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MODERN CONTROL THEORY APPLIED TO VEHICLE DYNAMICS

STATE-OF-ART REPORT

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Abstract

This paper gives a state-of-art report of the controlled vehicle dynamical research that has been conducted at the Technical University of Budapest and the Computer and Automation Institute of the Hungarian Academy of Sciences. The investigations into the application of the newest results of the modern control and identification theories are conducted on several fields: primary (wheel) suspension control, secondary vibration isolation (seat), steering systems, control of the hitch forces and torques at articulated vehicles, and stability enhancement control systems. There are some other fields addressed here as well, which are related to the interface between the vehicle and the environment (IHVS) and the investigations into the on-board vehicle electronics.

Keywords: modern control theory, robust control, automated vehicle systems, vehicle dynamics, identification of dynamical systems.

1. Introduction

With the sharply increasing number of the vehicles participating in the everyday traffic several other factors receive a growing interest:

- risk of traffic accidents, which is not only the consequence of the number, but also the higher performance of the vehicles,
- the need for increasing the traffic flow's velocity and safety,
- demands on the better communication between the vehicle, vehicle driver and the environment.

The above facts are pointed out very clearly that the vehicles of the end of 20th century should be smart and have a certain intelligence to fulfil all these requirements. It basically means significant increase in the job of the on-board computer. Its task will be changed essentially: it should not only communicate with the existing electronic system and control their behaviour (such as engine management, ABS, climate control, etc.) but also should handle complex problems such as vehicle identification, controller design, communication with the road properties (curvature, elevation, etc.) linked to a satellite communication system. The principal scheme of such a vehicle can be seen in *Fig. 1*. In this state-art-of report only the results related to some of the controlled dynamical systems installed on the vehicle are introduced. In the reference list, a completer picture about the publication activity conducted in this field is given [1]–[15].

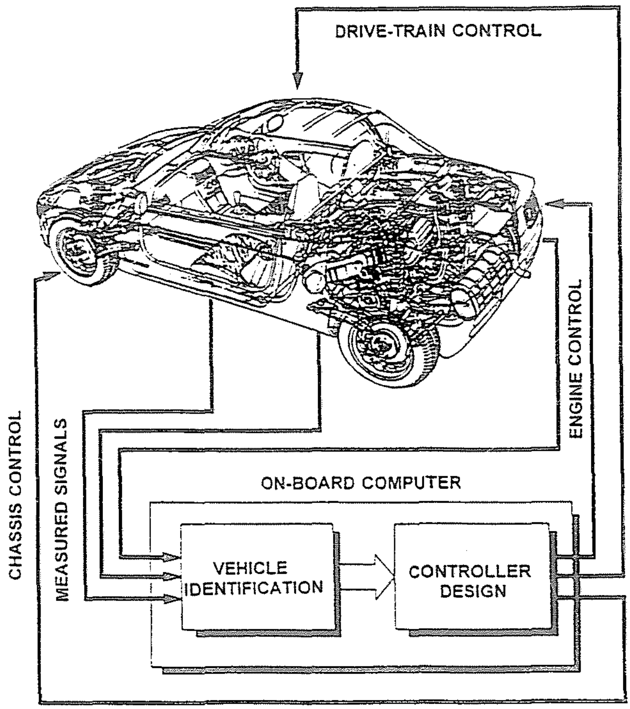

Fig. 1. General task of the on-board computer

2. General Problem Statement

Generally speaking, any controlled dynamical system can be transformed into the form shown in *Fig. 2*, which gives the basis for the controller design.

There are three blocks in *Fig. 2*. The middle one represents the nominal plant to be controlled by the lower block, which is standing for the controller. Since the real systems cannot be known perfectly (there are several facts making the system uncertain), the upper block describes the model of system's uncertainties. These can be structured (such as the physical parametric uncertainties) or unstructured (arising from high order or non-

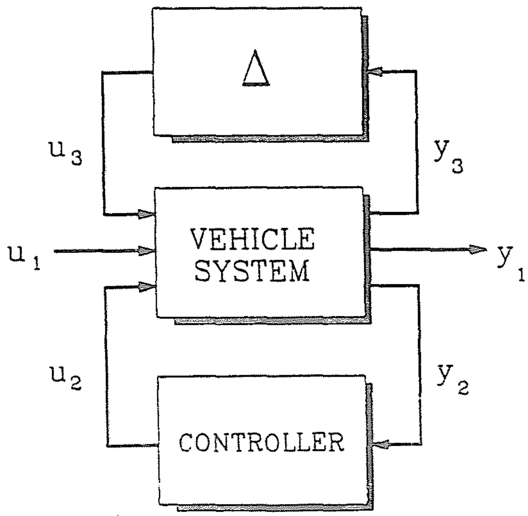

Fig. 2. Block diagram of an uncertain dynamical system

linear dynamics). The input vector u_1 stands for the external disturbances acting on the system, u_2 is the control input vector. The set of outputs denoted by y_1 is referred to as the performance output vector which has to be controlled, and y_2 contains the measurable outputs.

