space requirement and only two wiring layers.

Figure 7a shows the ABS computer bond zones (44 bonds) on an LTCC standard substrate compared with the 32-bit controller in the diesel-control system (240 bonds) on LTCC-fine-line in Figure 7b.

In addition, Figure 8 shows a comparison of the wiring density of the inner layers (Figure 8a) and the reverse side of the hybrid with the integrated resistors (Figure 8b).

7 Bondzone microcontroller LTCC compared with LTCC-fine-line

a

b

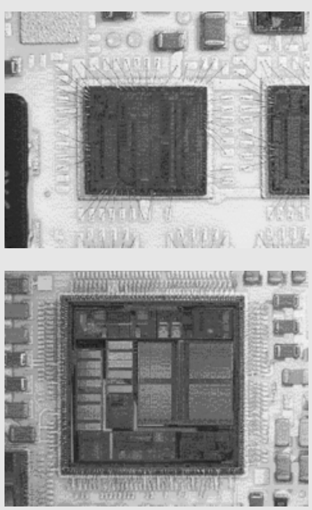


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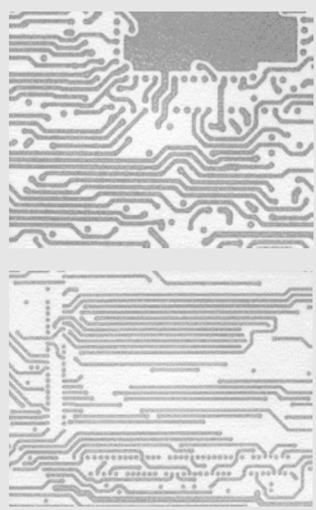
Fig. 7
a On LTCC standard substrate
b On LTCC-fine-line substrate

8 Hybrid wiring densities

a

b



UTS0319Y

Fig. 8
a Inner layers
b Reverse side with resistors

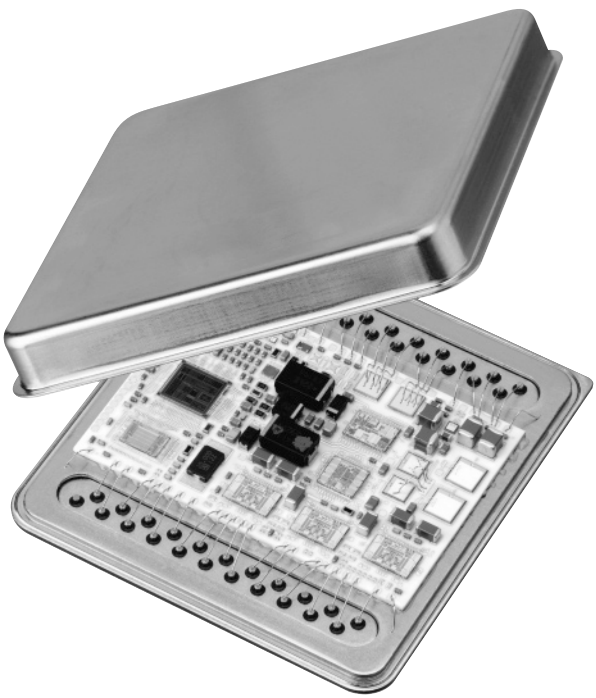
The following significant measures were taken to bring about an improvement in the process for microhybrid ECUs:
- use of finer punch needles,
- finer screens,
- adaptation of the pastes used, and
- tolerance optimization through adapted process management.

This compression of the layout makes it possible to produce the circuit for transmission control on an area measuring 2 x 1.2≤. In other words, a substrate with the working format of 8 x 6≤ alone can process 20 circuits in parallel.

For optimum cooling of ICs subject to high heat loss, thermal vias with a diameter of 300 µm are filled in parallel to the function vias. This increases the thermal conductivity of the substrate from approx. 3 W/mK to effectively 20 W/mK.

