Associazione Tecnica dell'Automobile
Sezione Toscana

Dipartimento di Meccanica
e Tecnologie Industriali
Università degli Studi di Firenze

Proceedings of the 3rd International Conference

INNOVATION AND RELIABILITY IN AUTOMOTIVE DESIGN AND TESTING

Under the auspices of the
Università degli Studi di Firenze

Vol. 1

Firenze - Italy, April 8 9 10, 1992

Theoretical Computation of Four-Wheel-Drive Vehicle Dynamics

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Abstract: Due to the physical couplings between the tyre longitudinal and the side forces, the lateral dynamic characteristics of all-wheel drive vehicles can advantageously be influenced by torque split control in addition to achieve optimum mobility itself. On the other side, the design of a controlled drive system is a difficult and expensive task. This leads to force the application of simulation techniques as a CAE tool already during the driveline conception phase.

The present Paper is concerned with modelling and dynamically simulatiing all-wheel drive (AWD) vehicles. In addition to the pure mechanical vehicle system the model has to be completed by controllers for drive torque split and the driver's actions themselves. The modelling of the vehicle and its subsystems is briefly decribed, including some hints on specific AWD aspects. Some selected simulation results are reported and compared with available prototype testing results.

1. Introduction

It is a well known fact that functional optimizations of automotive components are not longer practicable by pure measurement and testing when approaching to an optimization limit. In particular, due to the large number of free parameters in combination with the complex relations between the dynamical variables, effects of detailed modifactions can almost not be estimated.

Nowadays, the application of modern simulation techniques can be considered as the most important and useful combination to the development tools in automotive design. Typically all these methods constitute the group of mechanical CAE. In particular, the use of simulation techniques is motivated by the following items:

> identification of single ore multiple parameter influence, even under dispense with absolute accuracy,
> repeat investigations and perform variations without any change of environmental conditions,
> study of effects by *any* arbitrary parameter variation (even unrealistic),
> and finally by exceeding the driving stability limit without any risk.

In opposition to these obvious advantages, the application of simulation techniques is a rather expensive work, caused by:

> the need of powerful and efficient simulation software, the corresponding computer equipment
> *and* their application by experienced dynamics specialists with sophisticated knowledge on application of mechanical-electronical interfaces, such as typically existing in vehicle dynamics.

Moreover, further work should be taken into account for:

> model data acquisition and parameter identification,
> result presentation (postprocessing) including animation of motions,
> constructive design recommendations and
> system management and model documentation.

It can be stated, however, that computer simulation never will fully substitute the expensive testing and measurement series. But testing work can be reduced by better planning and easier realization.

2. Modelling of vehicle and driver

The present vehicle model is related to the all-wheel drive (AWD) PROMETHEUS-prototype based upon the sportscar LANCIA Delta Integrale. This prototype serves as a research vehicle within the European PROMETHEUS project and it is equipped with a permanent all wheel drive including active torque split, see figure 1.

Typically for complex vehicle systems under consideration of low frequency domain, the major modelling method in dynamics is represented by multibody systems (MBS method). According to the usual definition, a multibody system consists of multiple rigid model bodies, interacting by kinematic constraints, such as

> joints, bearings or other ideal rigid suspensions,

or by linear or nonlinear force-type connections like

> springs, dampers, active force elements and specific applied force laws.

Typically for modelling vehicle systems, there is an important scope for exact description of the acting tyre-road contact force laws, which are mainly responsible for a vehicles dynamic behaviour. Particularly, for AWD cars this aspect is very important.

In accordance to meet the expected accuracy of results, vehicle models are usually built up as three dimesional mechanical systems. Obviously for a complex vehicle system, the following decomposition seems to be useful:

> mechanical *vehicle model*, if necessary with separated *steering mechanism*,
> *driveline* submodel and
> *tyre force* computation module.

Furthermore, the tasks of longitudinal speed control together with lateral track steering have to be performed by external *driver controllers*.

Figure 1: AWD research prototype

2.1 Mechanical vehicle model

The existing three dimensional vehicle model consists of 21 rigid bodies. They have got 15 mechanical degrees of freedom, where at least 1 additional DOF results in case of elastic driveline, figure 2.

Model bodies	DOF		Constraints	
car body	3 trans	3 rot	0	
wheel 1...4		4x1 rot	4x5	
wheel axle body 1...4			4x6	
axle transversal link 1...4		4x1 rot	4x5	
spring-damper leg 1...4			4x6	
front stabilizing bar			6	
rear stabilizing bar			6	
steering rack	1 trans		5	
(engine flywheel		1 rot	5)	for torsional flexible driveline only

The torsional elasticity of the stabilizing bars are modelled by auxiliary springs at the lever suspensions, the bars themselves are considered as rigid. Wheel suspensions are arranged as closed kinematic loops by McPherson spring-damper elements, wheel axle bodies and transversal suspension links. Moreover, a further kinematic loop is built up by front axle loop, track rod levers, trackrods and steering rack. Four spring-damper legs support the car body in vertical direction.

The mentioned vehicle model leads to a complex multibody system, where a large set of nonlinear differential-algebraic equations can be found for description of the interactions between motion and the applied and constraint forces respectively. Concerning some useful ways for generation of the equations of motion and their numerical solution, the survey [1] can be recommended. For the present work, the simulation system DADS [2] has been using.

