Research Article

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Modelling of the system “driver - automation - autonomous vehicle - road”

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Abstract: The work presents a simulation model of a “driver–automation–autonomous vehicles–road” system which is the basis for synthesis of automatic gear shift control system. The mathematical description makes use of physical quantities which characterise driving torque transformation from the combustion engine to the car driven wheels. The basic components of the model are algorithms for the driver’s action logic in controlling motion velocity, logic of gear shift control functioning regarding direction and moment of switching, for determining right-hand side of differential equations and for motion quality indicators. The model is realised in a form of an application software package, comprising sub-programmes for input data, for computerised motion simulation of cars with mechanical and hydro-mechanical – automatically controlled – transmission systems and for models of characteristic car routes.

Keywords: autonomous vehicle, automatic gearing control system, mechanical and hydro-mechanical transmission, computer simulation of motion, control algorithms

1 Introduction

The vehicle which does not require the participation of a driver in controlling it in road traffic is called the autonomous vehicle. Gathering information about traffic conditions and the behaviour of other traffic participants, which are determined by the rules of law, is the responsibility of control systems generating relevant control algorithms and start appropriate elements and executive systems [1–3]. It also pertains to controlling the driving system of autonomous vehicles.

The system which gathers and processes information about the state and the traffic mode of the vehicle chooses the direction and the moments of gear switches, controls the engine, the engine brake, the clutch or the blockade of the torque converter and the gearbox in the process of switching, chooses and activates the demanded gear after braking or coasting, is referred to as the automatic system of controlling gear change [4–7].

In designing such a system it is necessary to choose its information parameters which will determine the rules of gear change. They determine the moments of switching to the neighbouring gear depending on the component values of the vector of information parameters [8–11].

The problems of choosing the elements of information parameters vector and the rules of gear change are solved by the joint usage of simulation and optimization models. From the mathematical point of view these tasks are included in the class of mathematical programming problems, in which a number of restrictions is algorithmically assumed by means of a simulation model [12–15].

Scientific publications address the problems of developing simulation and optimization models describing automatic control of car drive systems [16–18]. They concern drive systems with both mechanical and hydro-mechanical transmission. However, it seems justified to work out a simulation model with a clear representation of physical quantities which characterise driving torque transformation from the combustion engine to the car driven wheels. This will make the analysed field of scientific research into automation of car drive systems more complete.

Such a simulation model which is the base of solving the aforementioned problems is the model of the “driver-automation-vehicle-road” system. The model must meet the following requirements for cars with mechanical gearbox:

1. enabling the work of the engine on outside and partial characteristics of the load [19–21];
2. simulating the actions of the driver by controlling the speed of the vehicle and ensuring the following driving modes: bringing up to speed; driving at permissible speed; coasting, braking with the engine; braking with the brakes and the engine simultaneously [22, 23];

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3. synthesizing diversified variants of the rules of gear change with the use of operational and corrective combinations of information parameters [24, 25];
4. ensuring switching to higher and lower gears in compliance with the rules assumed in the system controlling gear switch automatically [26–28];
5. realizing the algorithm of the logic of the automatic gear shift control functioning;
6. simulating the motion of the car on the road with changeable profile and with different kinds of road surface and with speed limits on particular sections depending on safety conditions or the speed of transport stream motion;
7. variables which will characterize car motion, as well as its average speed and fuel consumption on the route should be the result of solving differential equations describing the behaviour of the model [29, 30].

In the case of modelling the motion of the car equipped with automatic hydro-mechanical transmission, the above mentioned requirements should be supplemented with:

8. making it possible to model the process of switching gears with the stream of power flow both intermittent and continuous [31–33];
9. ensuring automatic blocking and unblocking of the hydrokinetic torque converter according to the set rules of switching gears, as well as unblocking the hydrokinetic torque converter during gear change [13, 34, 35].

In scientific literature one can come across models of car motion which serve as the basis for the synthesis of automatic gear switch systems or optimization undertakings *e.g.* [2, 14, 36]. However, not always do they contain mathematical descriptions of applied algorithms.

