Steyr-Daimler-Puch Fahrzeugtechnik

FINAL REPORT

PROMETHEUS CED2 Demonstrator Project

of Steyr-Daimler-Puch Fahrzeugtechnik Ges.m.b.H.

System Introduction, Developement Status and Documentation on

Active Torque Split System

Driver Information System

Graz, March 29, 1993

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PROMETHEUS Demonstrator Project of Steyr-Daimler-Puch Fahrzeugtechnik GesmbH. Final Report

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Summary

The present report summarizes the final stage of the PROMETHEUS-CED 2 project "Proper Vehicle Operation" of Steyr-Daimler-Puch Fahrzeugtechnik (SFT), the major projects steps done, the system´s functional principles and some research results got. Some practical experiences are given as well as an conclusion on possibilities of serial realization.

Based on the activities carried out in the field of 4WD vehicle technology, three levels of project and research work were defined:

Level 1: Active torque split system for 4WD passenger vehicles in order to perform highest driving stabiltiy,

Level 2: Driver information system PDIS on current driving safety reserve,

Level 3: Integrated driver information and closed loop control system.
Note: Level 3 tasks have been decided to be carried out as an internal SFT continuation project.

During the project phase, SFT was in close cooperation with the Graz University of Technology. Other successful collaborations concerned the installation and test of the ISIS beacon system from Peugeot S.A. The contribution of the SEMPERIT company in the field of tyre technology was a very friendly one. Furthemore, Steyr-Daimler-Puch also held the leadership of the PROMETHEUS Working Group 2 "Actuators".

1. General Project Items

1.1 Time History of the Project

SFT joined the PROMETHEUS-CED2 group at November 1988. We periodically participated with the CED 2 meetings in order to coordinate their activities with the partners and to discuss the research experiences held. SFT hosted the CED2 meeting at Nomber 11-12, 1990 in Graz. The main work in 1991 was carried out for the preparation of the project presentation at the PROMETHEUS Board Member Meeting ´91 in La Mandria. The SFT-project was finished at the end of 1992.

During the project phases, SFT was in close cooperation with the Graz University of Technology. Other successful collaborations concerned the installation and test of the ISIS beacon system from Peugeot S.A. The contribution of the SEMPERIT company in the field of tyre technology was a very friendly one. Concerning the researches carried out in the field of friction detection and safety margin determination many good discussions and an extensive exchange of experiences took place, c.f. [3]. Finally, the resulting system features have been compared with those of the other PROMETHEUS partners, c.f. [4]. Furthemore, Steyr-Daimler-Puch also held the leadership of the PROMETHEUS Working Group 2 "Actuators".

As research prototypes, two 4WD vehicles based on type Lancia Delta Integrale were built up since they had already many important attributes to carry out the planned research and testing programme, Fig. 1.1 .

Fig. 1.1: PROMETHEUS research prototype

1.2 Project Goals

Addressing the "PROMETHEUS Functions" [1], the aims for the project have been defined as

F4 Monitoring Vehicle: reliable monitoring of the system specific safety related quantities for Level 1 and 2 systems,

F8 Dynamic Vehicle Control: autonomous vehicle dynamics control by torque split device (Level 1 system),

F6 Safety Margin Determination: on board safety margin computation in real time (Levels 2 system),

F9 Supportive Driver Information: driver information system on current safety reserve (Level 2 system).

It was clearly decided, that in principle the messages from the driver information system can always be overriden by the driver whereas the active torque split system is working autonomously.

2. State of the Art and Improvements

All wheel drives (AWD) in vehicles have been originally installed as rigid systems due to their advantages in mobility performance by improvement of traction. For advanced personal cars the traction aspect is only one of several criteria. Impact on driving behaviour and safety aspects such as braking stability and ABS-compatibility e.g. became more significant. Thus, various passive torque control systems have been developed to meet the requirements, such as viscous couplings and viscous-lock differential gears. In the area of 4WD vehicle technology, further improvements of vehicle dynamics behaviour and safety capabilities are available by continuous active torque split. In particular, these advanced systems enable to influence the lateral dynamics behaviour properly by means of the couplings between the longitudinal/ lateral tyre contact forces, c.f. [7]. For that, SFT has based its concept on the work carried out in the field of active torque split systems.

3. Active Torque Split System VISCOMATIC (Level 1 System)

With a few execptions, all manufacturers of modern AWD passenger cars on the market offer systems for a variable torque distribution. The main influencing variables are quantities such as acceleration, speed and varying friction values between tyre and road surface, the latter being the utmost significant one. Torque transfer is controlled by speed- or torque-sensing elements, such as viscous couplings (VCs) or Torsen differentials.

Some manufacturers have realized electronically controlled torque distribution systems, either multi-step or continuously controlled, by means of which it is not only possible to govern the torque transfer to the axles, but actually to influence the driving behavior due to the couplings between the combined longitudinal and lateral tyre forces.

The mechanical basis of the considered torque split system is the controlled viscous coupling VISCOMATIC, the characteristics of which are automatically adapted to the actual driving conditions of a vehicle. This chapter describes the design, arrangements, and control strategy of the Level 1 system "Active Torque Split" of the CED2-project of SFT.

3.1 Controlled Torque Transfer

Even passive viscous couplings (VCs) in the drive train of a four-wheel drive (4WD) vehicle are exellent elements for achieving safe driving behavior. Depending on the position of the directly driven axle, the slip (meaning axle-slip) principle with automatic torque distribution results in a constant under- or oversteering influence of the drive distribution.

The torque/speed characteristic of ordinary viscous coupling is a compromise between traction, driving behavior and comfort (drive train wind-up during parking maneuvering); to ensure braking stability, an additional device, such as a free wheeling mechanism or a multi-plate disconnection device, is used.

In order to cover the segment of the high-performance vehicles, SFT started the research project VISCOMATIC with studies on the parameter influence to the change of the coupling characteristics. Particularly, the evaluation of the safety margin by improvement of driving behaviour was an aim of the SFT CED2-project. At the same time, the necessary electronic system had to be developed, c.f. [8].

The charcteristic feature of the VISCOMATIC is the combination of the viscous unit with a planetary gear set (Fig. 3.1). The planet carrier is mainly used for the input of power, the ring gear for the output, the sun gear is supported by the viscous brake. The characteristics (torque versus speed) of this viscous brake can be continuously controlled. This is achieved by a hydraulically actuated piston that changes the gap between the plates and the internal volume. A simple disconnecting device can be arranged to allow towing or vehicle tests on roll benches.