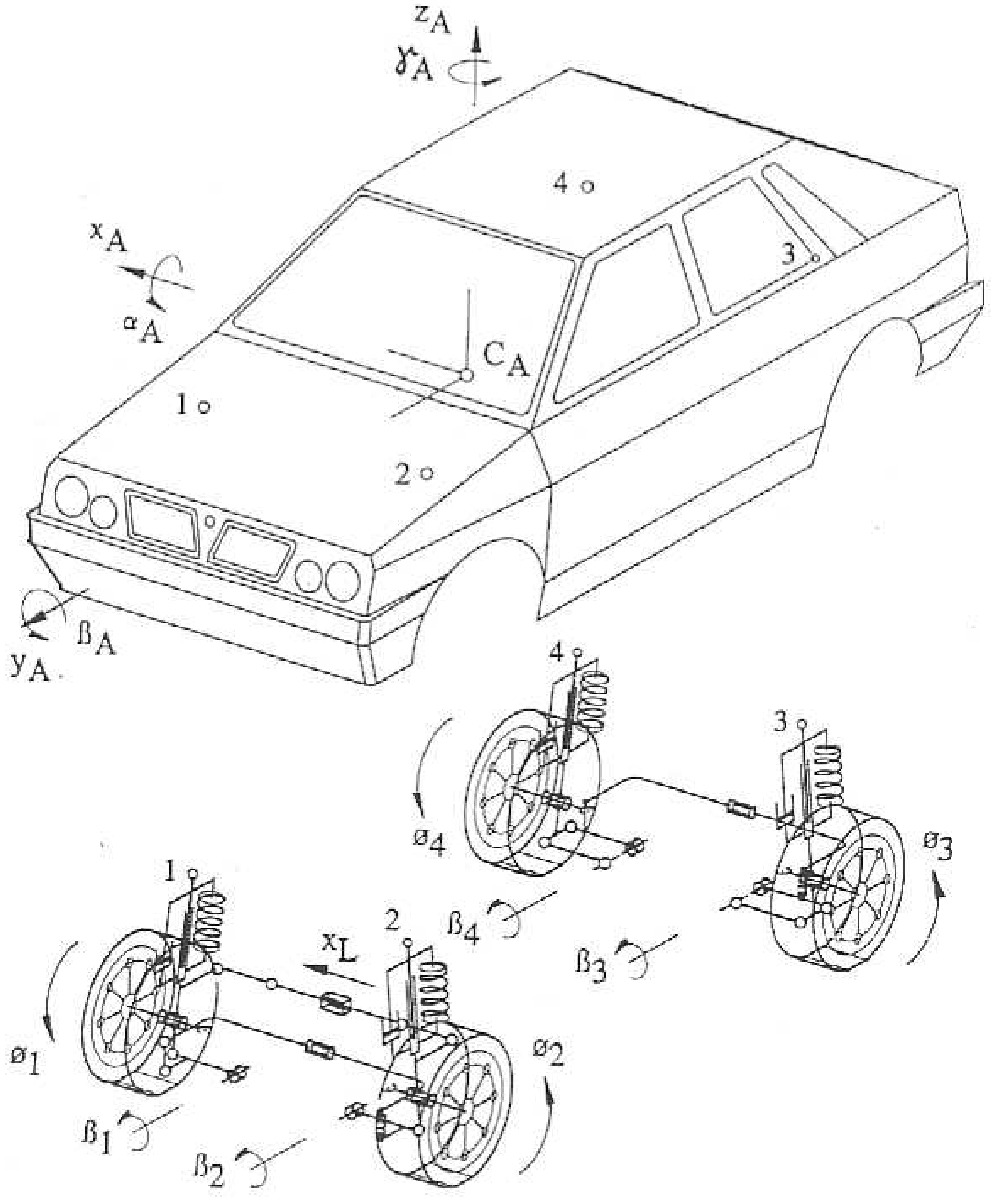

Figure 2: Multibody model of vehicle

2. Driveline model

The driveline system transfers and splits the engine torque via central differential case, the front and rear differential gears to the vehicle's wheels, see figure 3. In particular, a predefined torque split ratio can be achieved by application of a modified planet differential gear. In addition to this transfer gear, a viscous coupling (VC) performs a variable front-rear torque distribution depending on the current rotational speed differences of the axles. Thus, the VC is acting as parallel lock with both the extreme functions

> fully unlocked: pure central transfer drive,
> fully locked: rigid AWD system.

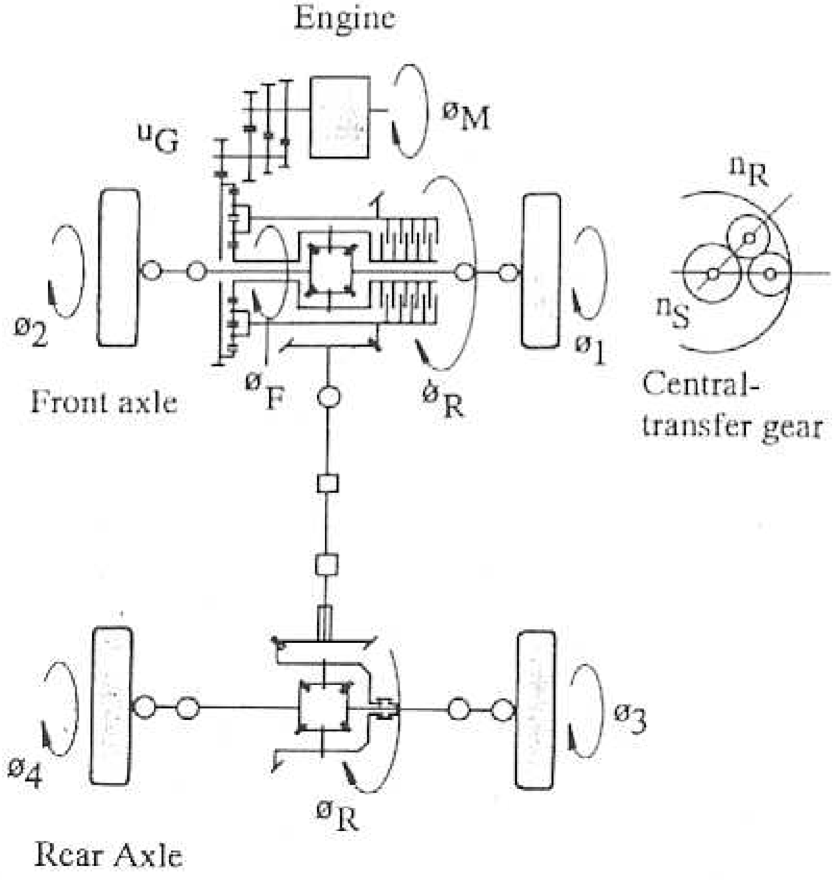

Figure 3: Structure of driveline

Assume the driveline to be ideal rigid, the driveline system has got 4 rotational degrees of freedom Φ_1 ... Φ_4. The constraint equation for the depending input angle Φ_M reads:

$$\Phi_M \, n_R/u_G = n_S \, \Phi_F + (n_R - n_S) \, \Phi_R \, , \tag{1}$$

using the auxiliary variables Φ_F and Φ_R ; u_G denotes the current total shift gear ratio. For both driven axles the simple relations hold:

$$2 \, \Phi_F = \Phi_1 + \Phi_2 \quad \text{and} \quad 2 \, \Phi_R = \Phi_3 + \Phi_4 \, . \tag{2}$$

Hence, the kinematical condition for the rotational drive line angles is obtained by

$$2 \, \Phi_M \, n_R/u_G = n_S \, (\Phi_1 + \Phi_2) + (n_R - n_S) \, (\Phi_3 + \Phi_4) \, . \tag{3}$$

Moreover, by applying the torque equilibrium conditions the resulting torque split ratio for the transfer gear yields

$$T_F / T_R = n_S / (n_R - n_S) \,. \tag{4}$$

Thus, any practicable basic torque split characteristic can be performed by proper choice of teeth ratio between ring gear and sun gear n_R / n_S . The additional locking torque of the VC will act depending on slip differences between front and rear axle in the required manner.

2.3 Driver control model

The driver's task is to keep both the nominal track and the chosen driving speed. For any computational driving simulation it is necessary to formulate the driver's active control function mathematically. For this purpose differently complicated control laws have been created up to adaptive driver controllers, which are well qualified to provide the driver's robust adaptability. However, the proper choice of the driver model application depends on what kind of results are expected for the present task. Hence, on principle there is no general *multi-tasking* driver model available.

The very frequent case in vehicle dynamics is to investigate a vehicle's reactions to standardized driving manoeuvres. More or less simple driver models can be applied, if *perfect* track keeping itself is of lower interest. The present task is to study the influences of several torque split strategies and how they change the longitudinal and lateral motion charateristics. *Vehicle speed control* can be performed by a rather simple feedback controller as shown in figure 4. Here the deviation between the actual longitudinal track speed YLD and the current reference speed YLD_n (nominal speed) serves as controller input. The controller has to take the driver's reaction time into consideration and futhermore the limited engine torque for acceleration and braking has to be modelled. Typically for the inner driveline circuit, the driver cannot directly influence the internal torque split between the axles. This is only performed by the central tranfer case and the torque characteristic of the VC, see figure 4.

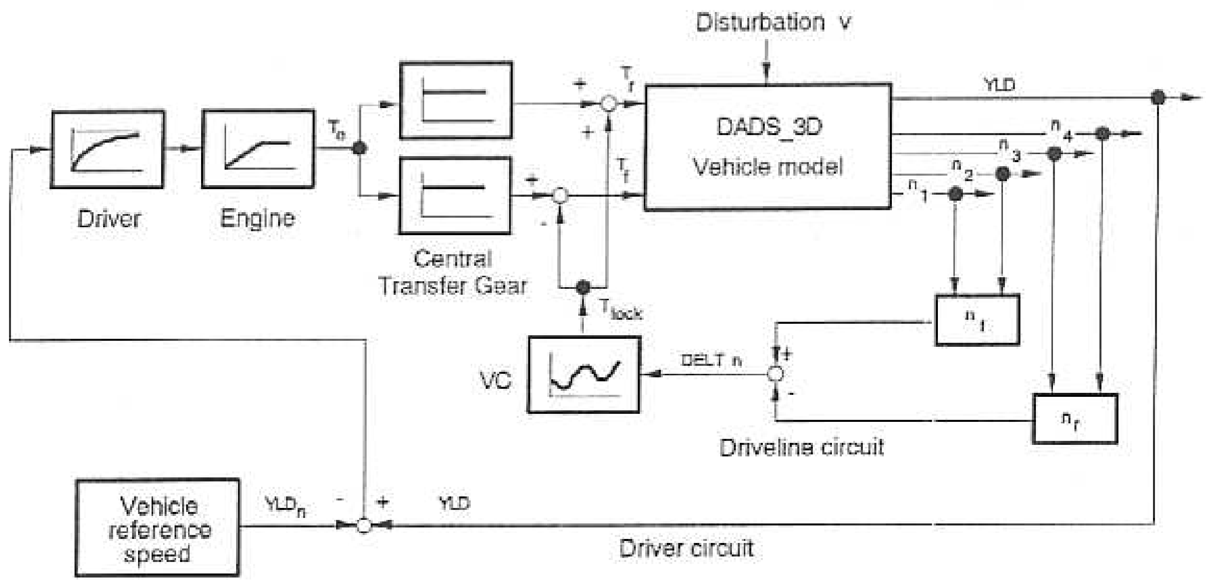

Figure 4: Longitudinal vehicle control

For the lateral track control the driver's action can be approximated by a PID controller under the above mentioned conditions. For that purpose, the feedback vector **Y***(t) includes the full kinematic state of actual vehicle motion, consisting of the generalized 6 by 1-position vector **Y**(t), the velocity vector **YD**(t) and the acceleration vector **YDD**(t), including all corresponding translational and rotational quantities. By comparison with the vehicle's current nominal position values (characterizing the defined driving manoeuvre), the input vector for PID amplifiers is found. Variation of amplifiers allow sufficiently the adaption of the driver's reacting behaviour. +)