2 Mechanical drive system of the car

The main components of the simulation model of the "driver-automation-vehicle-road" system are algorithms for calculating the right-hand sides of differential equations. They describe the behaviour of the car during different driving modes and determine the indicators of its motion quality. There are also algorithms of the logic of the driver's actions in controlling the speed of the ride and the logic of the functioning of automatic gear switching system in terms of selecting the direction and the moment of gear change.

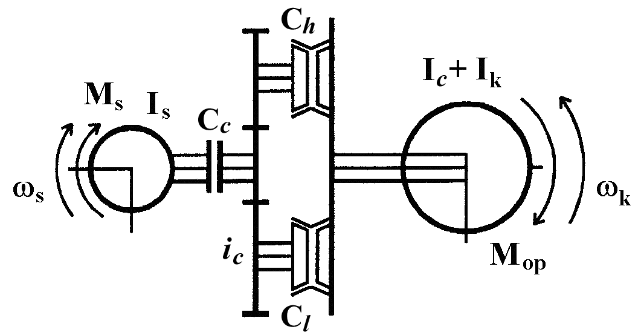

Figure 1: Dynamic scheme of the car system with mechanical transmission [own study]

Dynamic scheme of the car system with mechanical transmission applied for the synthesis of the simulation model of the "driver-automation-vehicle-road" system is presented in Figure 1.
The following symbols were introduced:
I_s, I_c, I_k - respectively: moments of inertia of the engine flywheel together with its rotating parts; moment of inertia equivalent to the car mass in translatory motion; moment of inertia of car wheels;
C_c - clutch; C_h and C_l - synchronizers of activating higher and lower gear, respectively, in the gearbox;
ω_s, ω_k - angle speeds of the engine flywheel and masses with moments of inertia, respectively I_c and I_k; M_s and M_{op} - engine torque and the moment of car resistance to motion; i_c - total ratio of the vehicle drive system.

The clutch C_c and the synchronizers C_h or C_l are usually closed, except for the cases when the car is coasting or gear change is being performed in the box.

The movement of dynamic system masses of the car with mechanical transmission (Figure 1) reduced to wheels driven at closed C_c, C_h or C_l is described by the following equation:

$$(I_s \cdot \eta_m \cdot i_c^2 + I_c + I_k) \cdot \frac{d\omega_k}{dt} = M_k + M_s, \tag{1}$$

where M_k - torque on driving wheels of the car; η_m - mechanical efficiency of the drive system.

While deriving the equation of vehicle motion, the last one is replaced by the equivalent mechanical system of material points consisting of vehicle mass moving in translatory motion and some masses which are in translatory and rotational motion at the same time.

Therefore by replacing in the equation (1):

$$I_c = m_c \cdot r_k^2; \quad i_c = i_i \cdot i_0; \quad \frac{d\omega}{dt} = \frac{dV_c}{dt} \cdot \frac{1}{r_k}; \tag{2}$$
$$M_k = F_k \cdot r_k; \quad M_{op} = F_{op} \cdot r_k$$

one receives:

$$\frac{dV_c}{dt} = \frac{F_k - F_{op}}{m_c + I_s \cdot \left(\frac{i_i \cdot i_0}{r_k}\right)^2 \cdot \eta_m + \frac{I_k}{r_k^2}} \quad \text{or} \tag{3}$$

$$\frac{dV_c}{dt} = \frac{F_k - F_{op}}{m_c \cdot \delta_m}$$

where

$$\delta_m = 1 + \frac{I_s}{m_c} \cdot \left(\frac{i_i \cdot i_0}{r_k}\right)^2 \cdot \eta_m + \frac{I_k}{m_c} \cdot \frac{1}{r_k^2} \tag{4}$$

δ_m - coefficient allowing for the rotational motion of inertial masses; V_c - linear speed of vehicle mass moving in the translatory motion, m/sec.; m_c - mass of the vehicle in translatory motion, kg; r_k - rolling radius of the wheels moving without skidding, m; F_k and F_{op} - tangent driving force and total force of resistance to motion, N; i_i, i_0 - ratio of the gearbox and the final drive, respectively.

