DEVELOPMENT OF DRIVER MODEL FOR VEHICLE CONTROL DURING STRAIGHT LINE MOTION

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INTRODUCTION

Driver has significant role in vehicle driving process. During driving the vehicle task realization driver is exposed to different environmental influences. Result is driver’s fatigue which can be manifested as traffic safety risk. Driver assistance systems are used in vehicle in order to reduce the driver influence on driving and to increase vehicle safety. These are: Lane Departure Warning, Adaptive Cruise Control, Antilock Braking System, Electronics Stability Program, Traction Control System, etc. These systems are briefly described in [15,20,21]

In [14,22,25] specific driver models for vehicle straight line driving task are presented. Since in reviewed literature there is no general compliance about optimal driver model, it was appropriate to simulate driver's role, in vehicle speed following, as well as optimal controller, [14].

Driver model for vehicle straight line motion is developed and it will be presented in the paper. Driver’s task is to follow desired vehicle velocity. Developed model flowchart is given in Figure 1.

According to Figure 1, driver cannot be considered without vehicle model. Driver needs to observe achieved vehicle velocity and compare it with desired. Depending on velocity difference, driver acts on acceleration command. There is driver reaction delay-time between moment of spotting the vehicle velocity difference and moment of driver’s feet action.

Vehicle is complex dynamic system which is exposed to different excitations, road conditions, environment conditions, etc. Due to complexity of considered problem it was appropriate to include road influence by road longitudinal inclination angle. In the following paragraphs, proposed model will be described in detail.

VEHICLE AND DRIVER MODELLING

Driver controls a vehicle by means of: the steering system, the braking system commands, acceleration and clutch command. During driving process driver compares achieved vehicle speed with desired one. Driver’s behaviour causes reaction time delay. Also, there is engine

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reaction time delay. Dynamic system Driver-Vehicle-Environment under straight line vehicle motion is not simple and assumptions should be taken into consideration. Optimal steering concept is adopted according to [13,17,18,20,23], where the gap influences on reaction time delay of transmissions is neglected.

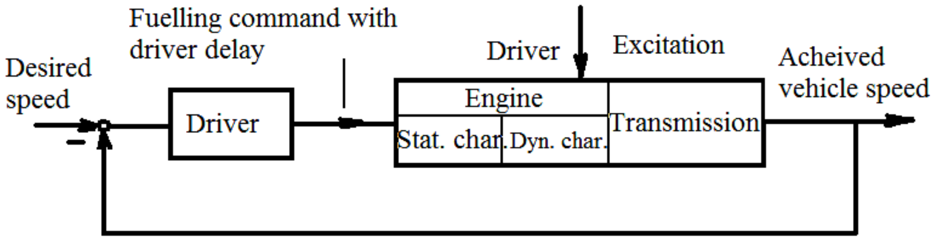

Figure 1: *Flow chart of dynamic system driver vehicle during straight line motion and realization of speed following task*

Engine modeling is a problem with several unknowns, with respect to its non stationary characteristics, which depend on driver acting on gas pedal, engine load, engine number of revolution, etc, [14, 17]. Vehicle with gasoline engine and continuously variable transmission and drive train is used in adopted model.

POWER TRAIN MODELING

In this case, vehicle transmission consists of: an engine (with its dynamics characteristics), ideally continuously variable transmission, final drive and wheels.

Engine modeling is complex and there is need to describe its non stationary speed characteristics because these are important for vehicle performance, [5-9]. Engine torque characteristics with full open throttle angle and partial ones, are given in Figure 2, [24].

Vehicle engine producers do not usually publish partial engine characteristics. Published papers [15,24] show that partial gasoline engine characteristics are not equal distanced. Diesel engine characteristics can be approximated by equal distance torque curves.

In this paper equal distanced partial torque characteristics of gasoline engine are adopted, due to absence of proper data.