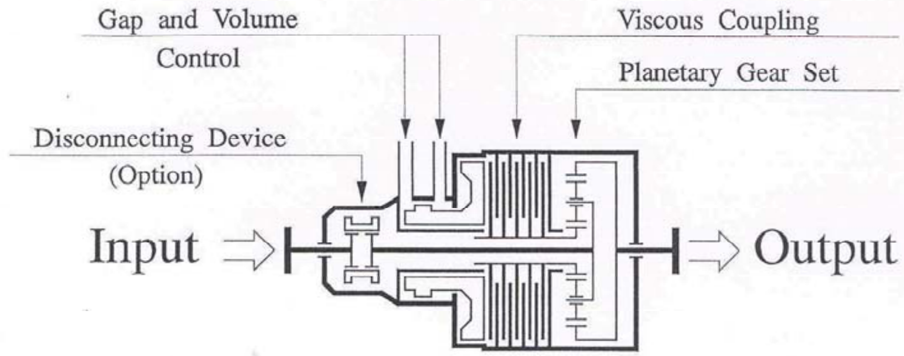

Fig. 3.1: Scheme of the VISCOMATIC

3.2 System Configuration

Since the proved principle of the slip-controlled torque distribution is used so far, of course, the first task is the acquisition and the processing of speed data.

Due to the position of its elements, the VISCOMATIC enables a good sensitivity regarding to the speed difference ("slip)" between the axles (Fig. 3.2). When the speed of the front anxle equals that of the rear axle the sun gear stands still, whereas a speed difference results in its rotation.

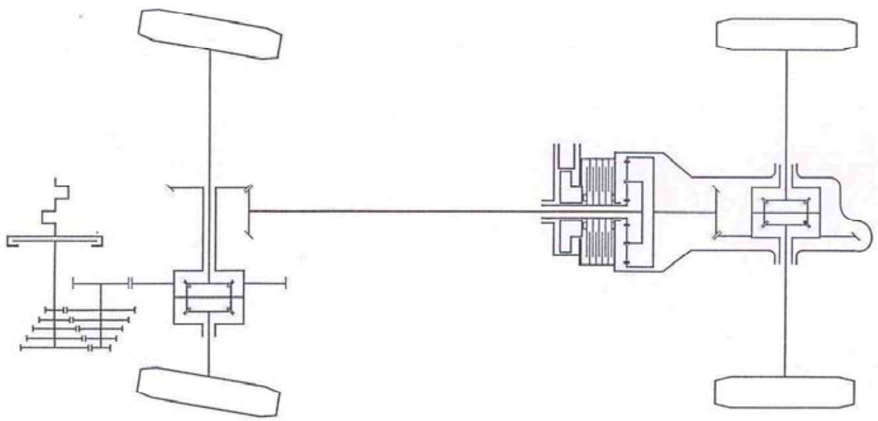

Fig. 3.2: Drive line of the SFT-PROMETHEUS-prototype

To save costs for the possible later serial production, data aquisition is done by means of the already existing ABS measurements (Fig. 3.3). The connection to the engine´s electronic system provides the engine´s speed and throttle position to determine the tractive force. The position angle of the steering wheel is recognized by a high resolution angular sensor. The zero position is identified via statistical methods by the software.

In order to recognize braking conditions or driving back maneuvers, the available switches of the vehicle are integrated into the system (brake switch, reverse gear switch).

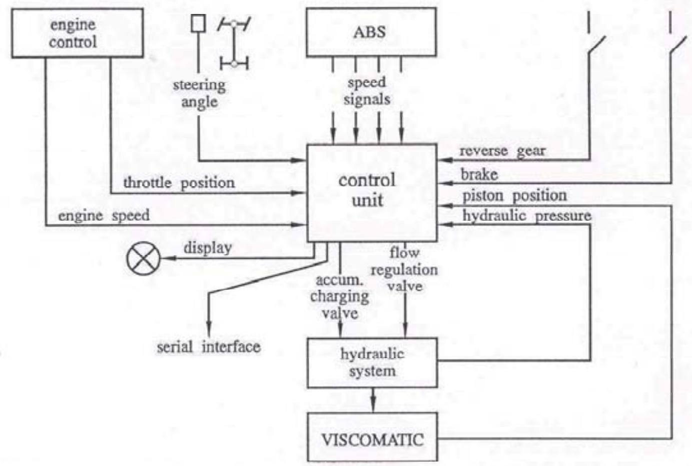

Fig. 3.3: Level 1 electronics (active torque split system)

The system is supplied with hydraulic pressure by the tandem pump of the steering servo and its level control. The pressure level in the pressure accumulator of the VISCOMATIC´s hydraulic system is monitored by a sensor; the algorithm guarantees the necessary level by actuating the accumulator-charging valve in priority over the vertical level adjustment.

A travel sensor is installed to recognize the piston position, such as it is used for electronically controlled Diesel injection systems to recognize the control rod position. The main output quantities of the control device are the control signal for the flow-regulation valve (motion of piston) and the accumulator charge valve. Since the electronic system is equipped with extensive software for defect identification and diagnosis, both a warning lamp and a diagnosis plug are installed.

3.3 Control Algorithm

The control strategy of the VISCOMATIC follows the principle of a slip control, i.e. the speed of both axles relative to each other (axle-slip). By controlling the axle-slip via the torque distribution the tyre circumference forces are influenced and such also the side force potential of the tyres and the steering tendency of the vehicle.

Fig. 3.4 shows the basic structure of the software. The total computing program is divided into individual software modules, each module carring out a defined task, e.g. calculation of the tractive forces.

