# REAL-TIME MODELING FOR GEAR CHANGE CONTROL STRATEGY RESEARCH

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**Abstract:** The paper deals with an efficiency estimation of automated mechanical transmission at off-line simulation stage. The simulation model of the vehicle with an automated power unit for gear change control strategy research is developed. The complex mathematical description of the analysis scheme is given. It allows adequately react to structure reorganization and parameters change of dynamic system at an application of external disturbances from the road and the driver. The distinctive feature of created software is opportunity of real-time modeling. This decision enables to estimate automated mechanical transmission operation at a wide spectrum of the driver actions. The experiment on driving simulation of the heavy truck with GCW 40000 kg on a road climb is carried out. The estimated values of driving on a route such as average speed, fuel consumption and truck specific productivity are compared during gear changes in automatic and command modes. The conclusions about efficiency of analyzing gear change control strategy are made. The perfecting strategy of automated mechanical transmission controlling algorithms is offered.

At constructing of automated gear change systems (AGCS) there is a question on their efficiency depending on the different gear change control pattern (GCCP), included in operation algorithm. GCCP represents a combination of information parameters, at which the gear change should be carried out. Vehicle speed, throttle position, vehicle acceleration and etc. are used as information parameters [1, 2, 3].

GCCP, personal for each gear, is obtained by approximation of engine's full-load and part-load curves and is submitted by a 2-degrees polynom:

$$\omega = K_0 + K_1 \cdot \alpha + K_2 \cdot \alpha^2,$$

where $K_0$, $K_1$, $K_2$ – numerical factors;
$\alpha$ – accelerator pedal position;
$\omega$ – rotary speed of a crank shaft, at which gear change should be occur.

For influence research of GCCP on an operation efficiency of AGCS the simulation model of the vehicle with an automated power unit (diesel engine, dry friction clutch and mechanical gearbox) is created. The model allows to change gears both in automatic, and in command (manual) modes.
The scheme of vehicle dynamic model used for simulation of its driving is shown in a fig. 1.

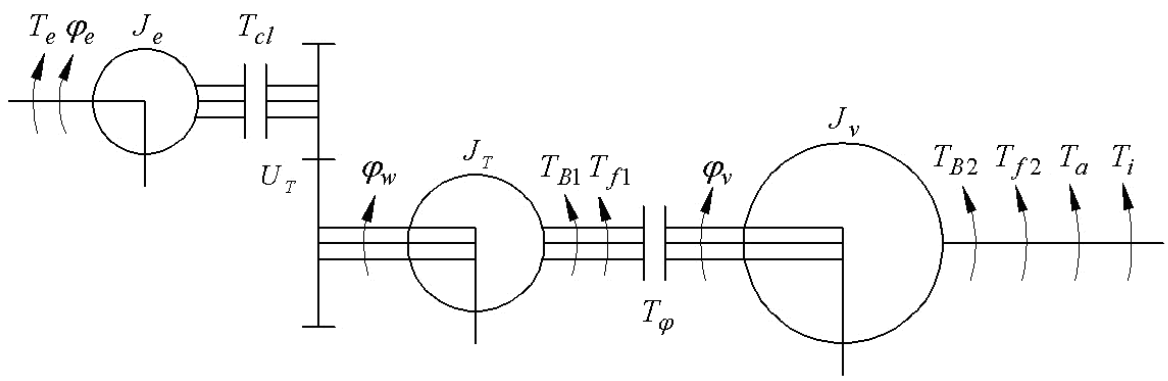



**Figure 1: The scheme of vehicle dynamic model**

On the scheme are designated: $J_e$ – inertia moment of engine rotating parts and part of the clutch; $J_T$ – inertia moment of the transmission and driving wheels; $J_v$ – flywheel inertia moment equivalent to reciprocating mass of the vehicle, and also driven wheels; $\varphi_e$, $\varphi_w$, $\varphi_v$ – angular displacement accordingly of crank shaft, driving and driven wheels; $T_e$, $T_{cl}$, $T_\varphi$ –accordingly engine, clutch and tyre adhesion torques; $T_a$ - air resistance torque; $T_i$ - drag resistance torque; $T_{f1}$, $T_{f2}$ - rolling resistance torque on driving and driven wheels; $T_{B1}$, $T_{B2}$ - brake torques on driving and driven wheels; $U_T$ - transmission ratio.