3 Hydro-mechanical car drive system

Unlike the car with a mechanical transmission in the drive system, which can be represented by a system with one degree of freedom, the car with hydro-mechanical transmission has a dynamic system with two degrees of freedom because the hydro-kinetic transmission divides the car drive unit into two parts, which are not rigidly connected.

A scheme of the car dynamic system (with hydro-mechanical transmission used in the drive system) is presented in Figure 2.

Adopted symbols:

I_P and I_T - moments of inertia, pump and turbine impeller wheels and elements rigidly connected with them, respectively, as well as working liquid in the wheels; M_P and M_T - torque on pump and turbine wheels respectively; Φ_B, Φ_H and Φ_L - friction clutches, blocking hydro-kinetic transmission, switching to higher and lower gears respectively; ω_P - angular velocity of pump impeller in hydro-kinetic transmission.

Equations of mass motion for a car with a hydro-mechanical transmission (Figure 2) when the inertia moment I_T is taken to drive wheels axles, with the clutch Φ_B disengaged and the gear switched on, take the form of:

$$I_P \cdot \frac{d\omega_P}{dt} = M_s - M_P \tag{5}$$

$$\left(I_T \cdot i_c^2 \cdot \eta_m + I_c + I_k\right) \cdot \frac{d\omega_k}{dt} = M_T \cdot i_c \cdot \eta_m - M_{op} \tag{6}$$

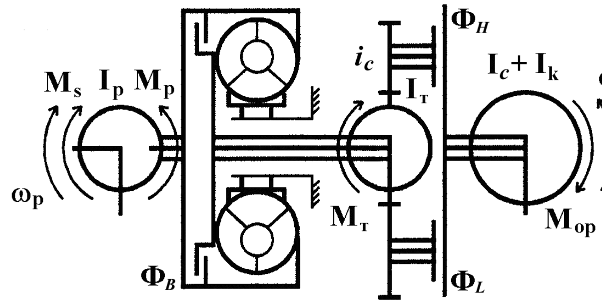

Figure 2: A scheme of the car dynamic system (with hydro-mechanical transmission used in the drive system) [own study]

$$M_P = k_{dh} \cdot \lambda_P \cdot \rho_f \cdot \omega_P^2 \cdot D_a^5 \quad M_T = i_d \cdot M_P$$

where: λ_P - torque coefficient of pump impeller, $\lambda_P = f_1(i_k)$; i_d - coefficient of torque transformation, $i_d = f_2(i_k)$; i_k - hydro-kinetic transmission ratio, $i_k = \frac{\omega_T}{\omega_P}$; ω_T - angular velocity of turbine wheel rotation; k_{dh} - coefficient of dimension harmonisation; ρ_f -working fluid density; D_a - active diameter of hydro-kinetic transmission.

Expressing M_T through M_P in equation (6), solving equation (5) relative to M_P and replacing the last in a rearranged equation (6), after simple rearrangement comes to:

$$\frac{d\omega_k}{dt} = \frac{M_s \cdot i_d \cdot i_i \cdot \eta_m - M_{op}}{I_c \cdot \left(1 + \frac{I_P}{I_c} \cdot i_i \cdot \eta_m \cdot i_d \cdot \frac{d\omega_P}{d\omega_k} + \frac{I_T}{I_c} \cdot i_i^2 \cdot \eta_m + \frac{I_k}{I_c}\right)}.$$

For a car motion as a system of masses moving in translatory motion at velocity V_c and connecting them to turbine shaft through gearbox transmission i_i and transmission of the final drive i_0, taking into account dependencies (2) but also $\omega_k = \omega_T \cdot i_c$ finally gives:

$$\frac{dV_c}{dt} = \frac{F_k - F_{op}}{m_c \cdot \delta_{hm}} \tag{7}$$

where

$$\delta_{hm} = 1 + \frac{I_P}{m_c}\left(\frac{i_i \cdot i_0}{r_k}\right)^2 \eta_m \cdot i_d \frac{d\omega_P}{d\omega_T} + \frac{I_T}{m_c}\left(\frac{i_i \cdot i_0}{r_k}\right)^2 \eta_m + \frac{I_k}{m_c \cdot r_k^2} \tag{8}$$