Curve given in Figure 2 is approximated by fifth-order polynomial. Approximations gave well coincidence with experimental results. Based on previously explained assumptions, expressions for partial engine characteristics can be written:

$$M_e\left(n_e,\alpha\right)=M_e\left(n_e,\alpha_{100}\right)\alpha \tag{1}$$

where: $M_e(n_e, \alpha)$ – is engine torque dependant on engine speed and fuel supply, $M_e(n_e, \alpha_{100})$ – is engine torque dependant on engine speed under full open throttle angle, α – is control variable which take into account engine fuel supply and has values in interval [0, 1].

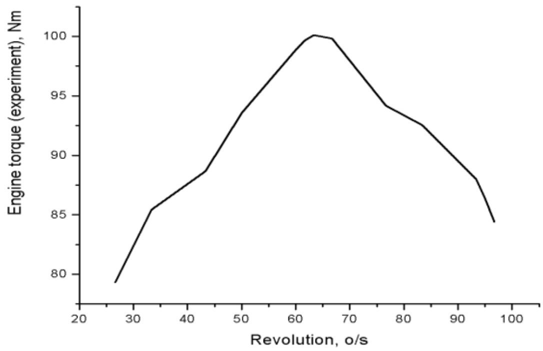

Figure 2: *Engine torque versus engine speed*

Approximated partial engine torque characteristics are given in Figure 3.

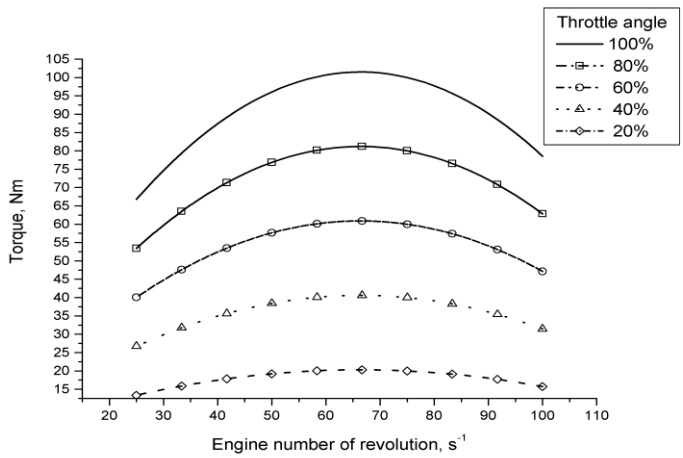

Figure 3: *Engine torque versus speed for different throttle angles*

In Figures 2 and 3 determined stationary partial engine characteristics are given. Curves are measured under constant engine load and constant engine r.p.m. In real operating conditions, engine operates in variable regimes with respect to change of fuel supply, which are usually called dynamic. Appropriate engine characteristics are dynamic. Vehicle engine producers do not publish dynamic engine characteristics, so researches are forced to apply assumptions. For illustration, engine dynamic behavior, under sudden throttle actuation and sudden de-actuation (tip in, back out), are given in Figure 4, [24].

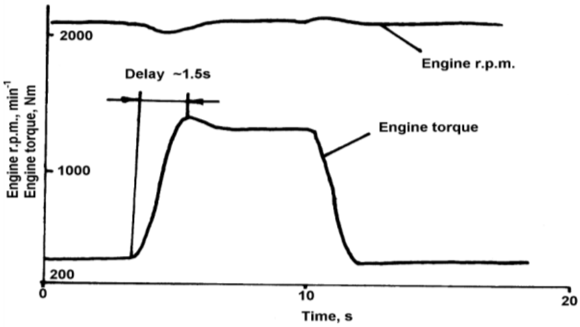

Figure 4: *Engine dynamic behavior under sudden change of fuel supply*

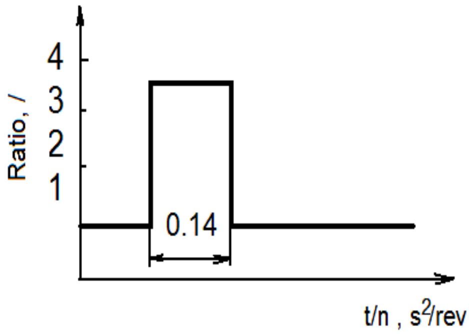

Figure 5: *Normalized engine characteristic*

As it may be seen in Figure 4, engine speed is changed under sudden throttle actuation and engine torque increased with time delay of approximately 1,5 s. Similar engine behavior exists when gas pedal is suddenly released. Exact modeling of engine dynamic characteristics uses approximation of experimental results. Determination of dynamic engine characteristics is very difficult, so following assumptions are adopted:

- engine torque diagram given in Figure 4 may be approximated by quasi Dirac delta function, given in Figure 5 and
- there is implementation of reaction time delay.

Amplitude of quasi Dirac delta function is obtained by normalization of data given in Figure 4. All values given in diagram are divided by stationary engine torque value, before sudden engine fuel supply..

In the absence of experimental data, time delay is correlated to engine number of revolution. All time delay values are divided by engine speed (in this case 2100 min^{-1}).

In this paper, based on data given in Figure 4, engine dynamic characteristics may be expressed by stationary speed engine characteristics multiplied by quasi Dirac delta delay function, given in Figure 5:

$$M_{edyn}(n_e,\alpha)=M_e(n_e,\alpha)dyn(\mathrm{t}) \tag{2}$$

where: *t* – is time, *din(t)* – is time function that takes into consideration engine reaction delay (adopted as 1.5 s), with respect to fuel supply change.

Powertrain consists of ideally continuously variable transmission, drive shaft and final drive, [15]. Adopted characteristic of continuously variable transmission can be found in [15]. Total final drive ratio is 4.

During straight line motion, road profile is included by longitudinal road inclination. According to [11-13,18,23], designations are:

- G – is total vehicle weight,
- r_d – is dynamic wheel radius,
- R_f – is vehicle rolling resistance, $R_f = Gf\cos u$,
- R_u – is vehicle grade resistance, $R_u = G\sin u$
- R_i – is vehicle inertial force: $R_i = \frac{G}{g}\delta j = \frac{G}{g}\delta\frac{dv}{dt}$
- R_v – is vehicle drag force, $R_v = KAv^2$

where are:

- f(v) – is rolling resistance coefficient,
- δ – is coefficient of rolling masses influence,
- u – is longitudinal road inclination angle ("-" for downhill).

During straight line motion, road profile is included by longitudinal road inclination. Vehicle dynamics balance equation is used to determine vehicles acceleration, [12,13,18,23]:

$$j=\frac{g}{\delta}\left[D(v)-f(v)\cos u-\sin u\right] \tag{3}$$

where: D(v) - is dynamic factor determined by expression: $D(v)=\frac{F_o-R_v}{G}$, F_o – is vehicle traction force determined according to [12,13,18,23]. Detailed description of equation (3) can be found in [18,23].

Traction force includes engine torque and transmission dynamic parameters, given in equation (2):

$$F_o = \frac{M_{edyn}\left(n_e,\alpha\right) i_t i_o \eta_t}{r_d} \tag{4}$$

where: i_o – is total drive train ratio, i_g – is total transmission ratio, η_t – is total efficiency coefficient. Achieved vehicle speed value is determined from vehicle differential equation of motion (3).

In order to define general driver model, it was appropriate to analyze the braking process. It was necessary to define universal control variable that include both acceleration and braking processes.

For vehicle with CVT, its longitudinal acceleration is determined by throttle angle, α, and brake pedal force, K. By defining α_b as:

$$\alpha_b = -\frac{K}{K_{\max}} \tag{5}$$

where: $K_{\max}$ is the maximal force exerted to the brake pedal, unified throttle angle $\bar{\alpha}$ can be given as,

$$\bar{\alpha} = \begin{cases} \alpha, \bar{\alpha} \geq 0 \\ \alpha_b, \bar{\alpha} < 0 \end{cases} \tag{6}$$

DRIVER MODELING

Driver's task was to follow desired velocity in straight line driving conditions. Driver model parameters are defined by optimal controller, which parameters are identified by stochastic parameter optimization method, [4-9]. Controller functions cannot be considered separated from vehicle.