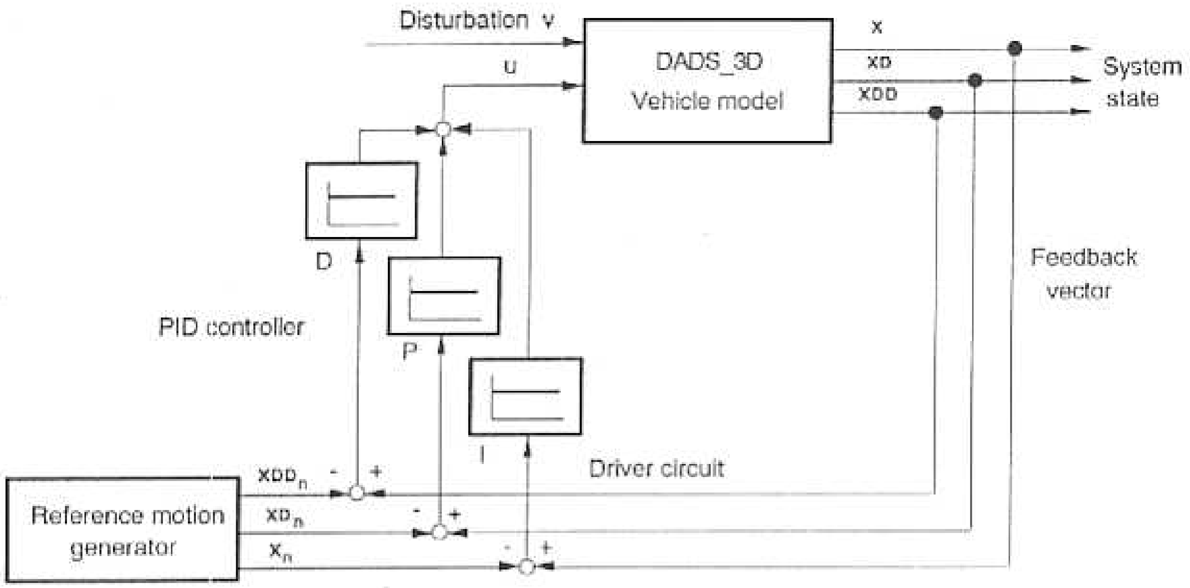

Figure 5: Lateral path control (driver)

Another important task is to define the critical limit, where the simulation run is to be terminated. Of course, this limit needs not to coincide with the formal stability criterion. Obviously, in practice the driving limit should be reached, if one ore more vehicle axles are leaving a predefined track zone. However, simple controllers as mentioned above are not expected to be able to stabilize critical driving situations beyond the stability limit, e.g. rotational skid.

+) The stability of a nonlinear system activated by a linear controller is uncertain. However, control stability can be expected, if assuming slow and limited variation of dynamical state.

2.4 Tyre-ground forces

As stated above, a vehicle's dynamic behaviour is mainly influenced by the interactions between vehicle motions and the acting tyre forces. Hence the major task is the exact description of the tyre contact force laws in any dynamical simulation. Another important, but unfortunately contrary aspect is to provide optimum computational efficiency for the applied method.

Focussing a balanced vehicle model, it does not make sense to refine a constisting vehicle model in detail without having an adequately qualified tyre model. Particularly, when modelling AWD vehicles, this aspect is very important due to the typical couplings between longitudinal and side forces. Several tyre models have been published and some of them are commercially available. However, the most successful methods for analysis purpose seem to be the *mathematical type* tyre models. There the force-slip relations are determined directly by approximation of measured force characteristics without regard for physical modelling [3], [4]. The additional transient force effects can be included by application of first order time-lag elements. This procedure is well known to work very effectively.

Tyre forces/torques processing is performed within three steps:

a) Provide current kinematical road-tyre quantities

> longitudinal slip SLP = $(v - R_e\,\Omega)/v$,
> side slip TAL = $\tan\alpha$,
> radial deflection DFL = $|r_{GP} - r_{WC}| - r_0$,
> wheel camber ß

based upon motion properties according to figure 6.

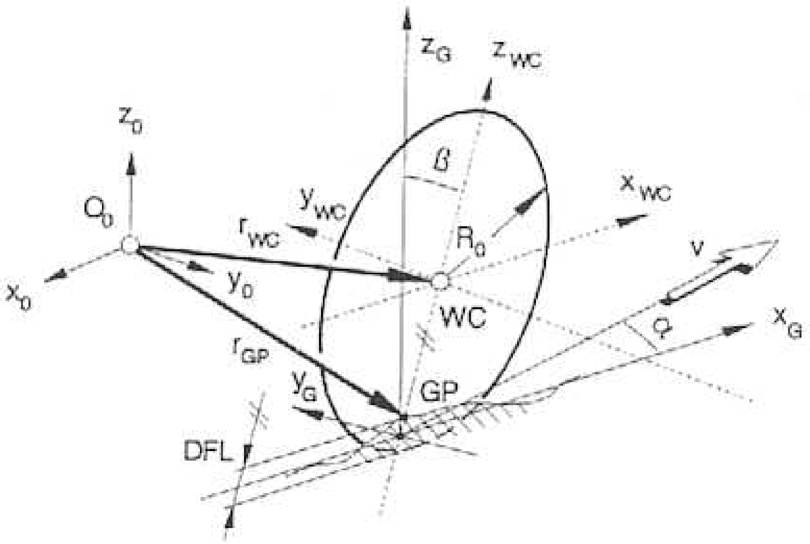

Figure 6: Definitions for wheel

It should be pointed to the importance of an exact description of the effective tyre rolling radius $R_e = R_0 + DR_e$, in particular when dealing with slip controlled AWD vehicles. Again, a direct measurement approximation for the effective radius difference function $DR_e = DR_e(F_z, \Omega)$ related to the undeformed radius R_0 is useful. Its tyre-type specific coefficients can easily be determined from experimental findings, mainly influenced by the tyre load F_z and the wheel angular speed Ω, [5]. For instance, the effective radius difference graph for tyre type 205/50 R15 is shown in figure 7. One can notice the major influence of vertical load F_z and the smaller dependency on rotational wheel speed Ω.

b) Pure tyre longitudinal and side forces

For this second step the methods described in [4], [5] are recommended, which are based upon the the mathematical approximation of measured tyre characteristics

$F_x = F_x(\text{SLP}, F_z)$	pure longitudinal force,
$F_y = F_y(\text{TAL}, F_z, ß)$	pure lateral force,
$T_z = T_z(\text{TAL}, F_z, ß)$	pure aligning torque.