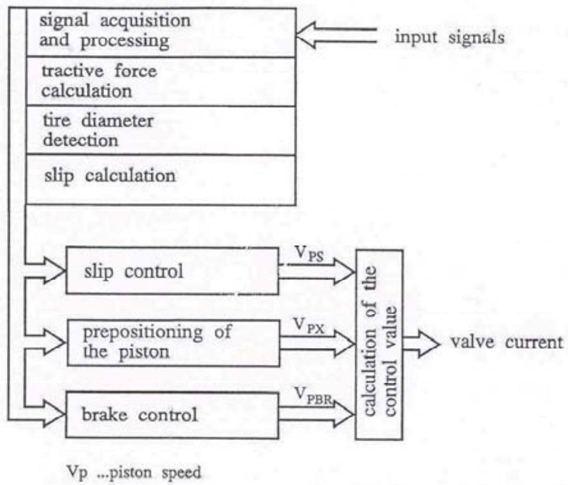

Fig. 3.4: Structure of control software

3.4 Tractive Force Calculation

The specified slip value is predetermined in consideration of the presently provided tractive force. The controller processes the throttle position and the engine speed, a stored map of characteristics provides the engine torque. The comparison of the speed of the engine and front axle provides the transmission ratio of the engaged gear, so that the total tractive force can be determined.

3.5 Identification of the Tyre Diameter

The dynamic rolling circumference of the tyres is influenced by different parameters, such as wheel load, air pressure, tread condition etc. The deviations at running can reach an extent

that is clearly within the range of the slip magnitude to be controlled, therefore they have to be taken into consideration.

Under suitable conditions - stationary driving condition, straight ahead driving, almost disconnected axles, low load - the mean slip between the axles is calculated over a short distance, and corrected by the difference value of the tyres that is already known. The wheel slip difference obtained, is identified as the change of the wheel difference value, and increased by a factor of importance, when it appears as a negative value in drive mode, and as a positive value in coast mode.

This method prevents the tyre difference value being obscured by the longitudinal slip, without being, however, limited to the rare condition of totally free rolling.

3.6 Slip Calculation

The slip S_{AX} between the axles is calculated from the wheel speeds, which are available from the ABS speed sensors:

$$S_{AX} = \frac{n_F - n_R}{n_R} * 100 \quad [\%] \qquad (3.1)$$

where n_F / n_R = front / rear axle speed - determined from the wheel speeds. This value must be corrected by the tyre diameter difference and the kinematic slip during cornering is also subtracted:

$$S_{CORR} = S_{AX} - S_{TD} - S_{STA} \qquad (3.2)$$

S_{TD} = slip correction corresponding to the tyre diameter deviation
S_{STA} = kinematic slip at cornering.

The speed differences occuring during cornering are taken from a memory stored map of characteristics as a function of steering angle, vehicle speed, and driving direction (forwards and backward respectively).

3.7 Slip Control

The corrected slip is passed on to the slip controller as the actual value (Fig. 3.5) and is compared to the specified values which are predetermined - separately for drive and coast - in dependence of driving speed, tractive force or steering angle.

By selection the functions of the specified values, an optimum torque distribution adapted to the actual tractive resistance is possible. The slip principle also takes any other reason for loss of tyre-road adhesion into account, such as rough surface uneveness.

As the basis quantity of the slip controller, a piston speed VP_S is suggested, which is compared to the results of the other two controllers.

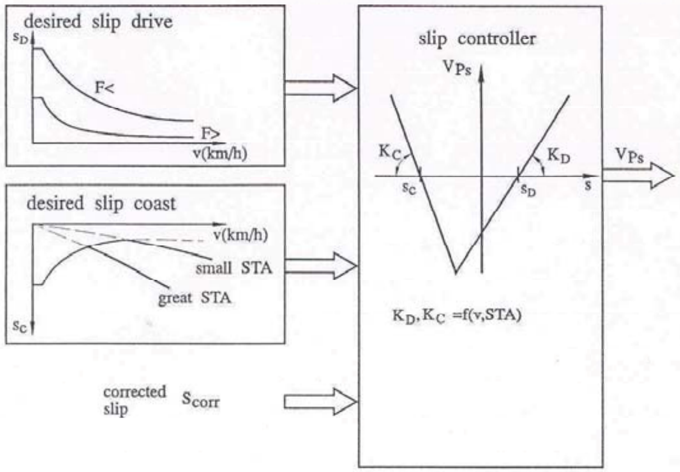

Fig. 3.5: Principle of slip control

3.8 Pre-positioning

The slip control fulfills to a high extent the dynamic driving requirements, but at sudden changes of the load condition the specified value can be exceeded shortly. These exceedings are noticeable in the steering tendency. Under certain conditions, it is therefore favorable to position the viscomatic piston directly, in order to reduce the regulating time for the slip control (Fig. 3.6).

The positioning suggestion dependent on the torque is shifted in a way that at any time more torque is transfered to the front axle than to the rear one. Thus, a wind-up of the drive train during cornering, or in case of different tyre diameters is prevented.

The positioning depending on the vehicle speed and steering angle is limited to the range of small kinematic speed differences because of the same reason (low speed, small steering angles). It serves as a means to avoid sudden slipping-through of the front axle, when abruptly engaging the clutch, and generally reduces the piston movement at low vehicle speeds (e.g. stop-and-go traffic).

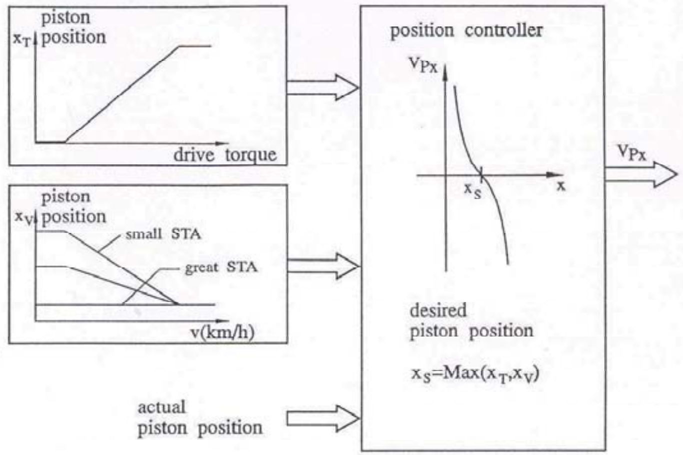

Fig. 3.6: Piston position control strategy

3.9 Brake Control

The objective of this controller is to maintain the advantange of a four-wheel drive system - distribution of the engine coast torque on all four wheels - at light braking.

Similar to the ABS a reference velocity (=speed) is defined (Fig. 3.7). As long as the rear axle speed does not fall below this reference speed less than the speed-barrier Δn, any position of the piston is allowed, and at the same time coast torque is taken away from the rear axle.

By selecting a suitable reference decelaration (0.15 g) and the function for the speed barrier Δn, the activation of the four wheel drive is limited to the range of low braking decelerations and driving speed.