$$U_T = U_{GB} \cdot U_{TC} \cdot U_0,$$

where $U_{GB}$, $U_{TC}$, $U_0$ – ratios of a gearbox, transfer case and axle gear.
The mathematical description is submitted by a unified system of logic-differential equations, which take into account all possible structural conditions of vehicle dynamic model. It is reached by use of logical switches L1, L2 and switching functions P1, P2 [4].

$$\dot{\omega}_e = \frac{T_e - T_{cl} \cdot sign(\omega_e - \omega_w \cdot U_T) \cdot (1 - L_1) \cdot (1 - P_1) - (T_{B1} + T_{f1}) \cdot L_1 \cdot P_1 / U_T -}{J_e + (J_T + J_v \cdot L_2) \cdot L_1 \cdot U_T^2}$$
$$\overline{- T_\varphi \cdot sign(\omega_w - \omega_v) \cdot (1 - L_2) \cdot L_1 \cdot (1 - P_2) / U_T - (T_{B2} + T_{f2} + T_a + T_i) \cdot L_2 \cdot L_1 \cdot P_2 \cdot P_1};$$

$$\dot{\omega}_w = \frac{T_e \cdot L_1 \cdot P_1 \cdot U_T + T_{cl} \cdot sign(\omega_e - \omega_w \cdot U_T) \cdot (1 - L_1) \cdot (1 - P_1) \cdot U_T - (T_{B1} + T_{f1}) -}{J_e \cdot L_1 / U_T^2 + J_T + J_v \cdot L_2}$$
$$\overline{- T_\varphi \cdot sign(\omega_w - \omega_v) \cdot (1 - L_2) \cdot (1 - P_2) - (T_{B2} + T_{f2} + T_a + T_i) \cdot L_2 \cdot P_2};$$

$$\dot{\omega}_v = \frac{(T_e \cdot U_T - T_{B1} - T_{f1}) \cdot L_1 \cdot P_1 \cdot L_2 \cdot P_2 +}{(J_e \cdot L_1 / U_T^2 + J_T) \cdot L_2 + J_v}$$
$$\overline{+ (T_{cl} \cdot sign(\omega_e - \omega_w \cdot U_T) \cdot U_T - T_{B1} - T_{f1}) \cdot (1 - L_1) \cdot (1 - P_1) \cdot L_2 \cdot P_2 +}$$
$$\overline{+ M_\varphi \cdot sign(\omega_\kappa - \omega_a) \cdot (1 - L_2) \cdot (1 - P_2) - (M_{T2} + M_{f2} + M_B + M_i)};$$

$$L_1 = \begin{cases} 1, \omega_e = \omega_w \\ 0, \omega_e \neq \omega_w \end{cases}; \quad L_2 = \begin{cases} 1, \omega_w = \omega_v \\ 0, \omega_w \neq \omega_v \end{cases};$$

$$P_1 = 0{,}5 \cdot \left[1 + sign\left(T_{cl} - \left|T_e + J_e \cdot \dot{\omega}_e\right|\right)\right];$$

$$P_2 = 0{,}5 \cdot \left(1 + sign\left[T_\varphi - \left| \begin{matrix} \left((T_e + J_e \cdot \dot{\omega}_e) \cdot U_T + J_T \cdot \dot{\omega}_w - T_{B1} - T_{f1}\right) \cdot L_1 + \\ + \left(T_{cl} \cdot sign(\omega_e - \omega_w \cdot U_T) \cdot U_T + J_T \cdot \dot{\omega}_w - T_{B1} - T_{f1}\right) \cdot (1 - L_1) \end{matrix} \right| \right]\right)$$

For an integration of the equations system it is necessary to set the torques of external disturbances, clutch torque and tyre adhesion torque.