4 Modelling features of car motion modes

Equations (3) and (7) make it possible to model the following modes of car motion, accelerating, moving at a velocity below maximum, constant motion, and in the special cases, motion at a velocity equal to a speed limit, deceleration caused by the change (decreasing) in the position of the engine control pedal.

For modelling coasting motion when the car is declutched or at neutral position of the gearbox, equations (3) and (7) take the form:

$$\frac{dV_c}{dt} = \frac{-F_{op}}{m_c \cdot \delta_m} \tag{9}$$

where for cars with mechanical and hydro-mechanical transmission

$$\delta_m = 1 + \frac{I_k}{m_c} \cdot \frac{1}{r_k^2} \tag{10}$$

In case of modelling car deceleration while using brakes and the engine, the following equation is used:

$$\frac{dV_c}{dt} = \frac{-F_H - F_{op}}{m_c \cdot \delta_{m(hm)}} \tag{11}$$

where F_H - total overall braking force on the car wheels.

While modelling the motion process of a car equipped with a hydro-mechanical transmission, with the hydrokinetic transmission blocked, equation (7) instead of δ_{hm} should use the coefficient of rotating mass δ'_{hm} defined by the following dependency:

$$\delta'_{hm} = 1 + \frac{I_P + I_T}{m_c} \cdot \left(\frac{i_i \cdot i_0}{r_k}\right)^2 \cdot \eta_m + \frac{I_k}{m_c \cdot r_k^2} \tag{12}$$

Tangent driving force F_k applied to the car driven wheels depends on the value of effective torque generated by the engine or torque on the pump impeller wheel of the hydrokinetic transmission, drive transmission ratio, its efficiency and rolling radius of the driven wheels.

When modelling car motion, engine torque is defined as a function with two variables: angular velocity of engine crankshaft ω_s and the lever position in angular velocity governor h_s, *i.e.* $M_s = f(\omega_s, h_s)$.

In the calculation process the current value of torque for a petrol engine or diesel engine is described as a linear interpolation of function with two variables, from tabular statement, with changeable steps for each argument:

$$M_s = \left[M_{i,j} - M_{i-1,j} - \left(M_{i,j-1} - M_{i-1,j-1}\right)\right] \frac{\omega_s - \omega_{i-1}}{\omega_i - \omega_{i-1}} \cdot \frac{h_s - h_{j-1}}{h_j - h_{i-1}} + \left(M_{i,j-1} + M_{i-1,j-1}\right) \frac{\omega_s - \omega_{i-1}}{\omega_i - \omega_{i-1}} + \left(M_{i-1,j} + M_{i-1,j-1}\right) \frac{h_s - h_{j-1}}{h_j - h_{i-1}} + M_{i-1,j-1}\,, \tag{13}$$

where: $M_{i,j} = f(\omega_i, h_j)$, $i = \overline{1,n}$; $j = \overline{1,m}$; i - the number of tabular value of the first argument (angular velocity); j-the other argument (the lever position in angular velocity governor of a diesel engine or throttling valve angle of a petrol engine); M_s, ω_s, h_s - current values of respective variables

Driving force on car driven wheels is described by the expression:

$$F_k = \frac{M_1 \cdot i_i \cdot \eta_m}{r_k} \tag{14}$$

where: M_1 - torque of input shaft of the gearbox (for mechanical transmission $M_1 = M_s$, for hydro-mechanical transmission $M_1 = M_T$).

Car resistance to motion F_{op} consists of rolling resistance, grade resistance, air resistance and forces of inertia generated when the car accelerates both in translatory motion and rotary motion of rotating parts.