Dynamic model Driver - Vehicle – Road system will be described in the following text.

Flowchart of driver-vehicle system, based on optimization method, [4], is given in Figure 7. The objective function is determined as:

$$\Phi = \left(v_{des} - v_{ach}\right)^2 \tag{7}$$

where: v_{des} – is desired vehicle speed and v_{ach} – is achieved vehicle speed.

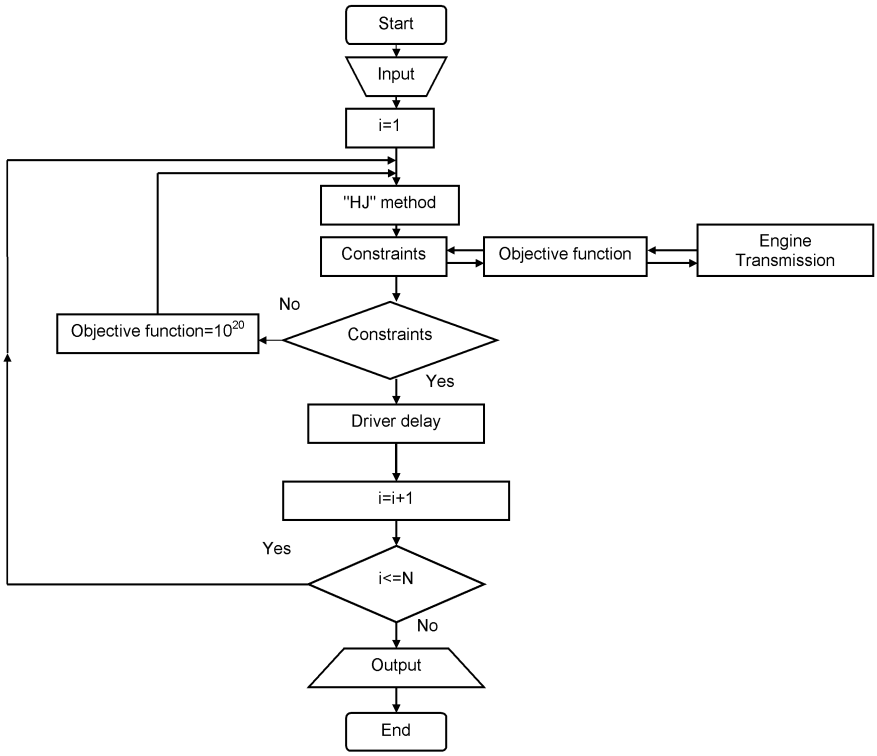

Figure 7: *Driver-vehicle system flowchart*

Achieved speed in expression (7) is a function of universal control variable, α, which value is in interval [-1,1]. Driver controls gas pedal and brake pedal with universal parameter α, which optimal values must be determined from minimum of objective function (7).

Minimum objective function is calculated by implementation of Hooke - Jeeves optimization method, [4], for each discrete time value. Iteration process is finished when difference between adjacent values of objective function is less than 10^{-5}. In order to implement constraints of fueling, external penalty function method is applied, briefly described in [4-11]. Authors developed software in Pascal. Driver's time delay (which is approximately 0.6-0.8 s), is included in dynamic simulation with adopted value of 0.7 s, [14,16,19].

ANALYSIS OF DYNAMIC SIMULATION RESULTS

Based on previous explanations, dynamic simulation is conducted by numerical solving of vehicle differential equation of motion (3). Differential equation (3) is solved by Runge Kutta method with time increment of 0.01 s and time period of 40.96 s. Frequency range, according to time signal, was 0.024 - 50 Hz, [1,2]. It is suitable frequency region for analysis of performed task, because frequency region of interest for driving analysis is around 2.5 Hz, [1,2].

Here will be presented some driver's tasks. Tasks of desired vehicle speed following are given in Figures 8 and 9.