When remaining under the barrier, the controller predetermines a piston position that may exceed the mechanical possible limit (in the diagram: negative). The resulting control deviation leads to the retraction of the piston at maximum speed, and thus to the seperation of the drive train.

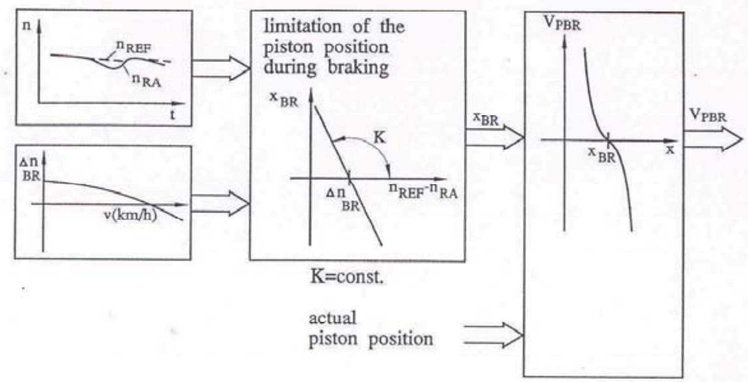

Fig. 3.7: Brake control strategy

3.10 Control Output Variable

The piston speed is determined according to Fig. 3.8.

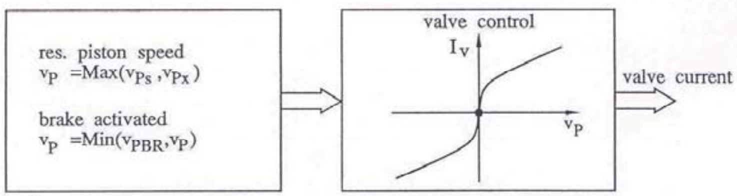

Fig. 3.8: Determination of control variable

At normal operation (no braking) the piston is moved at the mathematically higher speed out of the two suggestiones by positioning and slip control. That means that it is pulled forward quickly, but retracted slowly. Thus, disturbing traction influences can be compensated. In case of braking, the piston speed determined for normal operation is limited with the brake control speed, up to the maximum retraction speed; thus, negative effects on the braking stability and the control quality of the ABS are avoided.

4. Driver Information System PDIS (Level 2 System)

The contact forces acting between tyre and road in order to maintain a course or to keep the necessary distance result from multiple different, interacting quantities, which, depending on the causes can be divided into

- road associated characteristics and
- vehicle- and speed-associated characteristics.

Sole study of the coefficient of friction on the road provides insufficient information on the current tyre-road situation. On roads with good grip but considerable uneveness, for example, tyre road adhesion conditions typical of "low friction" prevail. It is therefor the objective of the method descibed to draw conclusions on the current tyre-road adhesion conditions and thus the current safety margin on the basis of observations of dynamic vehicles reactions. This approach then does not investigate the causes for the change in tyre-road adhesion condition, c.f. [6]. Instead of the coefficient of friction as defined conventionally, the tyre-road adhesion value κ

$$\kappa = \frac{\text{maximum achievable body acceleration}}{\text{acceleration due to gravity g}} \tag{4.1}$$

is defined as tyre-road adhesion indicator. This means that, dependig on the driving condition, κ_{LAT} can be indicated for pure cornering, κ_{LONG} for pure longitudinal dynamics and κ_{GEN} for combined cases. It is clear that this type of evaluation is limited to driving maneuvres with sufficiently low changes of the dynamic driving condition.

4.1 Determination of Tyre-Road Adhesion in Lateral Direction

Determination of tyre-road adhesion value at conering is based on the current steering angle which is necessary in order to pass a curve at a certain speed. From the steering angle δ_H the Ackermann fraction δ_A can be seperated, which ensues when driving a curve force-free:

$$\delta_K = \delta_H - \delta_A \,. \tag{4.2}$$

The remaining, corrected steering angle δ_K is the additional steering angle needed in order to maintain the equilibrium in lateral direction via the tyre slip. Due to the relationship between the necessary steering angle δ_K and transvers acceleration a_Y, typical lateral stiffness characteristics result for different tyre-road adhesion characteristics. They can either be determined by measurement techniques or by dynamics simulation, c.f. [5]. Fig.4.1 shows a lateral stiffness diagram determined by simulation for cornering with the prototype vehicle equipped with 205/50R15 tyres on 6J rims.

For quantitative correlation of the tyre-road adhesion value κ_{LAT} it is useful to consider the ratio between achievable critical acceleration and nominal acceleration of 10 m/s^2. Depending on the tyre-road adhesion, this critical acceleration occurs in the ranges $10\% < \delta_K < 20\%$. For the vehicle under investigation, the maximum lateral acceleration that is still well managed is about 9 m/s^2 (vehicle associated), which corresponds to a tyre-road adhesion value of $\kappa_{LAT\_MAX} = 0.9$.

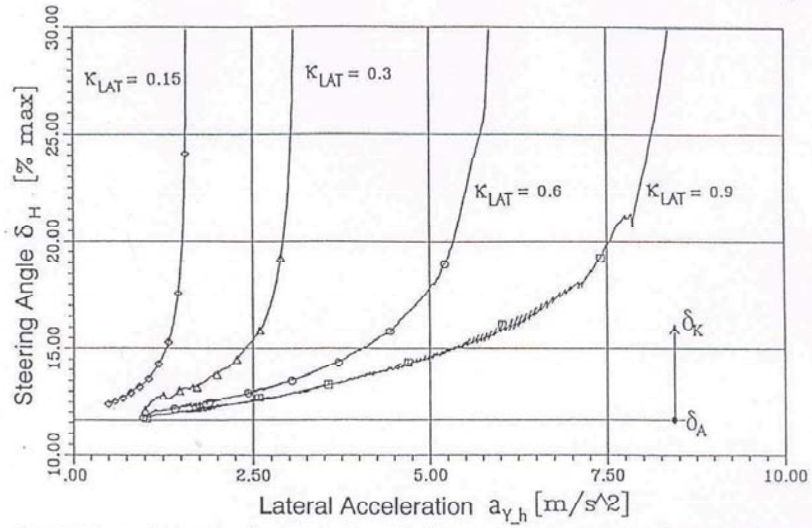

Fig.4.1: Lateral steering characteristics of vehicle under serveral adhesion conditions

For the practical application it is advantageous to relate the lateral acceleration to a body-fixed coordinate system (a_{Y_F}) and to use a more suitable form of the functional representation. Fig. 4.2 shows the modified tyre-road adhesion map for lateral dynamics with respect to a_{Y_F}, which again uses the corrected steering angle δ_K. The closed description

$$\kappa_{LAT} = \kappa_{LAT}\,(\delta_K, a_{Y\_f}) \tag{4.3}$$

can be achieved by way of a special algebraic approximation function.

