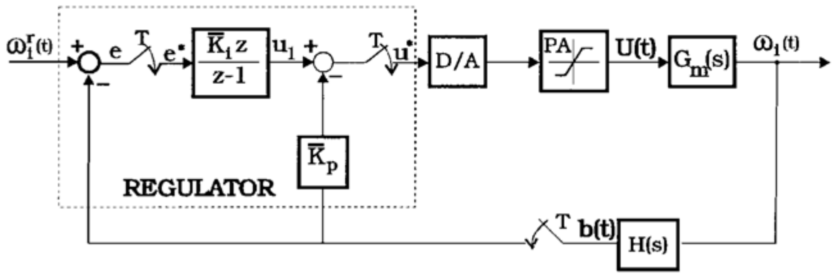

Fig. 4. Microprocessor control system structure of a dc motor speed control.

These coefficients depend on the fixed system parameters and on the gains of p- and i-actions, which can be tuned in such way, that the system acquires the desired dynamic characteristics.

IV. Parameters Estimation

The unknown model parameters used for the control synthesis should be determined by on-line estimation during the ride. Beside geometrical vehicle parameters, determined by design, practically all other parameters are time-variable. It would be quite complicated to estimate all system parameters, since the estimation is to be performed in real time. Hence, it is indispensable to know which of the vehicle dynamic parameters and the parameters of the interaction of the tire pneumatics and the road surface change more significantly, and in what range of values. First, those are the *vehicle mass* $\hat{m}$ and *moments of inertia* about its main axes $\hat{I}_x$, $\hat{I}_y$, and $\hat{I}_z$. Parameters of the suspension system and the tire pneumatics parameters are also changing depending on the temperature, fluid viscosity in the hydrocylinders, tire pressure, etc. Changes of these parameters during motion are relatively small under the condition that the vehicle is in good working order. Their influence on the dynamic system behavior, compared with the effects caused by the change of vehicle mass and inertia is practically negligible. For that reason, they can conditionally be considered as constant values. The parameters of the road surface state, i.e., the parameters of interaction between the wheels and the road, are very influential on the dynamics of vehicle motion and hereby can be significantly varying. This refers, before all, to the *sliding friction coefficient* $\hat{\nu}_{di}$ ($i = 1, \cdots, 4$) and the *rolling resistance coefficient* $\hat{f}_{ri}$ ($i = 1, \cdots, 4$). Also, a side elevation angle of road surface is of variable magnitude which can influence a lot the lateral and longitudinal dynamics of the vehicle motion. However, the standards of highway construction, especially if it is to be used for automated traffic regime prescribe zero or very small and slow varying changes of the road surface elevation. In [7] and [12] it has been shown that the information about the forthcoming road geometry, under the condition that it is known, can be sent to the vehicle, either if it is measured, or if it is known and guaranteed in advance by the construction standards.

For estimation of the stated, unknown model parameters ($\hat{m}$, $\hat{I}_x$, $\hat{I}_y$, $\hat{I}_z$, $\hat{\nu}_{di}$, $\hat{f}_{ri}$), we have used the *recursive least square estimation method*. Hereby we have used the vector dynamic model of the vehicle (1) and (2), which can be expanded into six scalar differential equations of behavior. Also, in order to estimate the road-tire interaction parameters, we have used the equations describing the tire pneumatic model, given by relations (5)–(6) and (7)–(8). Small variations of other, nonestimated parameters, can be treated as internal system disturbances, which have not influenced the system motion dynamics significantly.

V. Simulation Experiments

For the sake of the synthesized controller analysis, control scheme of which has been shown in Fig. 2, a model of the road vehicle has been simulated with geometrical and dynamical parameters given in [7]. The nominal, curvilinear vehicle trajectory used in simulation is presented in Fig. 5. Vehicle was tracking the desired path by a given constant speed of $V = 80$ [km/h]. The influence of the external disturbances, acting on the vehicle during motion, were simulated. These are: influence of the road surface profile variation $z_r(t)$, influence of the variation of a road surface side elevation angle $\gamma(t)$, influence of a sudden change of the tire sliding coefficient $\nu_d(t)$ of the wheel running over a slippery surface and the influence of a sudden side wind gust of variable intensity $F_{\text{wind}}(t)$. The model of road surface profile was given as a random function with a maximum amplitude $z_r^{\max} = 1$ [cm] with frequency $f_{\text{road}} = 22$ [Hz]. Hereby, it was assumed that the front and rear wheel pairs pass the same paths and that the same perturbation signals appear on front and rear wheels. They are time-shifted by the amount of time constant, determined on the basis of the vehicle motion speed V. A small, sinusoidal variation of side elevation angle of road surface was also given as $\gamma(t) = 2 \sin \omega_\gamma t$ [deg]. Here, ω_γ is the angular frequency of the angle amplitude variation, calculated from the condition that the side elevation angle is periodically changing at each 2 [km]. Changes of the vehicle dynamic parameters m, I_x, I_y, and I_z of ±25% during motion were allowed, as a consequence of fuel consumption and load rearrangement due to passengers dislocation in the vehicle. Estimation of the unknown values of the model parameters $\hat{m}$, $\hat{I}_x$, $\hat{I}_y$, and $\hat{I}_z$ was carried out during simulation. The efficiency of the controller presented by control schemes in Fig. 2, was analyzed based on the results of simulation obtained for three characteristic cases: 1) *appearance of a variable sliding*