In vehicle dynamics, several classes of models can be used. As a first approach to the controller design problem, linear time-invariant (LTI) models are used, and thean, when implementing it in highly non-linear environment, the models can be refined with time varying and/or non-linear disturbances and uncertainties. The mathematical system theory offers (a) transfer function models with norm bounded unstructured uncertainties and (b) state-space models with both structured (parametric) and unstructured uncertainties.

In vehicle dynamical studies the system's performance output varies from application to application. For example, in the suspension design problem it contains the vertical body acceleration and the dynamic wheel load, while designing controlled 4WS system, the controlled outputs are the lateral acceleration and the yaw rate. Similarly, the control inputs are also different, in the suspension case it is a force, while at the 4WS system it is the steering angle. These variables will be mentioned later at the specific application. When talking about integrated vehicle control as shown in *Fig. 1*, these output and input signals are collected into one vector and are used in control.

Generally speaking, the control problem can be formulated as follows:

$$\|T_{y_1,u_1}\|_A(K\Delta) \rightarrow \min_{K \in k_{stab}} \tag{1}$$

that is some norm of the transfer function between the performance output and external disturbances is minimized over a set of stabilizing controllers. The solution to the control problem given in *Eq.* (1) can be obtained as:

a) H_∞ optimal control or minimax differential game, where the L_2 induced norm is used in *Eq.* (1) and it is assumed that the signals are energy constrained, bounded, i.e. u_1 and $y_1 \in L_2$.
b) l_1 optimal controller design: it is assumed that the signals are amplitude constrained, i.e. u_1 and $y_1 \in L_\infty$ and the norm in *Eq.* (1) is the induced L_∞-norm.
c) H_2/H_∞ design problem (see in [11], [12]), when $u_1(t)$ is stochastic and Δ is norm bounded.

The design steps can be listed as: (a) construct the equations of motion, (b) convert it into the state-space form and $M - \Delta$ form (this is form when the plant and controller block in *Fig. 2* are combined into a single M block) and (c) depending on the control goal choose the right algorithm to synthetize the controller. The controller design problem can ususally be solved by a commercially available software, such as MATLAB or MATRIX$_x$. If the model cannot be obtained from e.g. first principle, system identification has to be performed prior to the controller design.

3. Suspension Control

In this part of the paper three suspension related problems will be addressed: active and semi-active control of the primary suspension of a passenger car, and the seat suspension design for heavy commercial vehicles.

3.1. Active Suspension Design

Fig. 3 shows the well known quarter-car vehicle model (representing the quarter of the vehicle body with a single suspension), which is widely used for active suspension design because of its simplicity, low number of state variables and parameters. The model is drawn similar to *Fig. 2*. The performance output vector contains the following variables to the controlled (see in references [4], [6], [11], [12]):

- *Vertical body acceleration*, which describes the ride comfort provided by the suspension. This quantity should be minimized in some frequency ranges, because the human body (driver, passenger) is sensitive to vibration transmitted from the road profile to the vehicle in a given frequency range.

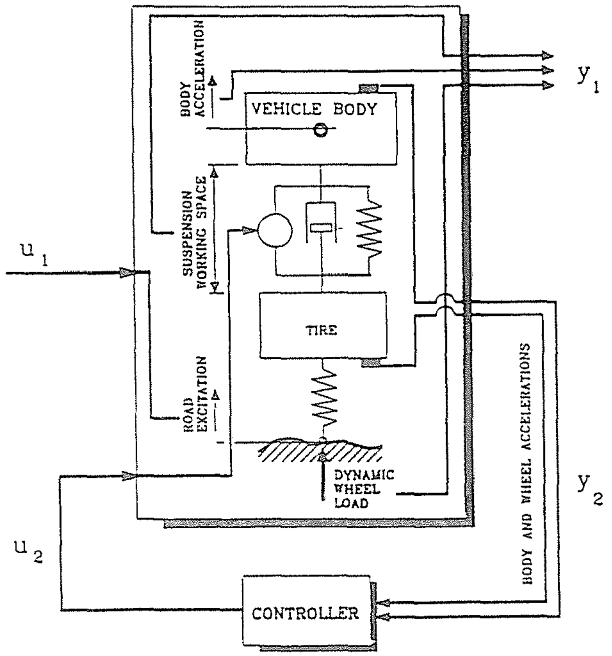

Fig. 3. Quarter-car vehicle model for active suspension design

- *Dynamic wheel load*, whose value should also be kept as minimal as possible to provide better road holding for the vehicle, increasing its lateral and longitudinal stability.
- *Suspension working space*, which is important to be as low as possible, since the large working space causing higher variation in the steering geometry is not permitted from constructional point of view.