Figure 9 shows the complete microhybrid in its housing. The following procedures are used for the assembly process:
- All the components are bonded with conductive bonding agent.
- Bonding is carried out using a 32 µm gold wire and a 200 µm aluminum wire.
- The hybrid is bonded to the steel plate with heat-conducting bonding agent.
- The connection to the glass bushing is made by a 200 µm aluminum bond.
- The housing is hermetically sealed tight.

9 Microhybrid ECU in steel housing



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ASIC Chips

In addition to computer and memory chips, application-specific integrated circuits make up a significant proportion of the electronic components in the ECUs.

Different functions have been combined into ASIC chips for the purpose of reducing costs and standardizing the electronic design of transmission-control systems. These ASICs are available in encapsulated and unencapsulated form and used for both microhybrid and PCB ECUs. Transmission control features various ASICs which are currently in volume production; the three ASICs used in the microhybrid ECU are discussed in the following.

Current-Regulator ASIC CG205

The current-regulator ASIC CG205 with integrated shunt was developed for high-precision pressure control in the transmission. It achieves a control precision of 1% over the entire temperature range.

10 ASIC chips

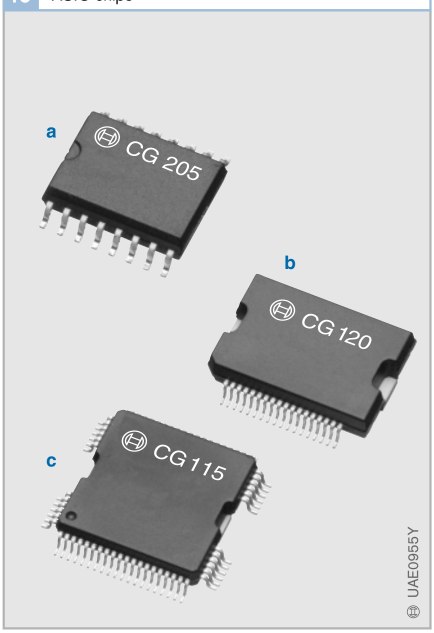


Fig. 10
a Current-regulator ASIC CG205
b Watchdog ASIC CG120
c I/O ASIC CG115

Figure 10a shows the encapsulated ASIC, as is used in PCB ECUs.

There is also the option of adjusting the current range and the PWM output frequency with the aid of an external circuit.

Watchdog ASIC CG 120

Due to the fact that the 6-speed transmission with integrated electronic module no longer has a mechanical connection between the selector lever and the transmission, the control system requires specific safety mechanisms. This function is performed by the ASIC CG120 (Figure 10b), which monitors the function of the microcontroller (see also the chapter entitled "Diagnostic functions"). The ASIC CG120 performs the following functions:

- power supply with 5 or 3.3 V,
- sensor supply,
- watchdog,
- serial interface,
- CAN interface,
- ISO 9141 interface, and
- programmable via SPI interface.

I/O ASIC CG115

In order to achieve the high level of integration in the microhybrid, it is necessary to combine as many functions as possible in one ASIC. Achieving this with individual components would therefore take up too much space on the substrate.

The following functions are integrated in the I/O ASIC CG115 (Figure 10c):

- voltage monitoring,
- inputs and outputs for digital signal transmission,
- 2 inputs for inductive Hall-sensor signals,
- 8-channel analog multiplexer,
- serial interface, and
- programmable via SPI interface.

Thermo-Management

Dissipating the heat losses generated in the ECUs represents a prime consideration in the design of mechatronic modules, particularly in those cases where "hotspots" cause a highly uneven distribution of the heat losses. Figure 1 shows the heat-dissipation model in an ECU to the point of a heat sink which is situated in the valve housing in this example. This is an LTCC microhybrid in a welded steel housing which is mounted on the aluminum housing of the hydraulic main control stage.

Effective heat management of the IC necessitates a close contact between the chips and the housing. Materials with high thermal conductivity are suitable for use in this respect.