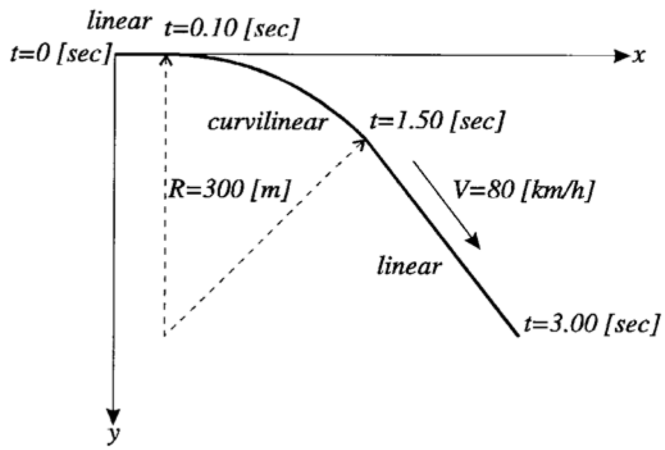

Fig. 5. Nominal vehicle trajectory.

friction coefficient; 2) *appearance of a sudden side wind gust*; and 3) *case of vehicle cornering with* 25% *higher speed along the same trajectory*. All three cases were simulated under the same initial conditions of motion along prescribed trajectory. Hereby it was assumed that the vehicle deviates from the prescribed trajectory in the initial instant, e.g., it is late in the longitudinal direction by $\Delta x = -0.20$ [m], has a lateral position error of $\Delta y = -0.10$ [m] and has an error of relative vertical position $\Delta z = 5$ [mm] of the vehicle MC, with respect to the road surface. Errors of initial orientation angles, from their equilibrium positions, respectively, were also imposed to be: roll $\Delta\Phi = +0.01$ [rad], pitch $\Delta\theta = -0.01$ [rad], and yaw $\Delta\varepsilon = -0.03$ [rad]. In case 1) it was prescribed that during vehicle cornering it encounters a slippery road, whereby the sliding dynamic friction coefficient reduces radically from $\nu_d = 0.7$ to $\nu_d = 0.21$, in time interval of motion from 0.5 to 1.0 [s] from the simulation initial moment. New value of the unknown sliding, coefficient $\hat{\nu}_d$ was determined by on-line estimation, in a way, briefly explained in the previous section of the paper. In case 2), the influence of a sudden, side wind gust of constant direction ($\psi_{\text{wind}} = 60$ [deg]), clockwise with respect to the x-motion direction was simulated. Hereby, it was prescribed that the wind intensity decreases according to the law $F_{\text{wind}}(t) = 5000 \exp^{-\lambda\Delta t}$ [N], in the interval $\Delta t = 0.3$–1.3 [s]. The wind "weakening" parameter λ was calculated from the condition that the intensity of wind gust reduces to $F_{\text{wind}}(t) = 0.1$ [N] in Δt [s]. Vehicle motion was simulated upon a dry road (with constant value $\nu_d = 0.7$) and without wind gusts, in the case 3). Hereby, vehicle motion was prescribed along the same trajectory, with constant speed of $V = 100$ [km/h] which is 25% higher than the nominal. An idea was to demonstrate that by using the proposed controller, the same vehicle trajectory can be "mastered" at higher speed without loosing motion stability or reducing ride comfort.

The controller, based on relation (11), was synthesized, choosing the gains in such a way that the control system stays stable and possesses the desired dynamic behavior. The gain matrices K_P and K_V have been chosen as quadratic, diagonal 6×6 matrices, with diagonal elements: $k_P^1 = k_P^2 = k_P^6 = 9.8696$, $k_P^3 = k_P^4 = k_P^5 = 355.3058$ [s^{-2}], and $k_V^1 = k_V^2 = k_V^6 = 6.2832$, $k_V^3 = k_V^4 = k_V^5 = 37.6991$ [s^{-1}].