The above described quarter-car model is not able to take several effects into consideration: effect of the pitch and roll motion, chassis flexibility, etc., but provides very good results for the theoretical investigations, testing several control schemes. This model can also be uncertain, since such variables as tire spring stiffness (which is a function of internal tire pressure) or the vehicle mass (varying with the number of passengers, or the cargo) are not perfectly known. The approach introduced in part 2 is able to handle such structured uncertainties. The solution to the combined H_∞/RLQR (Robust Linear Quadratic Control) is shown in *Fig. 4* (for more details see in [4], [12]).

Although the simulation results have shown the applicability of the developed control strategies, it should also be implemented in a physical model. For this reason, a 1 : 1 scale quarter car test rig was designed, as shown in *Fig. 5*.

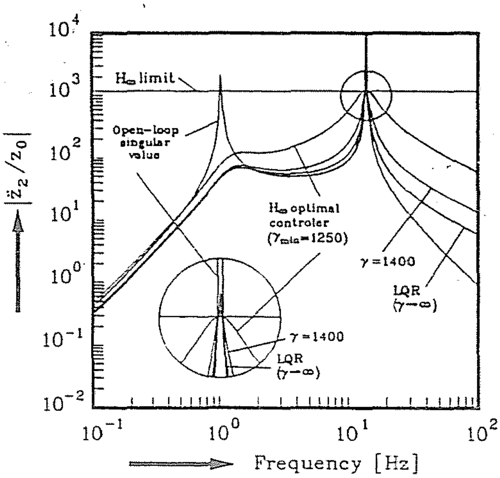

Fig. 4. Transfer functions with different control strategies

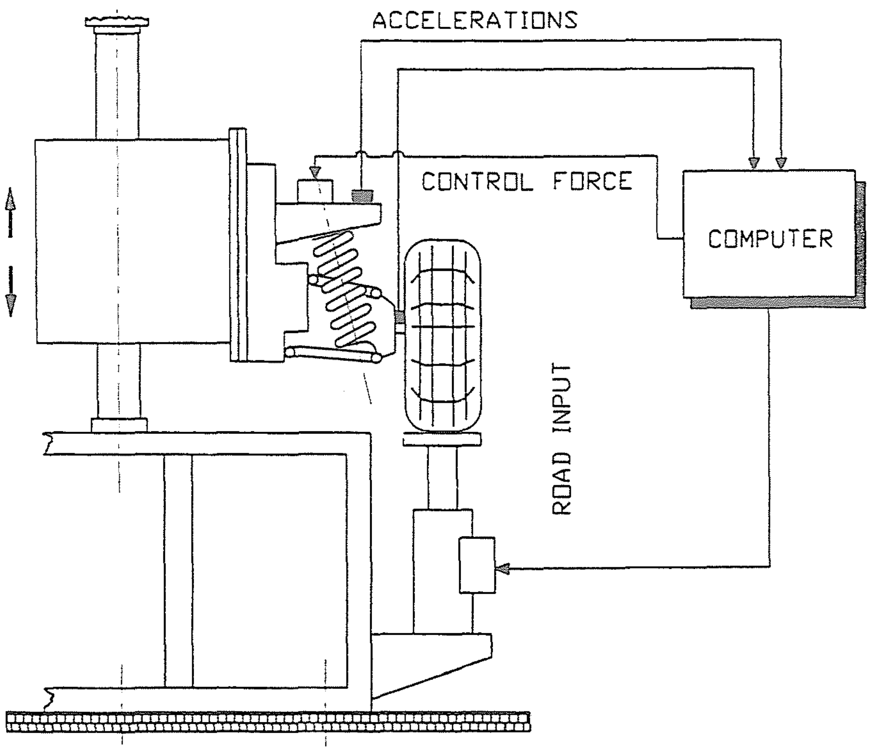

Fig. 5. Quarter-car model test rig

The vehicle body is represented by a vertically moving mass, onto which a two-arm front suspension is attached, keeping the original geometry of the suspension. The linear bearing needed a special attention because of the friction problems. The road excitation is generated by a Rexroth servo cylinder with maximum capacity of 25 kN (compensated for 15 kN lateral force) and bandwidth of 60 Hz. As active element, a 10 kN, 70 Hz servo cylinder is used, especially manufactured for this application and fitted into the suspension's coil spring replacing the shock absorber. Two accelerations are measured, one is on the body, the other on the lower wishbone. The system will be controlled by a PC based DSP board.