As other tests on the various substrates for high-temperature applications have shown, these materials demonstrate very different thermal-conductivity properties. LTCC glass ceramic shows itself to be less effective than aluminum-oxide ceramic (Al_2O_3) by a factor of almost 10. However, this drawback is compensated by thermal vias in the microhybrid to such an extent that LTCC technology demonstrates equally good levels of thermal conductivity as aluminum-oxide technology.

Figure 2 shows an area of thermal vias. The production process creates these heat-dissipating thermal vias (heat spreaders) in parallel to the electrical connections.

Essentially, the terms "power loss", "depletion-layer/junction temperature", and "heat dissipation" define the limits for a microhybrid ECU.

The power loss P_l can be described in simplified form for stationary operation as follows:

1 Heat-dissipation model in the housing of a microhybrid ECU

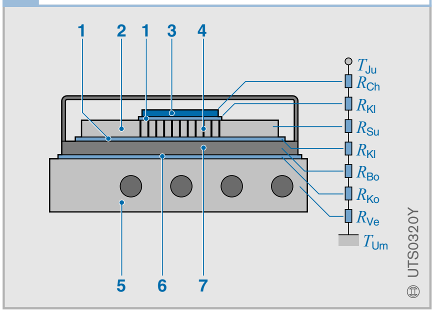
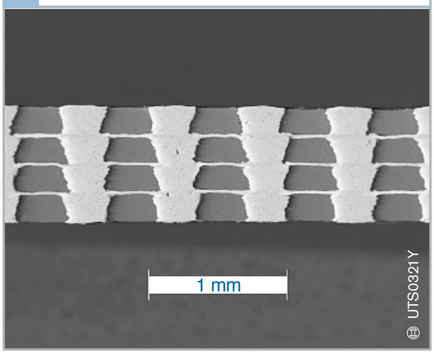


Fig. 1
1 Bonding agent
2 Substrate
3 Si chip
4 Thermal vias
5 Al valve housing
6 ATF
7 Steel housing base

2 Cross-section through a 4-layer microhybrid with thermal vias



$$P_l = (T_j - T_a) / R_{jth}$$

where

T_j Junction temperature
R_{jth} Thermal internal resistance
T_a Ambient temperature

The thermal resistance R_{jth} (auxiliary quantity) is dependent on the geometric parameters and the specific thermal conductivity of the material and is determined from measurements. The maximum permitted junction temperature T_j determines the maximum permitted power loss P_{lmax}, where T_j is dependent on the material (for silicon T_{jmax} = 150...200°C). Current specifications laid down by microcontroller manufacturers establish an upper limit of T_{jmax} = 150°C. Since T_j is also dependent on the design, a