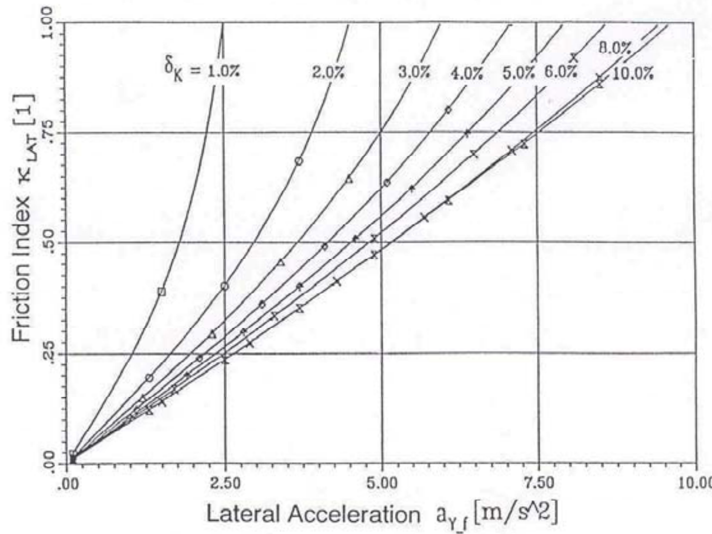

Fig.4.2: Lateral adhesion characteristics

Unique determination of the tyre-road adhesion is always ensured in case of understeering vehicles. Upon the implentation in the vehicle under real driving conditions, however, evaluation of this map is only possible within a limited area due to measurement errors and impact shock on the vehicle. This, however, is not discussed in the present paper.

4.2 Determination of Tyre-Road Adhesion in Longitudinal Direction

Determination of tyre-road adhesion in longitudinal direction is based on the relation for the longitudinal slip λ of a wheel

$$\lambda = \frac{v_X - r_{RE}\,\omega_{RAD}}{v_X} \tag{4.4}$$

which, by a differentiation with respect to the time yields

$$\dot{\omega}_{RAD} = \frac{\dot{v}_X(1-\lambda)}{r_{RE}} = \frac{a_X(1-\lambda)}{r_{RE}} \tag{4.5}$$

This means, that the ratio between rotational acceleration of the wheel $\dot{\omega}_{RAD}$ and the longitudinal acceleration a_X depends essentially on the longitudinal slip λ and to a less extent on the variable effective tyre radius r_{RE} . It can be shown that there is a vehicle specific correlation between rotational acceleration of each wheel $\dot{\omega}_{RAD}$, the longitudinal body acceleration a_X and the tyre-road adhesion condition. This typical correlation has been determined also by simulations at different tyre-road contact conditions as described in [5], Fig 4.3 . For practical reasons the front wheels are selected for identification because their slip is always greater during both driving and braking.

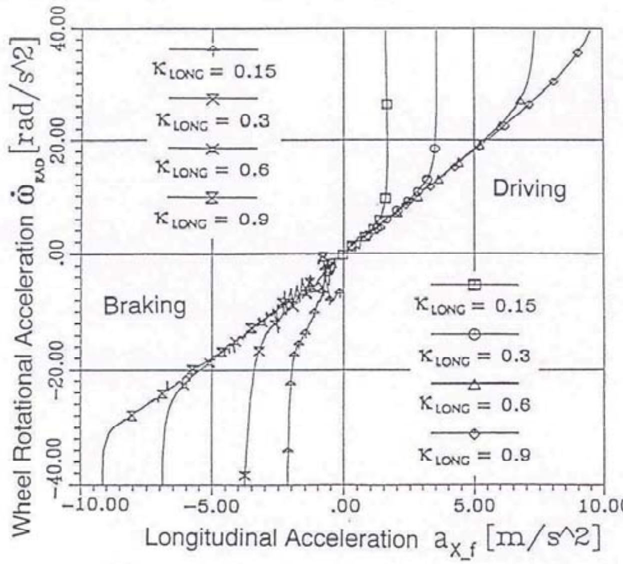

Fig. 4.3: Longitudinal characteristics of vehicle under several adhesion conditions

The resulting series of curves is in principle similar to the characteristics of tyre-road adhesion in lateral direction. Again employment of vehicle associated acceleration a_{x_f} is advised as well as the representation of tyre-road adhesion by

$$\kappa_{LONG} = \kappa_{LONG}\,(\dot{\omega}_{RAD}, a_{x\_f}) \,. \tag{4.6}$$

Fig. 4.4 shows an example of longitudinal adhesion characteristics, where only the braking range is given. Again one has got a non-resolvable area due to the existing signal noise.

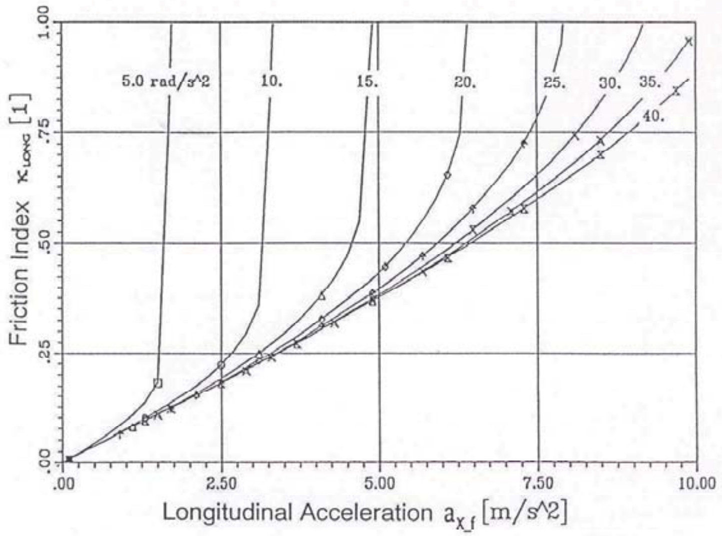

Fig. 4.4: Longitudinal adhesion characteristics

4.3 Tyre-Road Adhesion at Combined Driving Conditions

For the general case of combined longitudinal and lateral acceleration, the vector property of the tyre-road adhesion $\kappa_{GEN} = [\kappa_{LONG}, \kappa_{LAT}]$ where $\kappa_{GEN} = |\kappa_{GEN}|$ is asssumed. It describes in particular the decrease of the tyre-road adhesion value because of

$$\kappa_{LONG} = \kappa_{GEN} \cos\psi \tag{4.7 a}$$

$$\kappa_{LAT} = \kappa_{GEN} \sin\psi\,, \quad \psi = \arctan\frac{\kappa_{LAT}}{\kappa_{LONG}} \tag{4.7 b}$$

The correctness of this assumption can be confirmed experimentally as shown below.