When modelling car mode of decelerating with the use of the braking system and the engine (equation (11)), braking force F_H on the wheels is defined by the expression:

$$F_H = F_{HS} + F_{HB} \tag{15}$$

where: F_{HS} - engine braking force, reduced to drive wheels; F_{HB} - braking force on the vehicle wheels when the basic braking system is used.

Engine braking torque can be determined experimentally by rotating an engine driven from an external source and approximating it as follows:

$$M_{HS} = M_{HS}^0 + b_{HS} \cdot \omega_s \tag{16}$$

where: b_{HS} - constant coefficient; M_{HS}^0 - constant component of engine braking torque.

Now

$$F_{HS} = \frac{M_{HS} \cdot i_i}{r_k \cdot \eta_m} \tag{17}$$

When decelerating with the use of the basic braking system, braking force is defined in such a way that given uniformly decelerated motion with a given force applied, braking distance equals a set distance S_H and is calculated by a formula:

$$F_{HB} = \frac{m_c \cdot \delta \cdot \left(V_c^2 - V_{dop}^2\right)}{S_H} \tag{18}$$

where V_{dop} - car motion speed permissible for safety reasons at a set route section.

An allowed speed for a vehicle going downhill is the maximum speed of a safe downhill drive calculated as m/s on the basis of the formula [17, 27]:

$$V_{dop.tr} = \frac{\alpha_{tr}}{\alpha_{dop}^2} + \frac{\beta_{tr}}{\alpha_{dop}} + c_{tr}$$

where: α_{dop} - slope angle of the track, %; $\alpha_{tr} = -183,7$; $\beta_{tr} = 127,1$; $c_{tr} = 4,45$.

5 Calculation of car motion quality indicators

The results of differential equations (3), (7), (9) and (11), representing motion of a car with mechanical or hydro-mechanical drive system are car motion quality indicators on a set route such as average velocity of the car motion V_a and fuel consumption Q_a on a typical route.

Average velocity km/h is defined by:

$$V_a = 3,6 \cdot \frac{S_c}{t} \tag{19}$$

where: S_c - length of the distance covered, m; t - motion time, s.

Length of the distance covered is calculated by integrating the differential equation:

$$\frac{dS_c}{dt} = V_c \tag{20}$$

where: V_c - current value of car motion velocity, m/s.

Current velocity value is calculated, depending on motion mode and the type of vehicle drive, by integrating one of the differential equations (3), (7), (9) or (11).

Average fuel consumption l/100km is calculated by the formula:

$$Q_a = \frac{Q \cdot 10^8}{\rho_f \cdot S_c} \tag{21}$$

where: Q - total fuel consumption on the route, kg; ρ_f - fuel density, kg/m^3.

Total fuel consumption is defined by integrating the differential equation:

$$\frac{dQ}{dt} = \frac{G_h}{3600} \tag{22}$$

where: G_h - current hourly fuel consumption, kg/h.

In order to define current hourly fuel consumption according to engine full-load characteristic with $G_h = f(N_s)$, $\omega_{si} = const$, $i = \overline{1,p}$, where: p - number of engine full-load characteristics, one should make software tabulated dependency $G_h = f(\omega_s, M_s)$ using a computer program. Value G_h is calculated by linear interpolation of a function with two arguments: ω_s and M_s and a formula analogous to (13).

6 Logic of the driver’s action

The logic of the driver‘s action meets the requirement of full exploitation of car velocity properties.

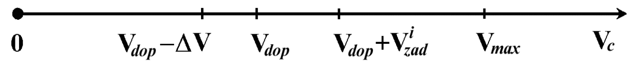

Figure 3: Scheme for determining the mode of car motion [own study]

Mathematical interpretation of the driver‘s actions is expressed by logical operations which are carried out in the process of simulation modelling of car motion and constitute an integral part of the model.

The modelling begins with the simulation of the motion of the car pulling away or moving at a lower gear with set velocity $V_{c.pocz}$, depending on which velocity has been accepted as the initial one, *i.e.* $V_c = 0$, or $V_c = V_{c.pocz} \leq V_{dop}$.