The computation of these basic force/slip functions can be done very efficiently and accurately by application of the methods referenced above. Figure 8 gives an example for basic slip curves for the denoted tyre type.

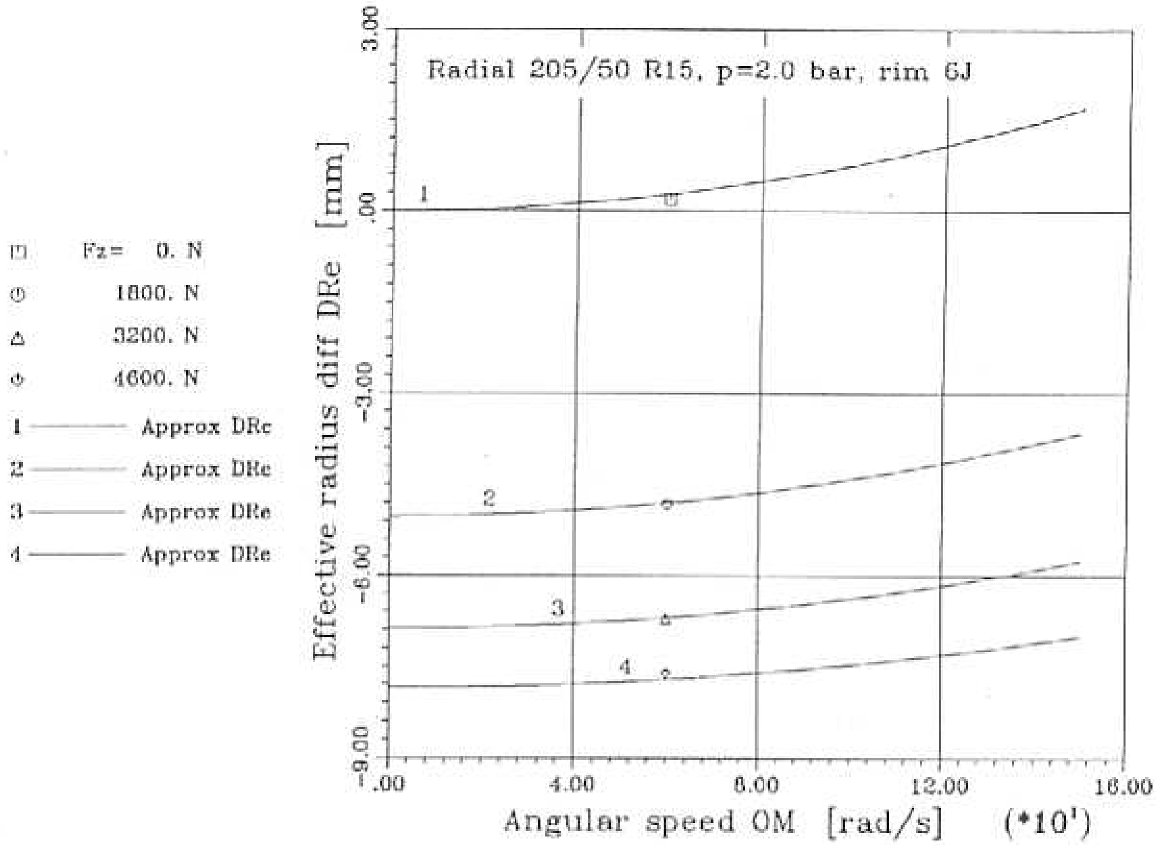

Figure 7: Effective tyre radius

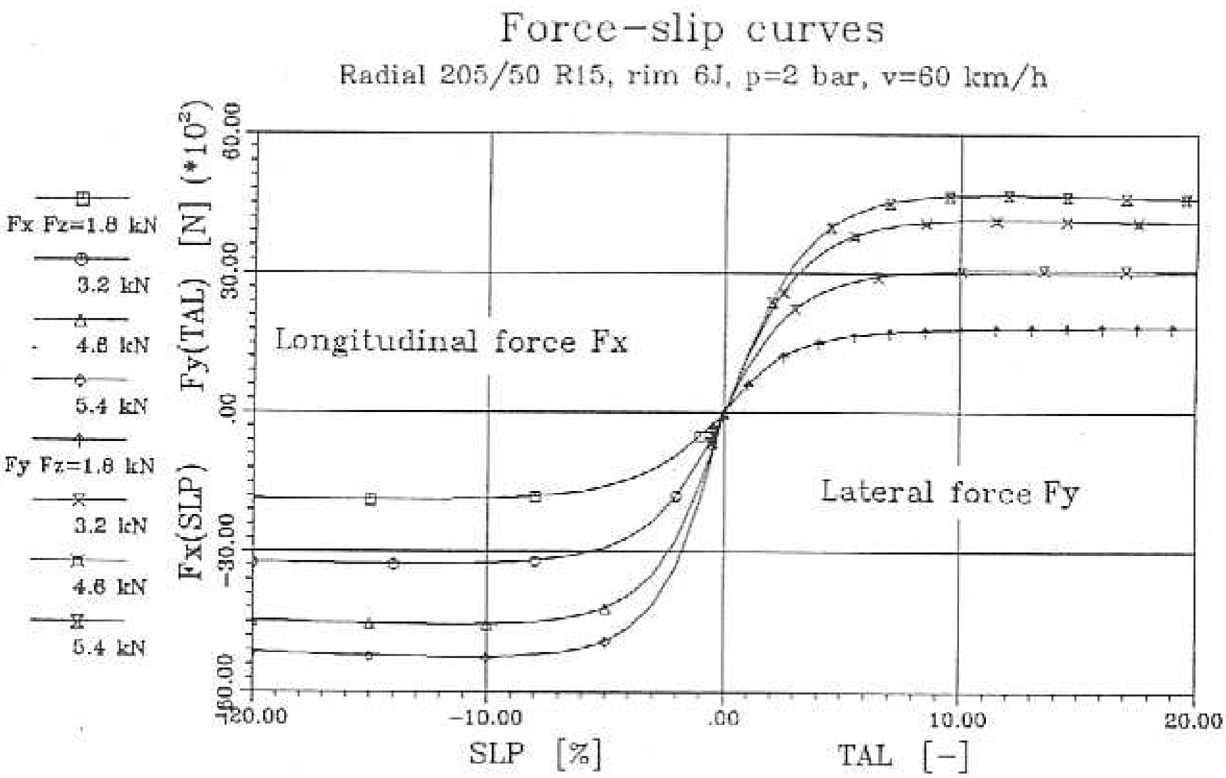

Figure 8: Basis characteristics of pure longitudinal and lateral forces

c) Combined longitudinal-lateral forces

In practice tyre forces are usually acting in a combined manner $F_{xy} = F_{xy}(SLP, TAL, F_z, \beta)$, where the well known effects of force loss caused by force superposition can be observed. Thus, these couplings between the tyre forces have to be modelled exactly when dealing with AWD vehicles. On the other hand, in addition to the optimum traction AWD vehicles advantageously allow to influence the lateral dynamical behaviour in a limited manner (active safety aspect).

A very efficient method is provided by application the *theory of similarity*, where physical equivalence of longitudinal and lateral slip is assumed. Nevertheless, non-isotropic force properties can be taken into account by specific superposition of the basic curves as shown in figure 8. In order to be able to apply this method, first of all one has to change to equivalent theoretical slip properties for longitudinal and lateral slip. For that aim, several transformations are used, e.g. [3], [4]. Figure 9 shows a calculated graph of longitudinal vs. lateral force. Smooth curves are an important condition for a successful return from an instable into a regular driving state during simulation.