The engine torque $T_e$ is determined as a function of an angular velocity of a crank shaft and accelerator pedal position, i.e. $T_e = f(\omega_e, \alpha)$. During the scores the current value of a torque is calculated with the help of linear interpolation on the given engine's full-load and part-load curves in a traction condition. And at the engine braking condition – on mechanical loss curve.

At definition of a transmission efficiency only mechanical losses are taken into account, hydraulic losses are neglected and average values of the gear efficiency is used.
Clutch torque $T_{cl}$

$$T_{cl} = T_{cl0} \cdot (1 - e^{-K_c \cdot t}),$$

where $T_{cl0}$ – clutch peak torque, N·m;
$K_c$ – clutch actuation rate, $sec^{-1}$;
t – current time, sec.

The clutch actuation rate is increased at transition to the following higher gear.

Tyre adhesion torque $T_\varphi$ for the rear wheel drive vehicle

$$T_\varphi = \varphi \cdot R_{Z2} \cdot r_w,$$

where $\varphi$ –adhesion factor;
$r_w$ – wheel rolling radius, m;
$R_{Z2}$ – road normal response on a rear axle, N.

Rolling resistance torques on driving and driven wheels $T_{f1}$, $T_{f2}$

$$T_{f1} = G_1 \cdot f \cdot \cos\alpha_R \cdot r_w,$$

$$T_{f2} = G_2 \cdot f \cdot \cos\alpha_R \cdot r_w,$$

where $G_1$, $G_2$ – weight on driving and a driven wheels, N;
$f$ – rolling resistance factor;
$\alpha_R$ - gradient of the road longitudinal profile.

$$f = f_0 + f_1 \cdot V_v + f_2 \cdot V_v^2,$$

where $f_0$ – tyre rolling resistance factor, rolling with fixed radius;
$f_1$, $f_2$ – factors depending on suspension system and road microprofile;
$V_v$ – vehicle speed, km/h.

Drag resistance torque $T_i$

$$T_i = G_v \cdot \sin\alpha_R \cdot r_w,$$

where $G_v = G_1 + G_2$ – vehicle weight.

Air resistance torque $T_a$

$$T_a = K_a \cdot A_a \cdot V_v^2 \cdot r_w,$$

where $K_a$ – air resistance factor, N·sec²/m⁴;

$A_a$ – vehicle cross-section area, m²;

$V_v$ – vehicle speed, m/sec.

The brake torques on driving and driven wheels $T_{B1}$, $T_{B2}$ can vary in depends on braking intensity, but their maximum values are limited by tyre adhesion.

$$T_{B1} = \varphi \cdot R_{Z2} \cdot r_w,$$

$$T_{B2} = \varphi \cdot R_{Z1} \cdot r_w,$$

where $R_{Z2}$, $R_{Z1}$ – road reaction forces on vehicle rear and front axles, N.

Described simulation model realizes driver operations logic on a route and allows to analyze vehicle dynamic at random external disturbances.

Distinctive feature of the designed software is capability of real-time simulation [5, 6]. The researcher visually receives the information about driving process by means of the virtual instrument panel (fig. 2), mapped on the computer screen. Thus the driver operations on vehicle control can be imitated with the help of the keyboard that enables to estimate AGCS operation at a wide spectrum of driver actions. For example, at error gear selection in a command mode, or at simultaneous pressing on a brake and accelerator pedals.