Although the results obtained from this investigation are promising, there are several factors putting strong limitation on the applicability of the fully active suspension:

- the cost of the active suspension's components (valves, cylinders, power supply, etc.) is very high, and makes it very expensive for the customer.
- the energy demand of the system (producing large oil flow and pressure) is very high and should be generated by the car engine, which results in a loss of the accelerating performance, decrease of the maximum speed.

These two facts explain that not too many car producers are dealing with active suspension development for serial production. However, there are some special vehicles where their application is feasible: transporting expensive goods (e.g. vibration sensitive instruments, computers) or systems (for example military command points with high speed).

3.2. Investigation of Semi-Active Systems

As it was pointed out at the end of the last paragraph, the active suspension system is not really feasible from point of view of the high energy demand. This problem can be eliminated by replacing the fully active actuator by a semi-active one. The semi-active actuator is a variable damping-coefficient shock absorber, which is able to vary its damping ratio with high bandwidth. The semi-active damper is able to dissipate energy only, but in a controlled way. The system's model is similar to the active suspension shown in *Fig. 3*, only the actuator is replaced by a controllable damper, and the control strategy should be modified. The derivation of the continuous as well the discrete semi-active control strategy is shown in *Fig. 6*.

As seen in *Fig. 6.a*, with the active control any relative velocity between the vehicle body and wheel can be controlled. The ideal semi-active damper should be able to generate zero and infinite damping coefficient, but only energy can be dissipated (*Fig. 6.b*). Since neither zero nor infinitely large damping ratio can be obtained, the control range in the 1st and 3rd quarter is narrowed (*Fig. 6.c*). The continuous system operates fine, but this is more

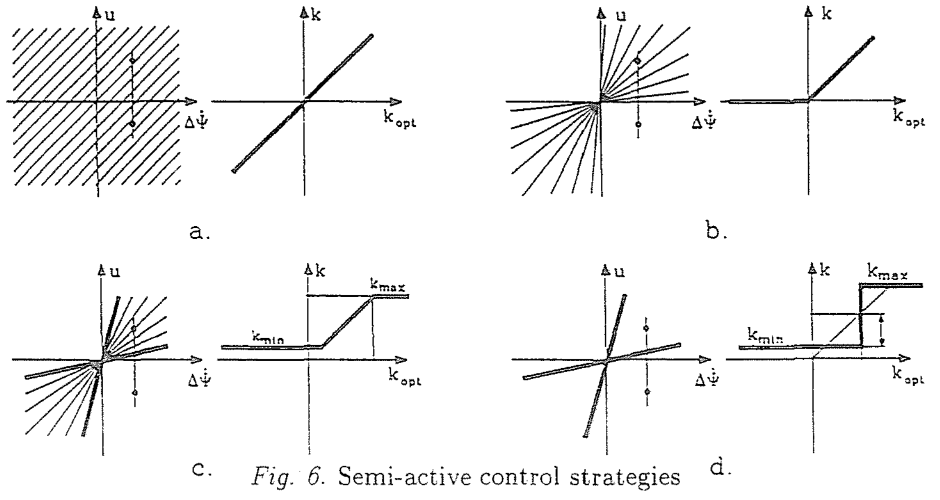

Fig. 6. Semi-active control strategies

expensive, since it needs more complicated hardware. That is why usually a two, or more stage damper is for semi-active suspension control. In *Fig. 6.d* a two-stage semi-active damper control logic is shown.

The semi-active suspension control project was conducted together with the Vehicle Research Laboratory of the Delft University of Technology, the Netherlands. Using the vehicle, fitted with controllable dampers at the rear, some of the measured results can be seen in *Fig. 7.* The figure shows measurements at the low-damper stage (about 300 Ns/m), high damper stage (about 900 Ns/m) and with the semi-active damper. As seen in the upper chart of *Fig. 7*, with low setting the peaks at about 1 Hz and 11 are high, the low damping results in large amplitude at the eigenfrequencies of the vehicle, while in the frequency range of 4–9 Hz the gain is low. With high setting it is in opposite, the peaks are eliminated, but there is large gain in frequency range of 4–8 Hz, where the human body is more sensitive to the vibration. With semi-active damper control (SAD) the optimum between these two limits can be found. However, two problems can be detected: (a) the improvement is not too large, which is caused by the small difference between the low and high damper settings, and (b) undesired oscillatory motion can be seen in higher frequency ranges. This last phenomenon, referred as chattering, is inherent problem of variable damper systems. Some solution to eliminate this vibration can be found in [2]–[11] using sliding mode control.

However, the semi-active suspension system is getting more widely used, all the leading car manufacturers offer this system for the products.