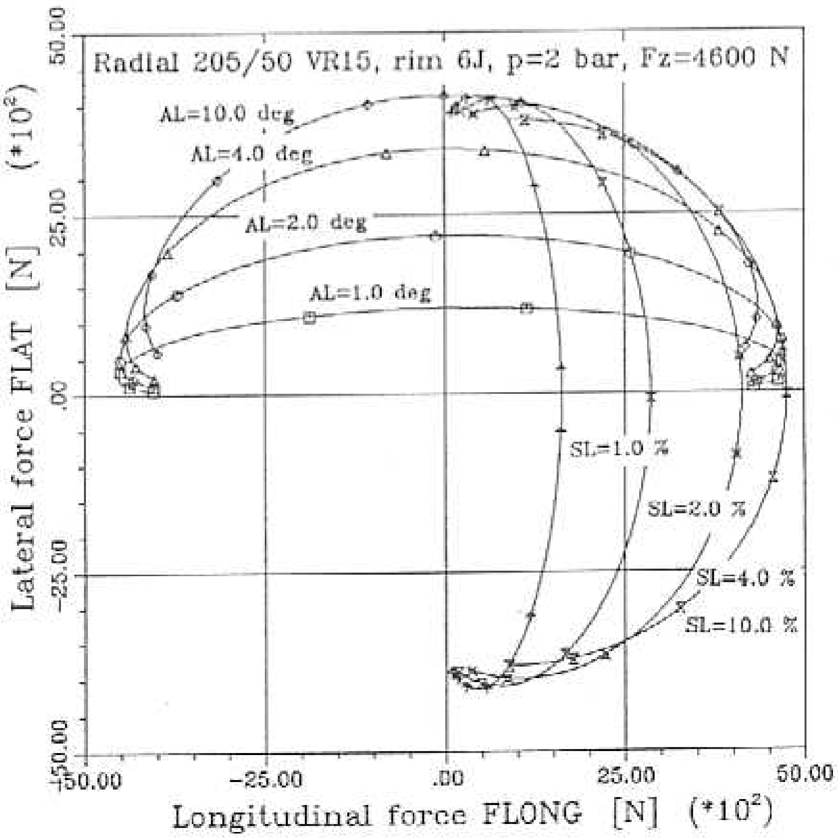

Figure 9: Combined horizontal tyre forces

3. Computational realization

The various aspects in vehicle modelling lead to the necessity for extensions of pure multibody simulation programs. Present task includes a problem combination of the type

continuum mechanics -	discrete mechanics -	system dynamics
(tyre, viscous coupling)	*(vehicle)*	*(controller)*

Due to the fact, that the closed solution of such a specific combination cannot be expected, it is important to look after the *open structure* of the simulation system used. Openness for user specific interactions should be valued just as high as the remaining requirements like processing efficiency, number of model elements, user kindness ect.

For the presented project, the multibody dynamics program DADS_3D is installed; its theoretical background and the algorithms can be found in [2]. The DADS interface *user defined forces* is a useful feature which allows to link the autonomously working tyre module DTIRE, which calls the modules DFTIRE for tyre forces computation and DRTIRE for effective radius processing. Further user defined interactions concern to the decription of torque laws for the VC. Finally, the realization of the driver controller is directly supported by a large number of standard control elements.