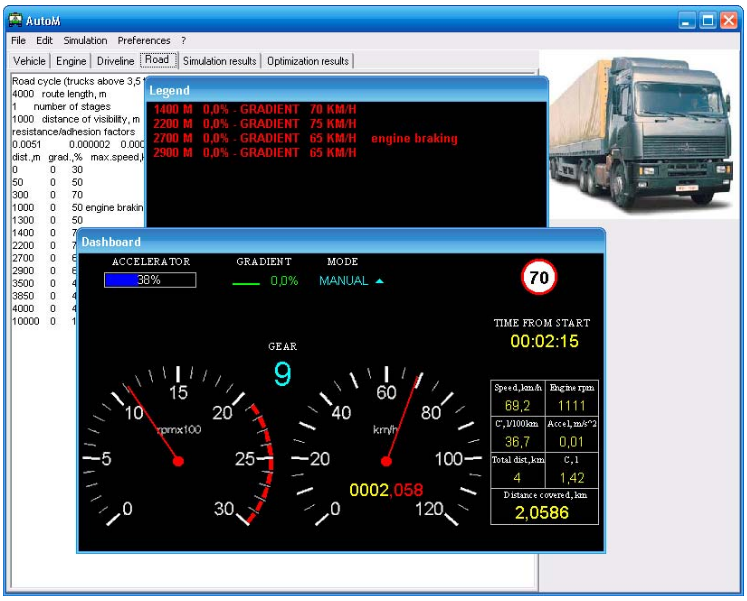



**Figure 2: General view of the screen**

For an estimation of quality and efficiency of AGCS operation in automatic and command modes (different GCCP are realized) it is offered to use such values as transiting time and average speed on a route, fuel consumption, number of gear changing and specific productivity [1]. Among over listed the main is the specific productivity – complex estimated value which is taking into account payload, average speed and fuel consumption:

$$W = P \cdot V_a / C_a \text{, kg·km}^2 \text{ / (0,01 l·h),}$$

where $P$ – payload, kg;

$V_a$ – average speed, km/h;

$C_a$ – average fuel consumption, l/100km.

The capabilities of the real-time simulation model are illustrated by circumscribed below experiment. The real road situation on overcoming long road climb by extent of 900 m with 5 % gradient was reproduced. Roadtrain consisting of tractor MAZ-544008 with semitrailer (see tab. 1) was taken at the modeling. Before the beginning of a climb the vehicle was moved with constant speed of 70 km/h, previously having accelerated on a stage of 1000 m. The purpose of speed loss minimization was put at driving on the climb. In this case the roadtrain did not become a hindrance for more dynamical traffic participants. Therefore driving was carried out with full depression of accelerator pedal.

**Table 1: Vehicle characteristics**

| Parameter | Type, model, value |
|---|---|
| GCW, kg | 40000 |
| Engine | YAMZ 7511.10, turbo diesel |
| Max power, kW (rpm) | 294 (1900) |
| Max torque, Nm (rpm) | 1715 (1200-1400) |
| Gear box | MAZ-543205, eight speed plus one crawler gear |
| Gear ratios | 10,08; 6,13; 4,51; 3,5; 2,78; 1,75; 1,29; 1,0; 0,795 |
| Axle gear ratio | 3,86 |
| Tyres | 315/80 R22.5 |

The results of simulation of driving on a road climb at command and automatic AGCS operation modes are shown in a fig. 3. The estimated values of a driving are represented in tab. 2.

**Table 2: Estimated values of vehicle driving on a route**

| Typical heat | Speed at the end of a route, km/h | Transiting time of a route, sec | Average speed, km/h | Number of gear changing | Average fuel consumption, l/100km | Specific productivity, kg·km2/ (0,01 l·h) |
|---|---|---|---|---|---|---|
| 1 | 26,7 | 91,4 | 35,4 | 5 | 117,9 | 7101 |
| 2 | 33,8 | 73,1 | 44,3 | 3 | 118,8 | 8819 |
| 3 | 40,6 | 63,7 | 50,9 | 1 | 126,1 | 9546 |

From a series of the heats made with a help of simulation model it is possible to select three typical, conditionally called:

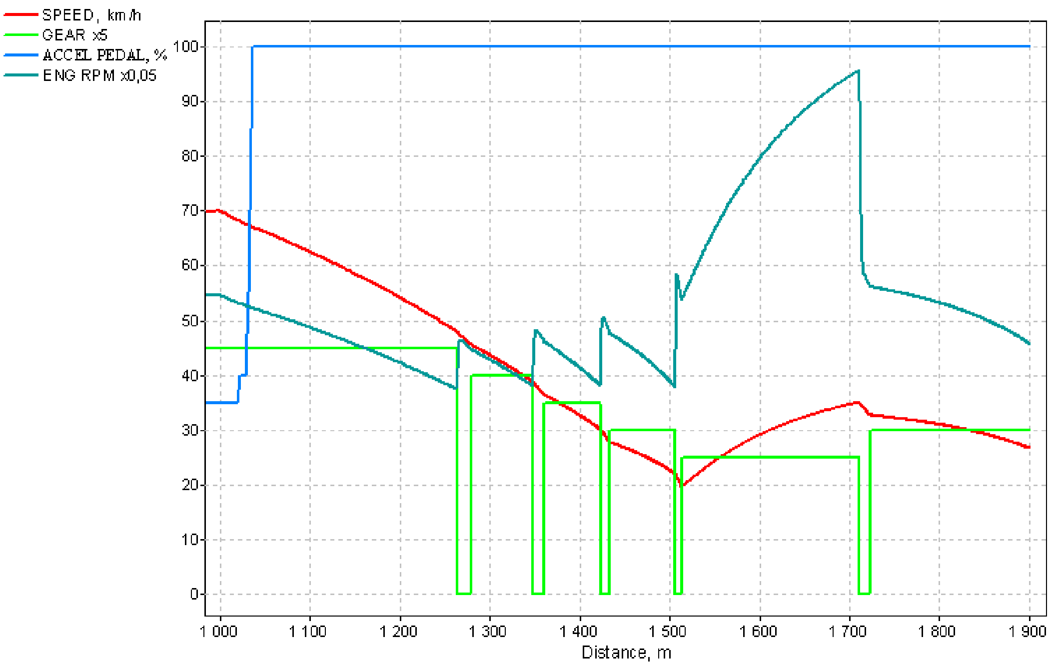
1) «Unskilled driver»;

2) «Automat»;

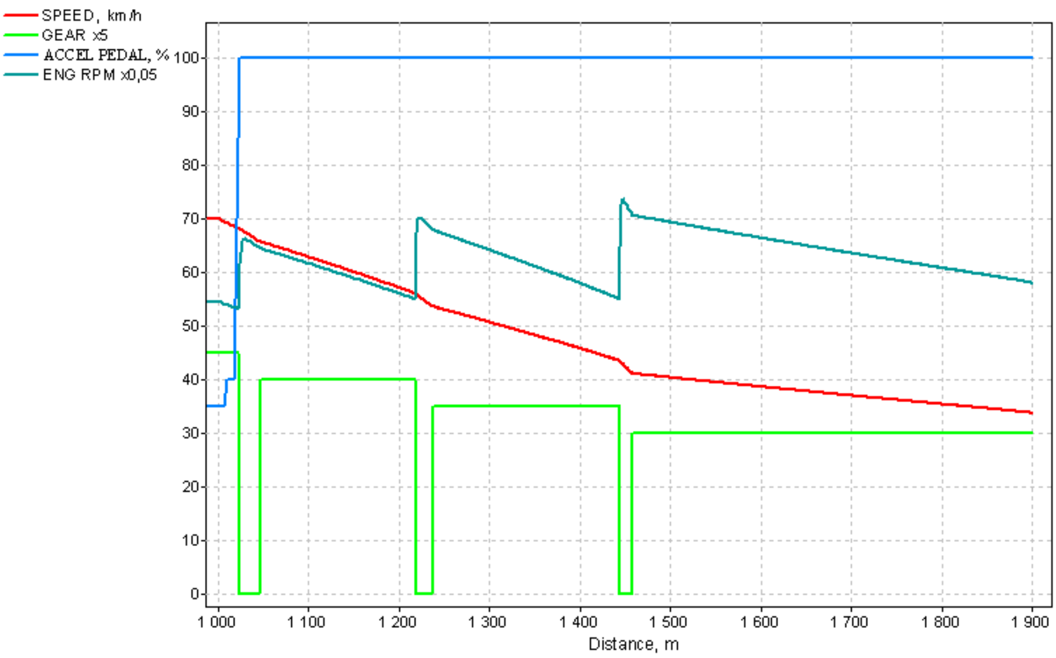
3) «High qualification driver».