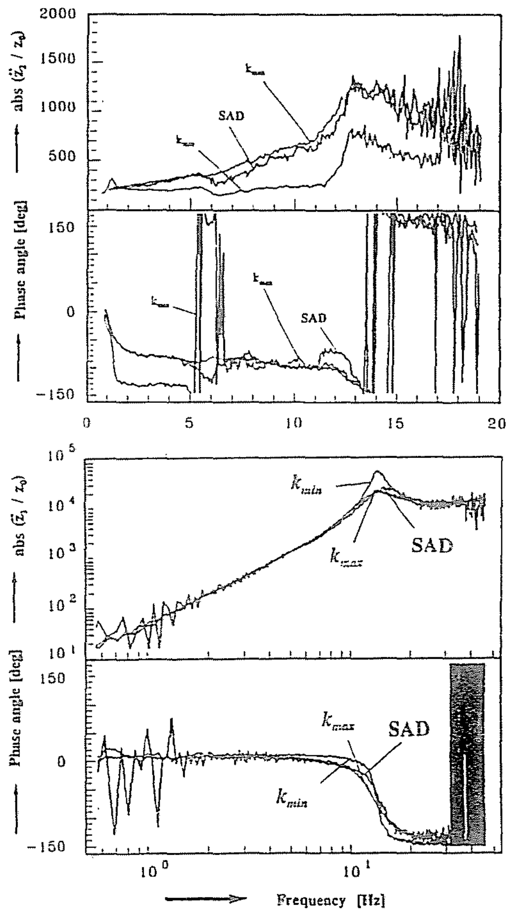

Fig. 7. Transfer functions between the body and wheel accelerations and the road excitation (measured results)

3.3. Controlled Seat Suspension Design

In heavy vehicles primary suspension controlled elements usually cannot be applied, because of high loads, which should be carried by the controlled actuators. The modification of the suspension itself is sometimes also not possible. However, the heavy truck drivers are subjected to a logn term vibration, whose effect is prolonged during their lifetime and might result in serious illness requiring millions for medical treatment, hospitalization, rehabilitation, etc. Similar problem can be found in other type of vehicles, such as agricultural tractors, mining vehicles, where the primary suspension is very stiff (if exists at all).

In this case the driver's seat suspension can be modified to give better isolation fot the driver. This type of activity has been started recently, and some very first results have already been published (see in [2, 10]). *Fig. 8a* shows the model of a combined human-driver and quarter-vehicle.

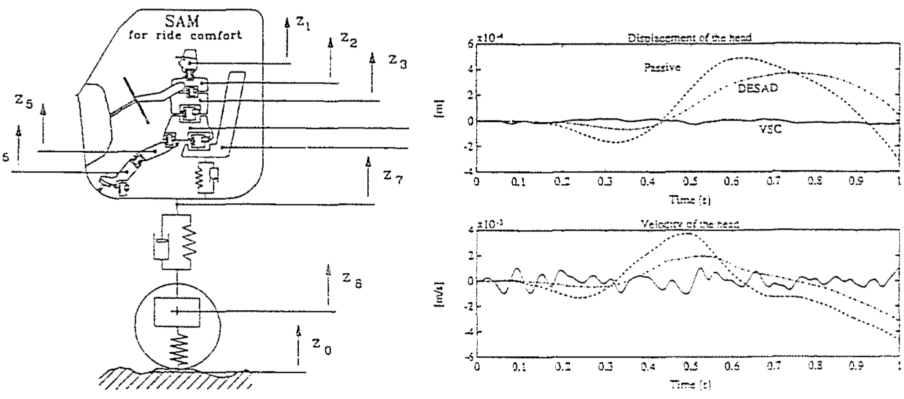

Fig. 8. (a) Combined model of a human driver and vehicle
(b) Some results with sky-hook controller

In this case similar control problem is solved as introduced before. The driver's seat is suspended by using a semi-active damper. The control strategy is designed based on the so-called 'sky-hook' principle, which means that the damper's switching algorithm is working similar to a damper installed between the seat and an inertial reference point (the 'sky'). When the required sky-hook force is having the same sign as the one can be produced by the shock absorber, the high stage is set, when the sign is opposite, the low stage is set. By applying this control strategy, some of the results can be seen in *Fig. 8.b.*

The results prove the applicability of this procedure. Recently, a practical realization of this theoretical investigation has been started. A heavy truck seat's suspension will be modified, applying the same actuator as for the active suspension model, and another servo cylinder is used to generate the vehicle floor excitation, based on a real vehicle's motion on a given road profile.

The driver seat suspension control is not a new topic for investigation, some adaptive systems can also be bought. These seats usually have pneumatic suspension, which varies the internal air-pressure according to the driver's weight, the road roughness, etc.