At the same time the pedal of engine work control is in the position $h_s = h_{s.\max}$, which makes it possible for the car to move at the maximum speed for a given pedal position.

The logic of the driver‘s action comes down to accomplishing the following operations:

1. in the process of modelling the motion, at every step of integrating, a current value of vehicle velocity V_c is compared with the velocity which will be permissible at the point of the road with the coordinate equal to the sum of the current coordinate value of the road and the range of visibility;
2. if the current value of vehicle velocity V_c exceeds the permissible velocity, it is necessary to check the relation between the current velocity and the velocity exceeding the permissible one by the determined value V^i_{zad}, where i - the number of activated gear. It is assumed that if $V_{dop} < V_c \leq V_{dop} + V^i_{zad}$ (Figure 3), the driver can reduce the car velocity to the value V_{dop}, pressing the pedal of engine control. For carrying out such control, one should determine the value ω_s corresponding to the motion of the car with the velocity permissible at this gear and the value M_s. Next, by interpolating the function with two variables, one should determine such a position of the engine pedal $h_s = f(\omega_s, M_s)$ that at the next step of integrating, the total current value of motion velocity becomes equal to the permissible one.
3. if $V_c > V_{dop} + V^i_{zad}$, the mode of breaking with the main brake and the engine is modelled. The power of braking is determined by the expression (15).
4. if $V_c \leq V_{dop}$, the current value of motion velocity V_c is checked in the range where ΔV - value characterising the accuracy of keeping the permissible motion velocity by the driver;

5. if $V_{dop} - \Delta V \leq V_c \leq V_{dop}$, the car motion goes ahead with the given velocity until the permissible velocity or the car velocity changes as a result of motion resistance changes. In order to maintain smooth motion with permissible velocity, preserving the equation $F_k = F_{op}$ is required. The required torque of the engine is determined by the expression:

$$M_s = \frac{F_{op} \cdot r_k}{i_i \cdot i_0 \cdot \eta_m^i} \tag{23}$$

where - the drive system ratio and the efficiency at i -th gear, respectively;
6. if $V_c < V_{dop} - \Delta V$, the car acceleration is modelled;
7. the car motion velocity on the slope is limited by the permissible velocity in accordance with the requirements of safety on the slope $V_{dop.tr}$. If $V_{dop.tr}$ is higher than the maximum velocity of the car $V_{c\max}$, it assumes the value of the latter.
8. if the force of summary car motion resistances assumes the negative value, its mode of coasting is modelled.

It is assumed that the driver has an influence only on the car velocity, adjusting his/her actions to the situation on the road. The choice of the moment of switching to higher or lower gear, the choice of the number of gear activated after the phase of braking or coasting, protecting the engine against spluttering, blocking and unblocking the hydrokinetic torque converter are conducted by the system of automatic gear switch according to the established rules of switching and the algorithm of its functioning.

7 Logic of the functioning of the automatic gear switch system

Depending on the number, the kind and interrelations of information, operating and correcting parameters, it is possible to realise diverse rules of switching gears.

The model put forward realises the logic of the functioning of control systems which select the moments of switching gears:

- when linear accelerations of the car at neighbouring gears of the transmission become equal;
- when the car reaches its threshold velocity either excluding engine load or considering engine load.

In the two previous cases it is possible to correct threshold velocities because of the value of acceleration (or deceleration) of the car or because of the direction of the car acceleration changes.