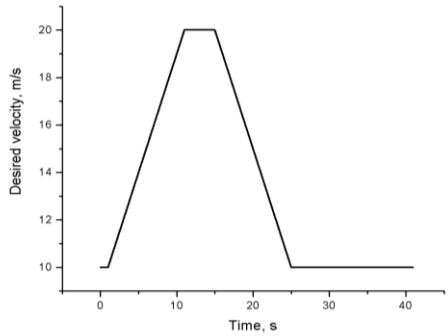

Figure 8: Driver`s task I

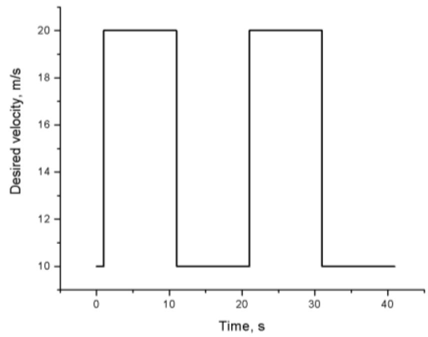

Figure 9: Driver`s task II

The driver's task is given in Figure 8. In this case driver should follow vehicle speed of 10 m/s, than accelerate vehicle to 20 m/s. Achieved speed of 20 m/s should be kept constant for a while. After that, driver should decelerate vehicle to 10 m/s and then continue to drive with constant speed.

The more specific task is given in Figure 9. Driver's task is to change gear transmission twice when accelerating and decelerating vehicle from 10 m/s to 20 m/s and reverse, and to continue constant speed driving at 10 m/s.

In order to get detailed analysis, ordinary coherence function is calculated by software DEMPARCOH [10]. Dynamic simulation is performed with sample size of 40.96 s and number of samples 1024 which caused averaging bias and rms errors of: 0.044 and 0.03 respectively, [1,2]. Ordinary coherence functions are given in Figures 10 to 11.

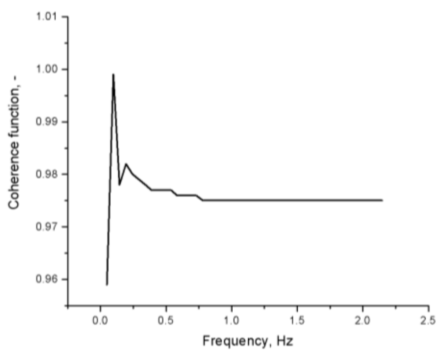

***Figure 10:** Ordinary coherence function, realization of speed following task I*

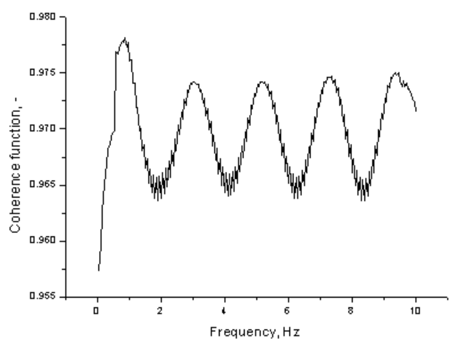

***Figure 11:** Ordinary coherence function, realization of speed following task II*

Ordinary coherence function between desired and achieved vehicle speed, for task defined in Figure 8, is given in Figure 10. Values of ordinary coherence function are between 0.96 and 0.98 (at frequencies higher than 0.3 Hz they are around 0.975). This shows reliable task performance, which is lower than in previous case, because it was more complex task, with sudden changes in driving conditions.

The most complex task was defined according to Figure 9 is between 0.96 and 0.98 (at frequencies higher than 0.3 Hz coherence function osculates around value of 0.98). High ordinary coherence value shows that there is high coincidence between desired and achieved vehicle speed despite of strong task demands, Figure 11, [1,2].

CONCLUSIONS

Based on performed analysis, the following conclusions can be derived:

- Adopted model based on optimization methods showed good performances with respect to task of speed following,
- Model may be used in order to solve different tasks performed during straight line motion.

ACKNOWLEDGMENTS

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