4. Stability Control

The other field of control application investigated by TUB and SZTAKI is vehicle lateral control, although beyond a certain point (integration of different control systems in order to achieve better performance, optimal sensoring) they cannot be separated. Three different fields are being investigated here: the potentials of the four-wheel steering (4WS), controlled anti-jackknifing systems, and stability enhancement control for combination vehicles (such as car–caravan combinations, articulated buses, and heavy vehicle configurations).

4.1. Four-Wheel Steering Systems

The control logic of the 4WS system can be drawn into the block scheme in *Fig. 2*. As seen in *Fig. 9*, the external inputs to the vehicle system are the driver's steering angle, expressing his intention, and a side disturbance, arising from side wind, for example. Control goal of the 4WS steering system is twofolded: at low speed to increase the manoeuvrability of the vehicle by steering the rear wheels in opposite to the front ones, while at higher speeds to decrease the vehicle lateral acceleration and yaw rate, avoiding the sliding or spinning-out of the vehicle. The control scheme for this latter case is shown in *Fig. 9*.

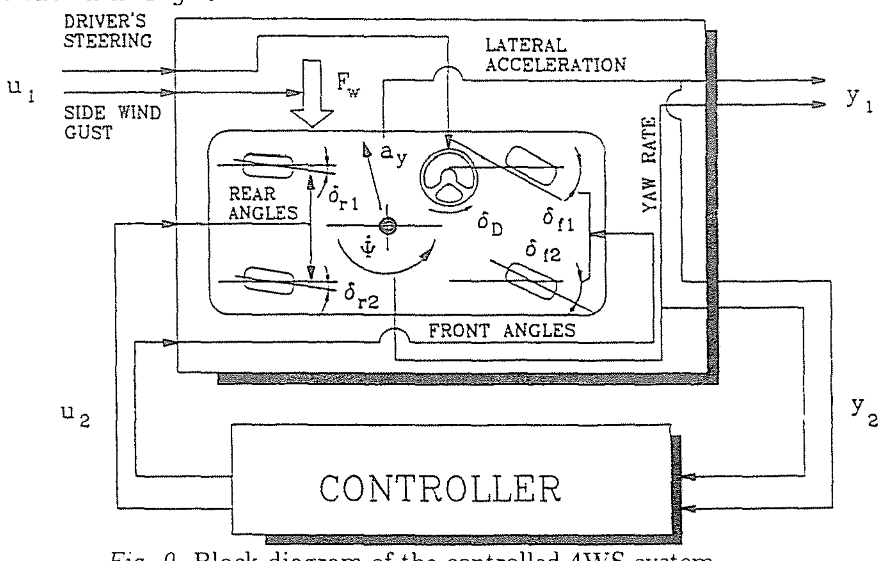

Fig. 9. Block diagram of the controlled 4WS system

The controller design procedure is not as obvious as in the suspension case, since it is not possible to compensate for any perturbation. Although the driver's steering angle is shown as external disturbance, the controller does not have to attenuate it. There exist several control implementations as well: (a) the one shown in *Fig. 9* is called autonomous control (see in [5]), where the steering wheel is only a sensor, but it is not linked directly to the wheels (b) only the rear wheels are steered only by actuators, the front ones are steered directly by the driver (c) so called additional steering, where the rear wheels are steered actively, the front ones are mechanically, but some correction is possible by actuators.

Additional advantage of the active 4WS systems is the ability of compensating for the lateral disturbances, such as a sudden side wind gust. In this case, possibly without any effort of the driver, the disturbance can be attenuated by only steering of the rear wheels, as shown in *Fig. 10.* This figure shows also the sensitivity of the 4WS system for some physical vehicle parameters (rear axle cornering stiffness). This explains the need for robust control design which was investigated in reference [1].

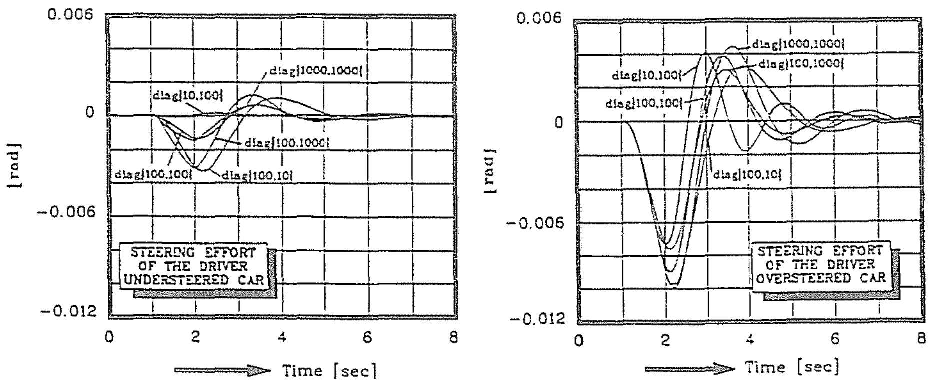

Fig. 10. Compensation for side wind disturbance (a) with the nominal model (b) with uncertain model