It should be emphasized that up to now vehicle integrated control was treated almost exclusively through the combined action of the driver and the individual control modules (partial autopilot control). For that reason, a valuable reference did not exist, based on which we could use our integrated dynamic control for comparison with a similar one. We are also of the opinion that even a skilled driver cannot control in explicitly successful way the system behavior in all directions (translatory-angular), which the automatic system is capable of in any case. For that reason, the paper gives the illustrations that point to the doubtless advantage of the autopilot for control of those coordinates, along which the driver is not capable of improving the dynamic vehicle behavior in special driving conditions.

The results of model simulation are presented graphically for three previously mentioned cases in Figs. 6(a), 6(b), 7, and 8. In Fig. 6(a) the quality of realized motion velocity along desired trajectory subjected to perturbations of various nature and intensity is illustrated. The efficiency of the proposed control scheme (Fig. 2) was analyzed based on the simulation results [Fig. 6(a) and (b)] obtained for the case of vehicle motion over the dry road, slippery road, dry road with side wind gust, and over the dry road with 25% augmented forward velocity. Looking at Fig. 6(a) it can be noticed that the synthesized controller enables satisfactory "velocity" tracking of prescribed trajectory, independent of nature of perturbation acting on the system. In Fig. 6(b) positional accuracy of trajectory tracking in longitudinal, lateral, and yaw directions for all four mentioned cases is presented. By comparison of obtained plots it can be concluded that the synthesized controller ensures the satisfactory accuracy of nominal trajectory realization. The proposed autopilot control scheme eliminates the initial position errors and guarantees the satisfactory dynamic behavior of road vehicle during the motion. It can be noticed that for the chosen control gains set, K_P and K_V, the dynamic vehicle behavior is not the same for the different perturbations types. So, the nonadaptable increasing velocity on the slippery road more significantly deteriorate the quality of trajectory tracking as compared to the influence of the wind gust of moderate intensity. The influence of a slippery road and increasing of forward velocity during cornering, particularly deteriorate the system behavior in lateral and yaw directions. Besides, the synthesized controller ensures the ride stability [Fig. 6(b)] with the remark that the controller is realizable only for the case when the system parameters are estimated with high-degree accuracy.

Concerning the precision of vehicle body position in vertical direction and precision of chassis pitch and roll angles, the following facts can be concluded. By comparing the results given in Fig. 7 the transient regimes of motion of three state values z, Φ, and θ are essentially worse in the case with no automatic control. The indicators about the errors $\Delta z(t)$, $\Delta\Phi(t)$, $\Delta\theta(t)$, as well as the suspension system deflections for instance on the first wheel, are presented in

(a) (b)

Fig. 6. Simulation results: (a) Vehicle forward velocity during motion under various disturbances. (b) Accuracy indexes for programmed trajectory tracking in longitudinal, lateral, and yaw direction.

the Fig. 7. Concerning the estimation of ride comfort quality, indexes of the vehicle dynamic behavior in the direction normal to the road surface were analyzed. Aiming at that, the so-called *average square relative error* of the observed variable was introduced expressed in percents. Let us denote by ξ_i some of the following magnitudes: P_i, F_{zi}, e_i, and e_{ti}. They are denoting successively the vertical payload of the ith joint of the SS, the vertical tire load of the ith wheel, suspension system deflection and tire pneumatic deflection. Then the average relative square error is defined

Fig. 7. Simulation results: accuracy indexes of vertical position, roll, and pitch angles; amplitudes of suspension system deflection.

TABLE I
PRESENTATION OF THE AVERAGE RELATIVE SQUARE DEVIATIONS OF FORCES P_i, F_{zi}, AND DEFORMATIONS e_i AND e_{ti} FROM THEIR NOMINAL VALUES

	tire	σ_{P_i} [%]		σ_{Fz_i} [%]		σ_{e_i} [%]		$\sigma_{e_{ti}}$ [%]	
without control	1st 2nd	5.7482	5.7371	4.5536	4.5659	0.5618	0.4182	1.2428	1.3190
	3rd 4th	6.9911	7.2442	5.5632	5.8477	0.4442	0.4982	1.8607	1.9220
with control	1st 2nd	0.1450	0.0694	0.7735	0.5516	0.5396	0.3588	0.2294	0.1087
	3rd 4th	0.0201	0.0687	0.8324	1.2218	0.4641	0.6551	0.1450	0.2013

as

$$\sigma_{\xi_i} = \frac{\sum_{j=1}^{N}\left(\frac{\xi_i^j - \xi_{i0}}{\xi_{i0}}\right)^2}{N} \times 100[\%] \tag{22}$$