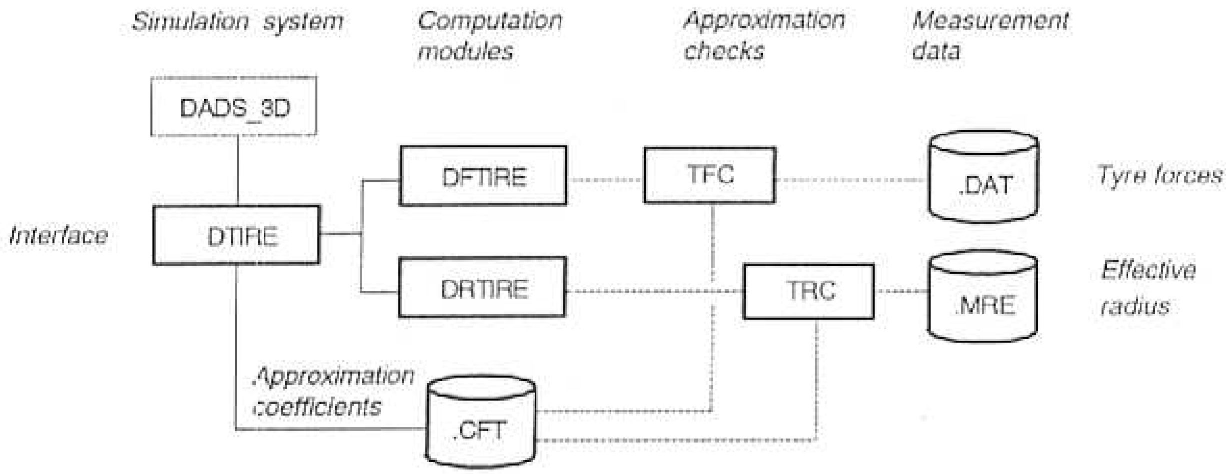

Figure 10: Structure of tyre computation module

4. Comparison of selected results

The following reference test procedures are considered:

> Steady state conering on dry asphalt,
> Slowly accelerated J-turn on slippery ground (gravel).

The comparisons between simulation and measurement results show, that modelling as well as parameter identification have obviously been done in a correct manner. Figure 11 shows a result comparison related to the nominal vehicle setup, where the required steer angle δ according to the current lateral acceleration Ay is plotted. There can be seen a linear phase of increasing steer angle, followed by understeering tendency up to the vehicle´s controllability limit.

The effect of different torque split settings are considered in figure 12. First the vehicle in basic configuration is investigated, where a statical axle torque split 47/53 % (front/rear) and additionally the VC lock is modelled. This driveline combination allows maximum lateral acceleration on slippery ground (gravel); nevertheless, the driving tendency keeps advantageously understeering.

Furthermore, there are two driveline variants considered:
Variant 1: Rear axle dominant drive without VC lock; torque split 30/70 % (front/rear). The abrupt oversteering in this case is clearly indicated.

Variant 2: All wheel drive with pure VC driven rear axle and without transfer gear. This vehicle reacts similarly to the reference car, but it cannot fully reach its maximum lateral accceleration.