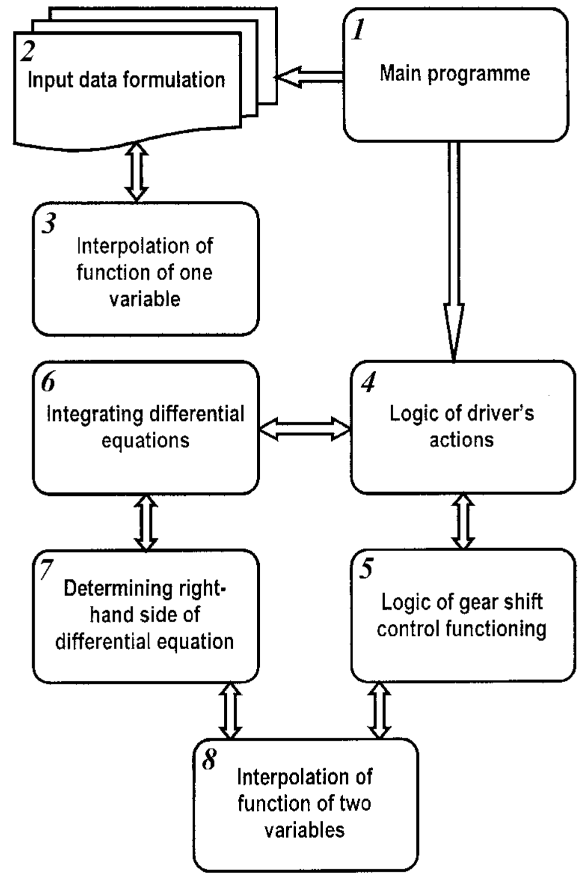

Figure 4: Structure of relations among algorithms of "driver-automation-vehicle-road" system modelling [own study]

In all three cases – due to the direction of changes in the difference between current values of hourly fuel consumption for neighbouring gears of the transmission. Additionally, in the logic of automatic gear switch functioning, the choice of the proper gear in the transmission is predicted after the phase of coasting motion or the car braking.

When the car is coasting or braking, a need to switch to one or a few lower gears is checked, but the signal of the chosen gear activation appears only when the car velocity drops below the value $V_{dop} - \Delta V$.

Proceeding from the phase of coasting or braking to the mode of engine drive takes place in motion if the permissible motion velocity does not equal zero. In addition, only the gear which is appropriate for car motion velocity at the end of the braking or coasting phase is activated in the transmission.

The structure of the relationship between the modeling algorithms of the "driver-automation-car-road" system is presented in Figure 4.

A car equipped with an automatic transmission is a complex system and belongs to a class of logical-dynamic systems [15, 31]. In such systems beside functional dependencies described by numeric functions and differential equations, there are logical connections between parameters and processes which may be analysed with the use of mathematical logic. In a given case logical connections between each motion mode are defined algorithmically. They are of a random character and rely on the logic of the driver's behaviour which depends on road conditions such as topographic profile of the road and the permissible speed at a given section of the route. Moreover, the order of calculations is influenced by relations resulting from the logic of gear shift control system.

When modelling motion of a car with hydro-mechanical transmission, at each step of integrating it is necessary to do calculations concerning matching of hydrokinetic torque converter and engine operations as well as calculations of a coefficient of rotating masses inertia depending on the gear activated and the hydrokinetic torque converter operating mode.

In the modelling process information is obtained every set period of time in order to be printed. It concerns the current time, road distance covered, car velocity, number of the gear activated, drive force tangent value, total car motion resistance force, position of engine control pedal, engine angular velocity and torque, hourly fuel consumption and car acceleration. When motion of a car with hydro-mechanical transmission is modelled, additional information is printed about turbine impeller torque and hydrokinetic transmission operating mode.

After modelling of the whole route is completed, the following information is printed: total motion time, the distance covered, mean velocity, number of gear shifts, total and mean fuel consumption.

8 Conclusion

A mathematical analysis of processes underlying driving torque transformation from the combustion engine to the car driven wheels has been performed. It forms the basis for synthesis of a simulation model representing processes of car drive systems control. This model can be used for designing and optimisation of automated mechanical car drive systems as well as automated hydro-mechanical transmission systems.

The packages of programmes which are taken into consideration allow investigating motion processes of cars equipped with mechanical and hydro-mechanical drive systems with automated gear shift control. This makes it possible to compare and evaluate gear shift rules in relation to all information parameters processed by automatic gear shift control system.

If systems replacing the driver in his role of controlling the car and evaluating the situation on the road are introduced, the developed methodology of synthesising drive system control may be successfully applied to autonomous vehicles.

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