3 Comparison of thermal resistance of aluminum-oxide and glass ceramics (LTCC)

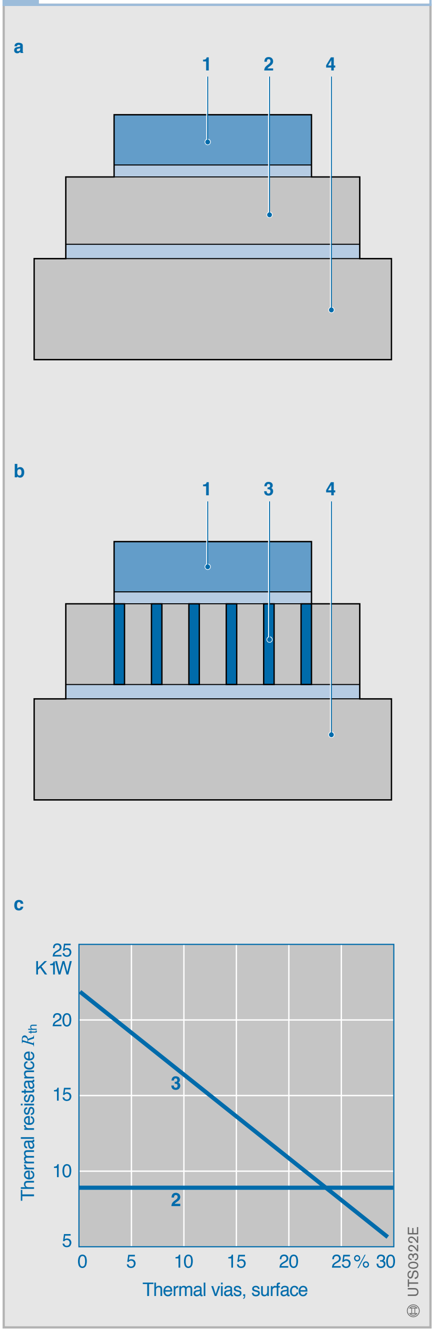


Fig. 3
a Aluminum-oxide ceramic
R_{th} = 12 K/W (CS 200 on aluminum)
b LTCC
R_{th} = 10...11 K/W (CS 200 in ABS 5.3)
c Comparison

1 Chip
2 Al_2O_3
3 LTCC (thermal vias)
4 Aluminum

consideration of design rules is important for optimization of the junction temperature, e.g.:

- Circuit sections subjected to thermal load should not be designed in accordance with the otherwise applicable minimum criteria, rather affected driver-stage areas (transistors/pn junctions), for example, should be geometrically enlarged.
- "Hotspots" should not be positioned in the corners of the ICs, this enables the IC substrate material to act in all directions as a heat spreader.

The design of the ASICs for transmission control described in the "ASIC Chips" section is suitable for maximum junction temperatures of T_{jmax} = 175°C.

In relation to the complete microhybrid system, designers are working towards optimizing the level of sophistication between increasing IC size, using thermal vias, and mounting on special substrates such as DBC or PCs.

Explanation of abbreviations:
DBC (**d**irect **b**onded **c**opper): copper-coated ceramic
and
PCs (**p**ower **c**hips): chips soldered onto copper plates.

Figure 3 shows a comparison of the thermal resistance of aluminum oxide (Figure 3a) and glass ceramic, i.e., LTCC (Figure 3b), and also specifically in chart form (Figure 3c).

Processes and Tools Used in ECU Development

Simulation Tools

Simulation of individual components and of the complete system is playing an increasingly important role in improving and stepping up the development process. The advantages of mathematical modeling over actual physical models (prototypes) are as follows:

- frequent reproducibility as desired,
- deeper understanding of system behavior,
- individual parameter analyses possible,
- lower costs,
- less time required for model modifications, and
- flexible application in all technical fields.

The following text deals briefly with some tools for simulation that are used.

Figure 1 shows by way of example the thermo-simulation of a microhybrid ECU.

It clearly shows the high temperature of the hotspots in the electronic circuitry which in this case occurs at a voltage regulator. This simulation is used to optimize the positioning of the thermal vias and the distribution of the components before the real ECU is built.

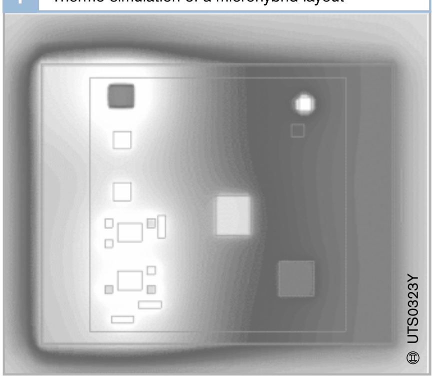
Circuit Simulation with SABER

Before a circuit is built with real components, it is possible to test its function using the SABER® simulation tool. Many component manufacturers already offer the data on their products in a SABER library for this purpose (Figure 2).

These data can be used to test the circuit with regard to its robustness, thermal properties, worst-case performance, and EMC behavior and thereby facilitate any necessary circuit optimization at a very early stage in its development.

This approach can reduce the number of redesigns.

1 Thermo-simulation of a microhybrid layout



2 Simulation with SABER