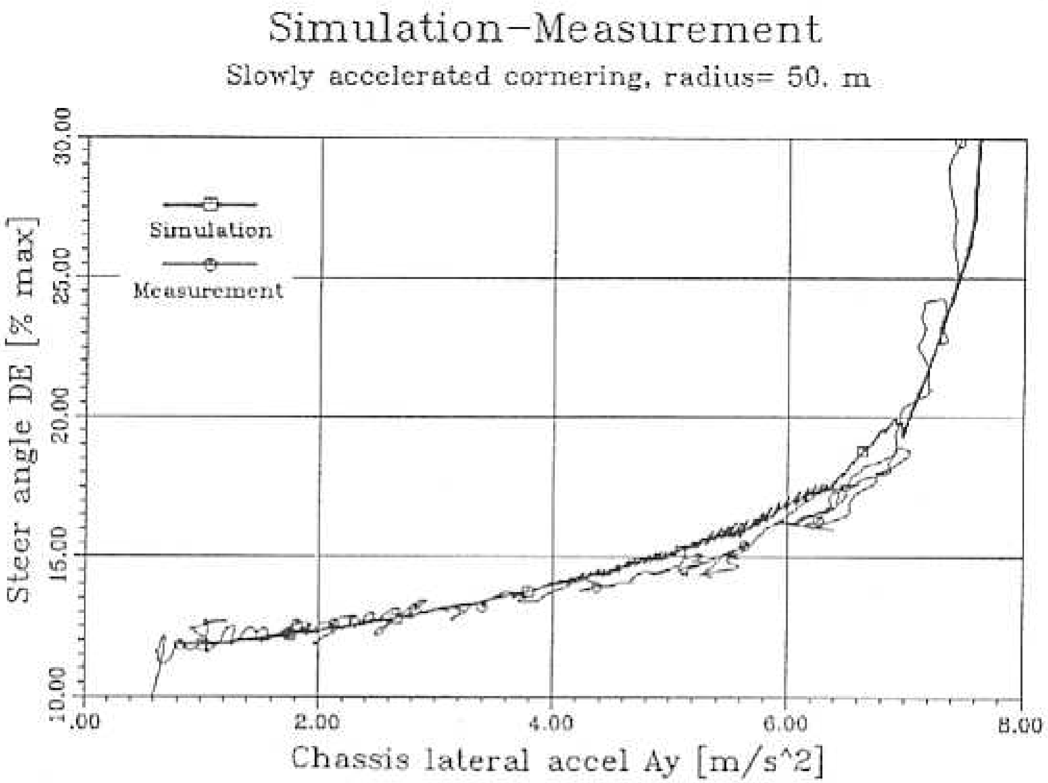

Figure 11: Steering stiffness, simulation-measurement comparison

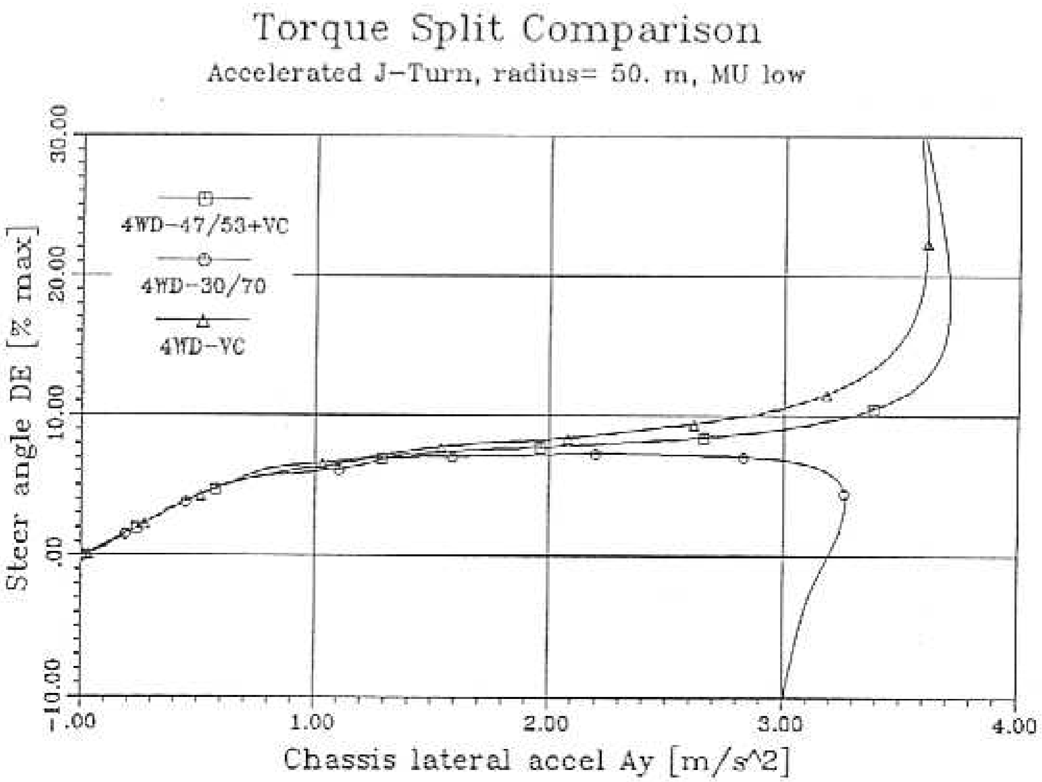

Figure 12: Comparison of different AWD-settings

The animation plot in figure 13 gives an impression of the motion during the testing manoeuvre *accelerated J-turn on slippery ground.* Again the vehicle is leaving the track curve in an understeering manner. Since the road friction is poor, the vehicle roll angle is only small and as expected, the resulting slip angles are very large.

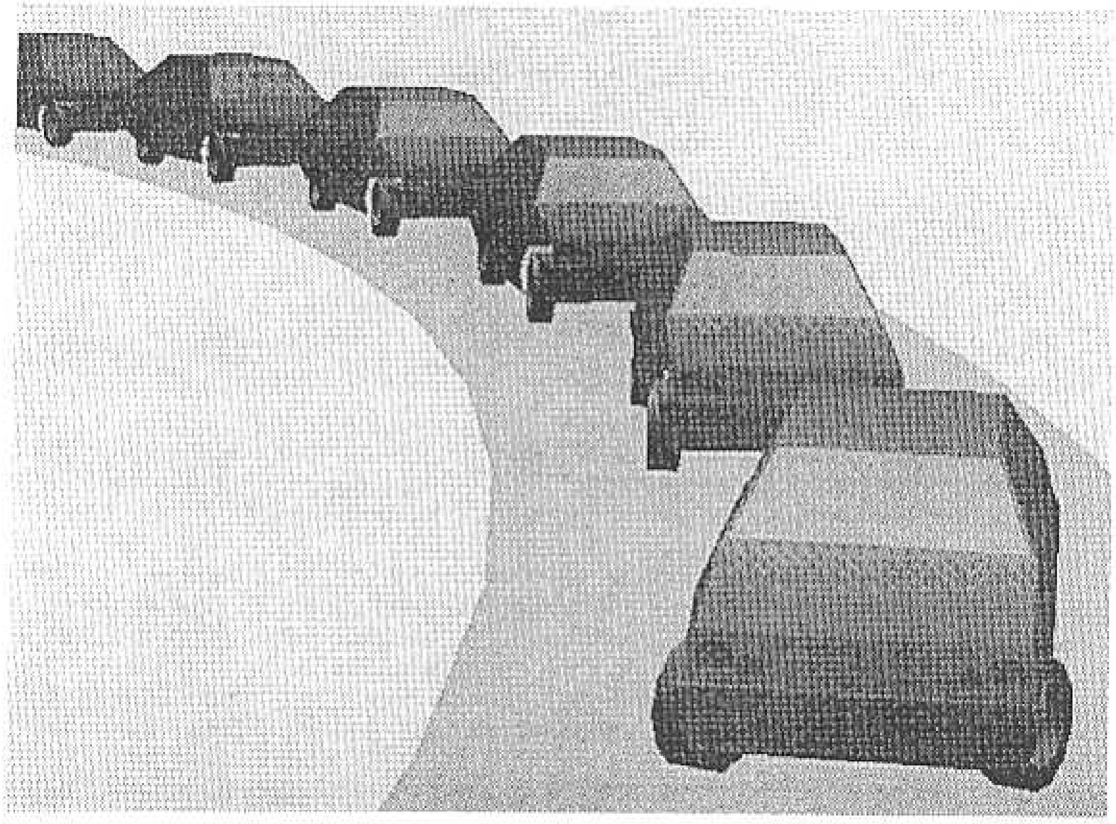

Figure 13: Animation of turn on slippery ground

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