where ξ_{io}, is the nominal value of ξ_i and N is the number of samples. The values of the indexes σ_{P_i}, $\sigma_{F_{zi}}$, σ_{e_i}, and $\sigma_{e_{ti}}$ are systematized in Table I, and given for the example of motion on uneven road surface with and without active suspension on the wheels. These indexes represent measures of force and acceleration amplitudes values in the vertical direction, as well as measures of the suspension system and tire pneumatics deflections. By comparing the values in Table I, it can be clearly noted that the effects of bobbing, due to ride over uneven road surface, are many times lower for the case of automatic suspension control, than for the case of a passive suspension system. Therefore the ride comfort has been significantly improved. Vertical vibrations (accelerations) were decreased by action of the active damping forces P_{di} in the viscous cylinders of the VSS. Using control scheme from Fig. 2, relatively small tire deflections $e_{ti} = z_t^i - z_r^i$ (see Fig. 1) were also ensured. We note that in the simulation we have not performed the signal prediction of road surface variations.

Fig. 8. Output signals of vehicle actuators for the case of a slippery road.

By using the block for the calculation of uniform distribution of control actions for all wheels (14), the proposed control scheme guarantees the minimum variations of control variables $\delta(t)$, $\omega(t)$, and $P_d(t)$. In this way less energy consumption, more uniform dynamic loads of system elements, as well as better vehicle maneuverability are achieved. The above mentioned control variables, realized at system actuator outputs are presented in Fig. 8, for the case of uneven and partially slippery road. To conclude, by analyzing the obtained results, it becomes evident that the proposed controller essentially improves motion stability and the dynamic system performance under perturbations of various nature and intensity.

VI. Summary and Conclusions

A dynamic controller of a road vehicle was presented in this paper. Control strategy is based on the knowledge of the entire nonlinear system model. The unknown model parameters were estimated on the basis of measured values on the wheels, suspension system, and on the vehicle chassis. The control strategy is based on the centralized hierarchical principle of distributed control, in the way that the centralized control signals were distributed to the actuators of the particular dynamic subsystems. The obtained simulation results indicate the expected advantages of such control type of the vehicle as a multibody dynamic system. It should be stressed out that a conditionally complete dynamic system model has been used in this paper. Due to the impossibility of an experimental verification of the results of the synthesized controller implementation at this moment, the obtained results should be considered critically. These are, before all:

- *Issue of sufficient accuracy of the used dynamic system mode.* The nonlinear system model which was implemented within the proposed control scheme, represents a model describing system behavior with a broad range of input variables and perturbation magnitudes. Delays in the servo actuators were not taken care of in this model. The internal perturbation magnitudes represent the physical time delay in the actuator cylinders. To a certain degree this delay influences the system behavior in a bad way. These and similar effects can be avoided by using the combination of conventional and unconventional methods of control, before all by the synthesis of neural or fuzzy-controllers, which should compensate for the unmodeled dynamics of actuators lags, as well as other system uncertainties.
- *Issue of estimation accuracy* of the vehicle dynamic parameters and the estimation of tire-road interaction

parameters depends, to a large extent, on the equipment at our disposal, which is necessary for the measurement of the model state values. Some of the input values, like road surface elevation (longitudinal and lateral), are unmeasurable, or very difficult to measure from the vehicle. This information, as well as data about the road geometry, have to be sent to the controller from the "outer world." Here, the problem of measured signals filtering is also imposed, notably of the measured accelerations and forces, contaminated by unuseful information.

- *Problem of prediction of perturbation signals* acting on the system. It is well known that external and internal perturbations of various nature can be acting on the system. Some external perturbations can be measured in advance, before they start acting on the system. Prediction of such signals can be done with the signal of road surface profile variation.

The control laws synthesis based on the integral knowledge of vehicle dynamics is capable to ensure the desired system performances to some extent. However, it is possible to improve the control quality by corrections of different subsystems models by using some experimental results. For instance, instead of nonlinear tire model (5)–(8) used in the synthesis of the controller presented in this paper, some other, e.g., empirical tire model based on experimental data [20], can be used. Beside that, there is a possibility of combining the proposed *model-based control* synthesis with some of existing *knowledge-based control* techniques, by means of which the model inaccuracies and parameters uncertainty would be practically compensated.

In the future, the investigations of practical stability of the decentralized dynamic control scheme, as well as the synthesis of neural and fuzzy compensators will be performed.

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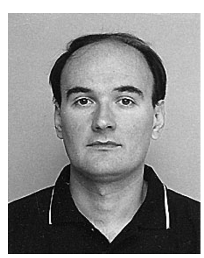

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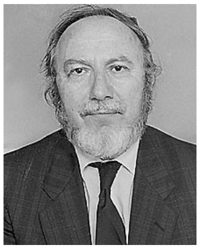

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