4.4 Determination of Current Driving Limits

Efforts are aimed not only at describing the current tyre-road adhesion conditions, based on observation of the above mentionend vehicle reactions as they read

$$\kappa_{GEN} = \kappa_{GEN}(a_{x\_f}, a_{y\_f}, \delta_K, \dot{\omega}_{RAD}) = \kappa_{GEN}(a_f, \delta_K, \dot{\omega}_{RAD}) \;, \tag{4.8}$$

but also to derive suitable warning strategies for the driver on the current safety limit. Based on the inverse relation (8)

$$a_f = a_f\,(\kappa_{GEN}, \delta_K, \dot{\omega}_{RAD}) \tag{4.9}$$

and assuming the "saturation values" of $\delta_K = \delta_{K\,MAX}$ and $\dot{\omega}_{RAD} = \dot{\omega}_{RAD\,MAX}$, the vehicle-specific curves of the maximum achievable combinations of longitudinal and lateral accelerations can be described for fixed values of the tyre-road-adhesion vector κ_{GEN}. Thus, the critical curves

$$a_{f\,MAX} = a_{f\,MAX}\,(\kappa_{GEN}, \delta_{K\,MAX}, \dot{\omega}_{RAD\,MAX}) \tag{4.10}$$

are determined. Fig.4.5 shows the typical shapes for the prototype Lancia Delta Integrale for

- $\kappa = 0.9$ dry asphalt
- $\kappa = 0.6$ wet road with poor grip
- $\kappa = 0.2$ snow-covered road and summer tyres.

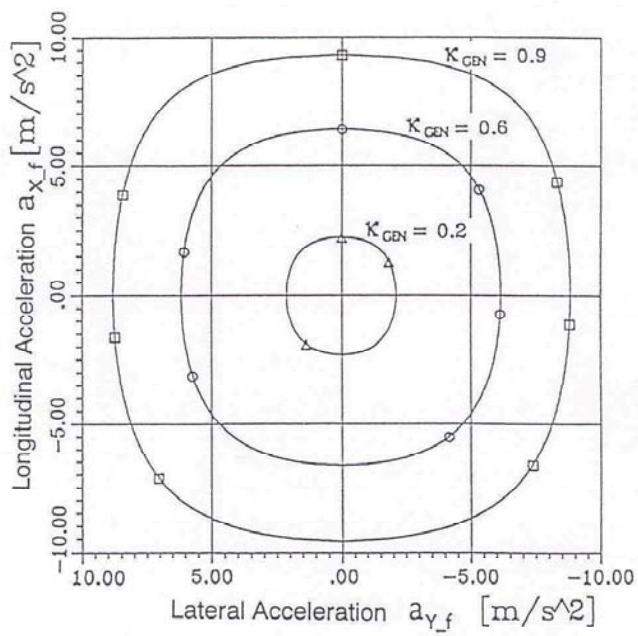

Fig. 4.5: Theoretical driving limits for the prototype at different adhesion situations

A series of measurements was performed to verify the theoretical description of the driving limits. In Fig. 4.6, two driving maneuvers on dry asphalt are compared with their corresponding theoretical driving limits. The results of an ABS-braking at steady state cornering (measurement 1) and a jerking of the vehicle at fully braking (measurement 2) can be seen. Furthermore, Fig.4.7 shows the curves for ABS braking at steady state conering on natural snow (with summer tyres) and the driving limit curve corresponding to a Tyre-road adhesion value of κ=0.2 .

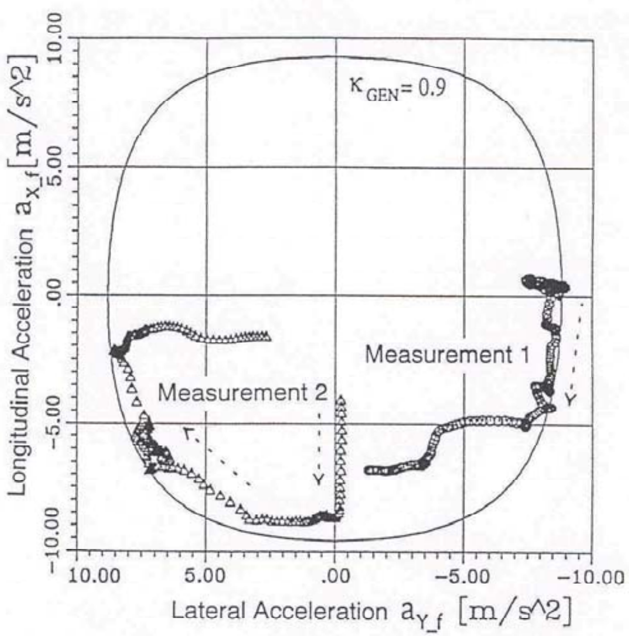

Fig. 4.6: Comparison between theoretical and measured driving limits (on dry road)