4.2. Jackknifing Control in Combination Vehicles

The stability problems of the combination (or articulated) vehicles differs from that of the single vehicles. There are some typical forms of loosing stability for articulated vehicles: roll-overn jackknifing and the self-excited oscillation of the towed vehicle unit. The problem of jackknifing can be found in both tractor–semi-trailer configurations and articulated buses. This is very important consideration in the pusher types (whereas the engine is located at the second unit, and the third axle is driven), since the traction force produced by the tires on the third axle is transmitted to the front through the fifth-wheel. When the front and rear units' angle is larger, there is a considerable force acting on the hitch in lateral direction (see *Fig. 11*), which pushes the second axle out, leading to jackknifing. In this type of the vehicle this is normal phenomenon, which should be hindered. However, jackknifing can occur in tractor–semi-trailer vehicles especially when braking on slippery road. To avoid the jackknifing in this case the ABS helps a lot until a certain point, but the physical limits cannot be overcome.

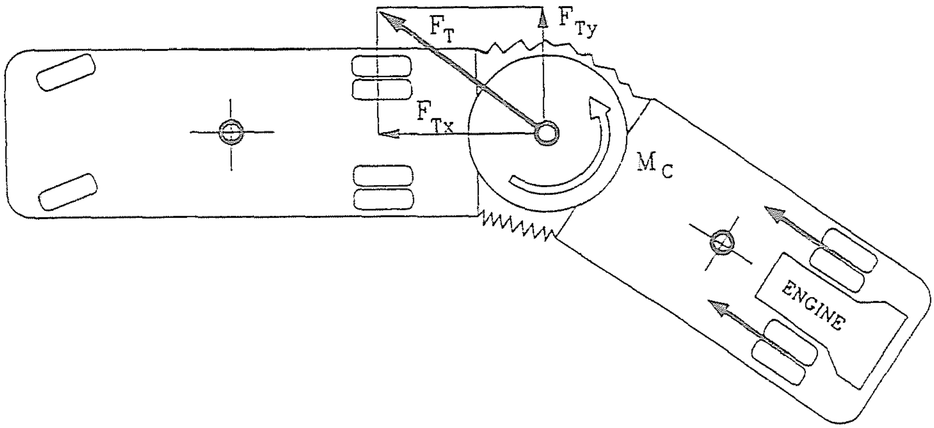

Fig. 11. Hitch moment control for articulated bus of pusher type

The key of controlling such a motion in combination vehicles is the control of the torque M_C in *Fig. 11*. In articulated buses, where the turntable provides enough room, either controllable dampers or large diameter disk brakes are fitted. Both these strategies are semi-active since none of them is able to introduce additional energy, only dissipate it. Beyond a certain hitch angle, both systems are locking and do not allow any larger angle, hindering the jackknifing.

For a tractor–semi-trailer vehicle configuration similar system was investigated in [3]. As was shown, the semi-active hitch torque control has shown improvement in the oscillatory behaviour of the trailer, but it was not able to control the vehicle's roll motion. Furthermore, for tractor–semi-trailer it needs additional devices to be installed making the vehicle price higher at lower benefits.

4.3. Stability Enhancement Control (SEC) of Combination Vehicles

The combination vehicles' stability can considerably be improved by generating a stabilizing yaw torque on the towing vehicle unit. This torque could only be produced by operating the vehicle's brake on one side or on the other. The main advantage of this system is that does not need any additional actuators and sensors, since the brakes are already installed, yaw rate, acceleration sensors will be a part of the new generation on-board computers installed on the ECU card. This system is described in [3], [7], [8], [13]. *Fig. 12* shows a car–caravan combination with SEC.

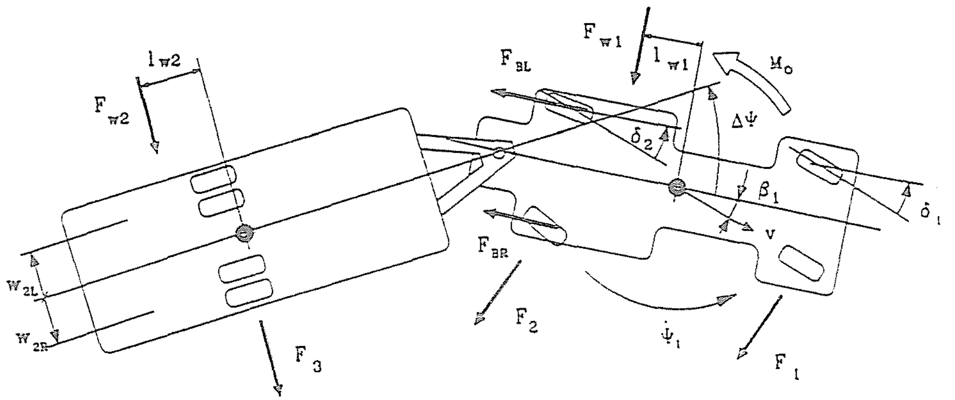

Fig. 12. Car–caravan combination with brake controller

The system has two operation modes:

Mode #1: In this mode the controller is working autonomously, independent of the driver's intention. When the combination of the vehicle state variables exceeds a predetermined (or adaptive) limit and the vehicle's motion starts to be irregular (spinning out, tends to jackknife, etc.) the controller automatically applies the appropriate side brake on both axles at passenger car, and on the rear axle on heavy commercial vehicles. The generated moment stabilizes the vehicle's motion.

Mode #2: This mode operates when the vehicle is being braked. The system is also able to release the brakes on the appropriate side. In this mode the controller coordinates the brake forces and produces stabilizing torque.

Fig. 13 shows the behaviour of a car–caravan combination (a) without any control, (b) with ABS only, (c) with ABS+SEC, (d) ABS+4WS. As seen, the ABS+SEC provides the best stability during this combined braking and turning manoeuvre (braking from 65 km/h on a road surface with $\mu = 0.4$) *Fig. 14* depicts the same four vehicles at μ-split test (the left hand side is $\mu = 0.9$, the right side is $\mu = 0.3$, braking from 80 km/h). The ABS–SEC vehicle goes on a straight line, the other three spin out in different extent.

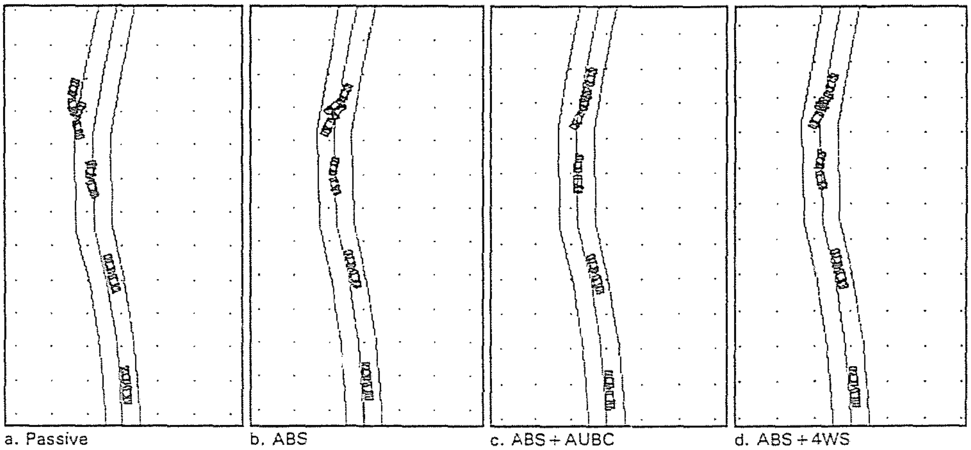

Fig. 13. Top views of the combined braking and turning manoeuvre

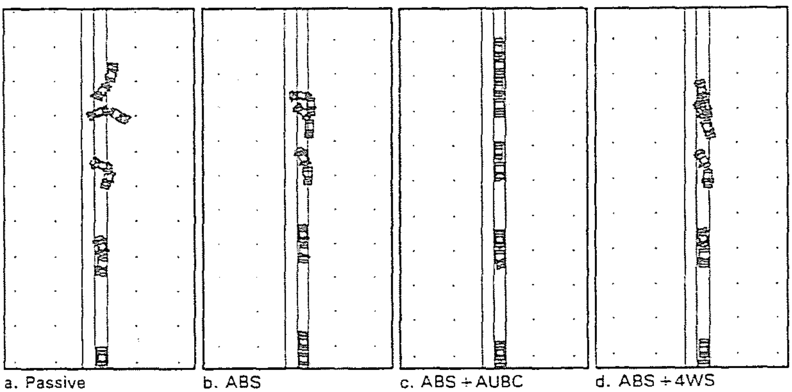

Fig. 14. Hard copies of the μ-split manoeuvres

5. Summary

In this state-of-art report the research activity of the Systems and Control Laboratory of the Computer and Automation Institute of the Hungarian Academy of Sciences and the Department of Road Vehicles, TUB conducted in the field of controlled vehicle dynamics. In the present state of the research the on-vehicle systems are emphasized, but the first steps are already done in the direction of the investigations into intelligent highway-vehicle systems as well.

Acknowledgement

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