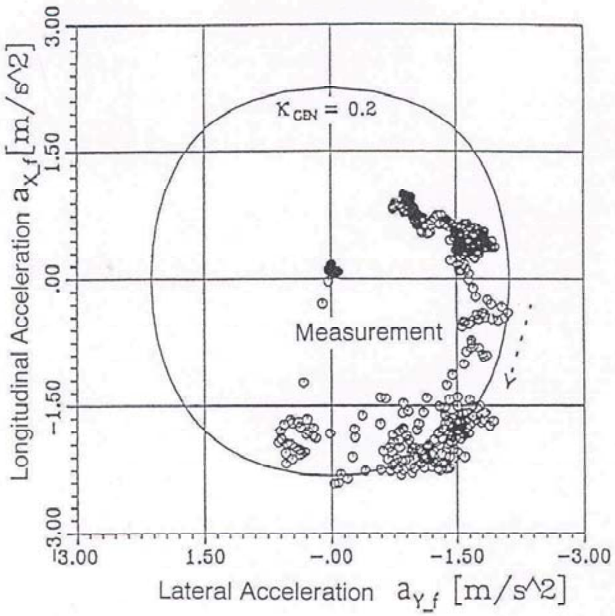

Fig. 4.7: Comparison between theoretical and measured driving limits (on snow)

4.5 Determination of Remaining Safety Margin

If the limits concerning the vehicle´s dynamical capabilities are known, it is possible to determine the current safety margin of the vehicle. The following definition seems appropriate: the current safety margin is determined by comparing the driving state vector $\mathbf{a_f}$ and its related maximum possible limit vector $\mathbf{a_{f\ MAX}}$. The maximum possible potential, which is given by the relation

$$d_{SAF} = \frac{|a_f|}{|a_{f\,MAX}|} \qquad (4.11)$$

is indicated to the driver on a diplay in the vehicle, Fig. 4.8. The device presently installed in the vehicle serves only to demonstrate the function of the system in the research and testing phase. Efforts have to be made to improve the man-machine-interface, If the safety margin distance is very small, the driver is warned by a red ligth combined with a danger-signal.

Presently, the display unit for driver information consists of

- green for large safety margins
- yellow for medium margins
- red for small or no safety margins.

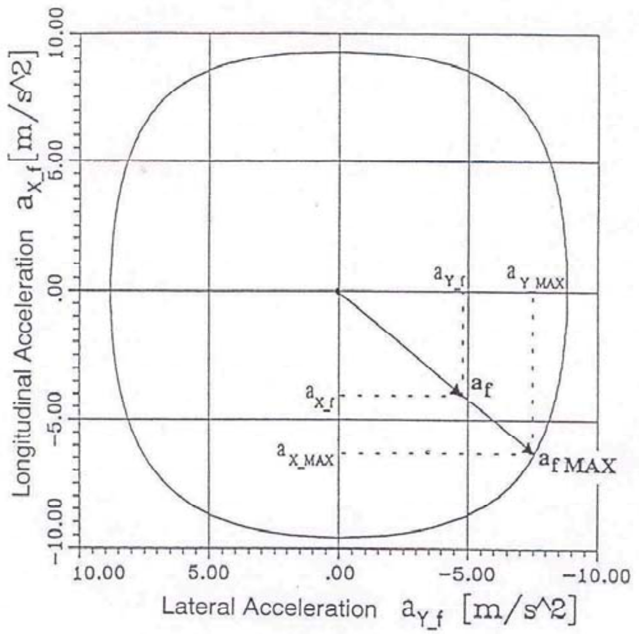

Fig. 4.8: Definition of the current safety margin

5. Electronic Realization

Practical application of the theoretical methods as described in the previous chapters require an efficient real-time data aquisition and process computation device in the vehicle. Special micro-controllers (MC), which are qualified for automotive on-board operation serve for this tasks. In particular, MCs build up a very compact contol unit due to their high integration density. The great advantage of these MCs is that the hardware for

- analog-digital conversion
- time series analyses for rotational speeds
- pulse width modulation for analog signal output
- monitoring functions (watchdogs),
- process memory etc.

are already contained on one chip. Thus, disturbances to data exchange between these elements can be reduced to a minimum. In particular, the sensor signals which are transfered to the MC must first be processed properly. The sensors available nowadays in automotive engineering have already integrated the necessary signal processing hardware to supply standard input signals for processing.

5.1 Sensor Technology and Signal Processing

The following sensor signals are processed as input quantities for the driver informations algorithm (and partially for the AWD torque split device).

- longitudinal, lateral and vertical body accelerations
- steering wheel angle
- wheel speeds (via ABS)

	Acceleration	Steering angle	Wheel speeds
Measuring range	±1.5 g	±720 deg	0.1 - 200 rad/s
Resolution	0.01 g	0.35 deg	0.003 rad/s
Linearity	±0.015 g	0.1 % (for ±90 deg)	
Frequency range	0 - 20 Hz	0-700 deg/s	
Relative error	< 2 %	< 0.5 %	< 0.002 %

The signals from the accelerometers and the steering angle sensors must be additionally processed since they are not located in the immediate nearness of the control unit. Therefore the sensor signals are subject to considerable disturbances, which, however, can be compensated in a special filter. Furthermore, level adjustment is performed for maximum possible resolution of the subsequent analog-digital (A/D) conversion. The controllers used feature A/D converters with 10 bits of resolution according to the method of successive approximation. Eight input channels are available, which are fed to the sample and hold (S/H) unit via a multiplexer. The conversion time per input is 20 µs at a central processing unit (CPU) pulse frequency of 12 MHz.

The A/D conversion is followed by an error and linearity correction module. The subsequent phase of digital signal filtering under real time conditions is an further important step in the signal processing line.

At the same time the additionally necessessary operations

- differentiation with respect to the time
- coordinate transformation
- time lag shift and
- phase sychronization

must be applied to the individual signals. The time-lag element, in particular, is needed to delay the steering angle signal in accordance with the lag in the build-up of the tyre side forces. Then all signals are available in a suitable form for the subsequent computations, Fig. 5.1 .

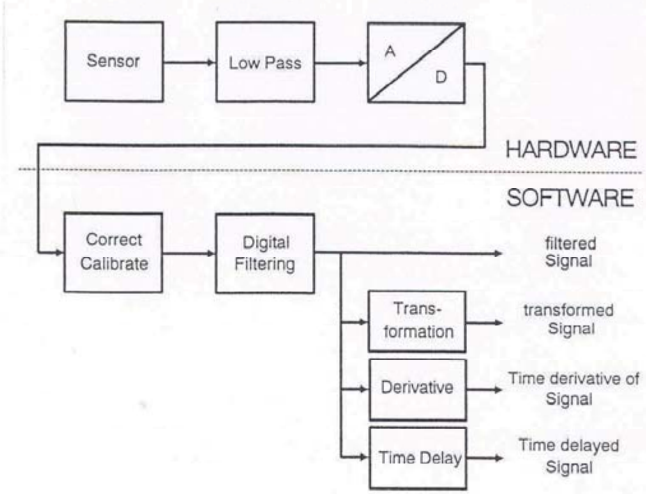

Fig. 5.1: Signal processing

The longitudinal speed of the vehicle can be approximately determined from the wheel speeds (due to the vehicle´s AWD) and current angular accelerations of the wheels are got by speed differentiations with respect to the time.

The sensor signals of the wheel speeds (from the ABS sensors) are fed in analog form, via the ABS control unit, to the high speed input (HSI) unit, where the timestep between two or, depending on the speed, several teeth of the sensor disk is measured to calculate then the angular wheel speeds. From there the data are transmitted to the information control unit via the fast serial interface CAN-bus, Fig.5.2 .

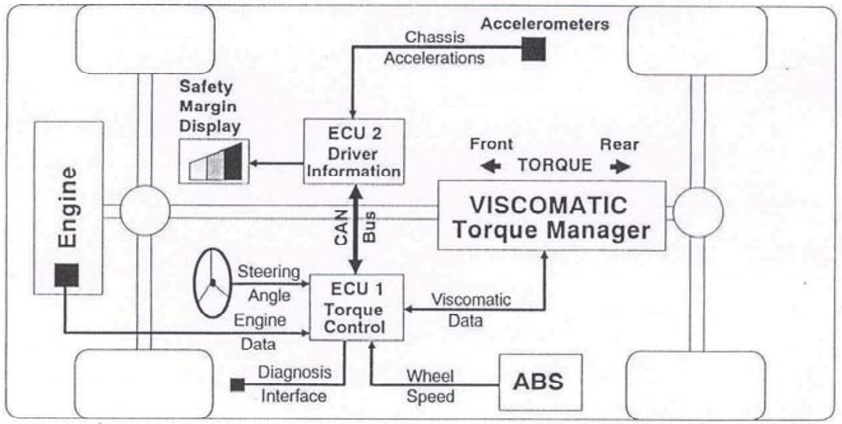

Fig. 5.2: Electronic driver information system and torque split management

5.2 Software

The structure of the comprehensive program system for identification of tyre-road adhesion and for determination of the current safety margin as descibed is shown in Fig.5.3. The kernel of the system is the "driving condition identification" module, which handles not only the logic of the procedure but also provides various checks on possible irregularities, e.g. oversteering driving situations. A seperate program module identifies and corrects road slopes to compensate for the effect of acceleration due to the gravity on the accelerometers and to detect only accelerations which are caused by tyre forces.

With the results obtained from combined tyre-road adhesion determination the margin to the reserve to the current driving limit is then assessed. The current results of the computations are displayed on the clearly visible display unit, which indicates the safety margins for the driver.

Due to the limited on-board hardware capabilities and because of the possible serial production in future, the software is written in the programming languages ASSEMBLER and C in order to fullfill the real-time requirements for cylce time of 10 µs. The multitasking structure of the operating system allows, apart from the actual execution of the program, also simultaneous data acquisition of additional data as shown in Fig. 5.3

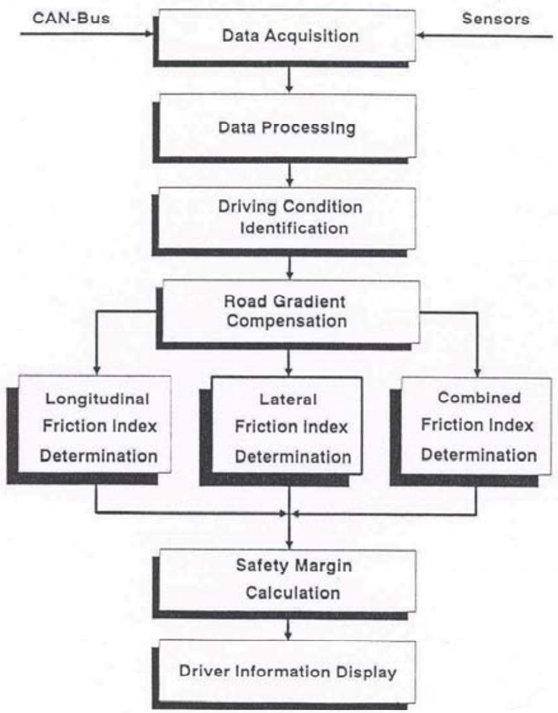

Fig. 5.3: Structure of program system

5.3 Hardware

The 16 bit CPU, which is imployed, has got the following features:

- pulse frequency: 12 Mhz
- power supply: 9 ... 30 V
- memory 64 kB
- temperature range: -40 ... +85 deg C
- interfaces: RS232, CAN.

6. Conclusions

The researches and improvements carried out on active torque split system of AWD vehicles led to a slip controlled system under elimination of the well known weak points of ordinary viscous coupling systems, namely poor braking stability without additional device, loss of steering comfort at tight cornering etc. For this, the electronically controlled and continuously working torque split device VISCOMATIC represents the actuator. Particularly, by making use of the existing couplings between the longitudinal and the lateral tyre contact forces for an enhanced control stratgy the lateral dynamics can advantageously be influenced too. Excellent traction is a self-understood benefit. The respective works have been carried out within the *Level 1* phase of the SFT-project.

The second step was set in the *Level 2* project phase, where efforts were done for driver information on current driving safety reserve. For the resulting information system PDIS some already existing signals and hardware components respectively could be used, whereas new algorithms and the display device transfer had to be developed. Although many, but not all states of driving can be clearly identified for subsequent driver warning, the project goal is considered to be reached with respect to the previous expectations. Especially in combination with other CED2-systems, a wide potential of additional safety improvement can be expected.

The demonstrated devices could be reduced down to compact size and they are working in a sufficent simple and reliable manner. Therefore, a medium term utilization for serial production can be expected.

While the technical aspects of the SFT-project can be considered as clearly solved there remain some questions around the MMI interface problem and, on the other hand, other non-engineering problems came up concerning human acceptance, risk compensation etc. and finally the situation of highly automatized, electronic systems in traffic within the legislation is still unclear. There were, however, some good discussions with the specialists of the PROMETHEUS PRO-GEN and the PRO-MAN working groups on the above mentioned fields. Finally one can summarize, that the results got from the CED2 project participation were the efforts spent worth.

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