It is necessary to mark, that in the first case took place incorrect gears selection by the driver (i.e. by the PC operator) in a point of reaching a specified purpose – speed loss minimization. In the second case the gears selection was carried out not by the driver, but by the AGCS under the developed gear change control pattern. In the third case gears selection by the driver was the most effective. It was established by practical consideration after the analysis of all made heats.

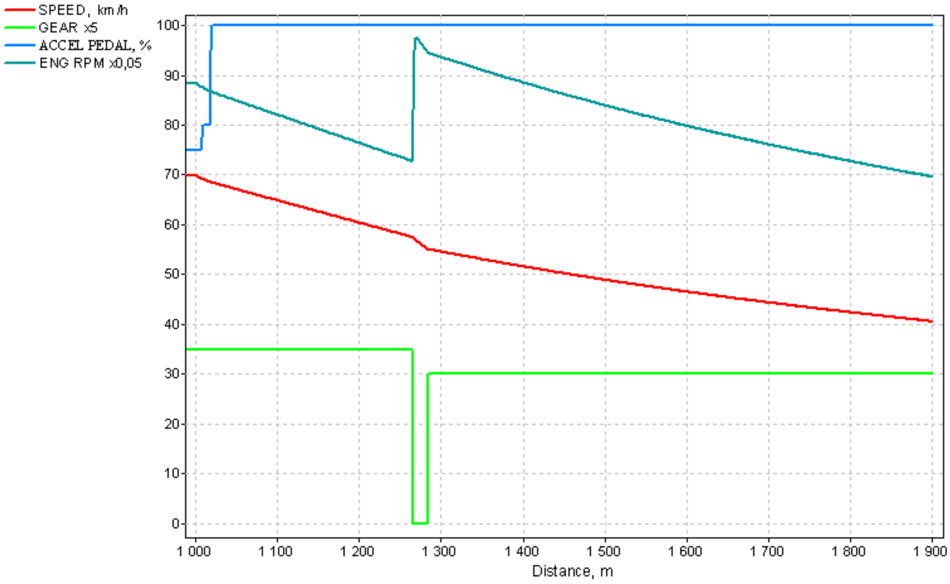
In a case of «Unskilled driver» (fig. 3, a) the driving was carried out in a «lug condition», i.e. in order to «hold on» to the end of road climb and not wishing engine operation on heightened revolutions, the driver switched to the lowest gear as later as it possible. However it had lead to loss of too much speed, that had forced to select the 5-th gear already through 500 m after the beginning of road climb. Besides it was the greatest number of gear changing, owing to increasing the time of power flow gap. The slowest average speed result in decreasing of specific productivity on 25,6 % in compare with version «High qualification driver», despite of the lowest fuel consumption (tab. 2).
In a case of «High qualification driver» (fig. 3, c) the preliminary switch on 7-th gear was realized, and at the end of a road climb the vehicle was moved on the 6-th gear. The selected gears have allowed to support maximum traction force on driving wheels, that in the total has given the greatest average speed on a stage at some change for the worse of fuel economy. However specific productivity has appeared maximal in this occasion.
A fig. 3, b illustrates the driving on road climb in automatic AGCS operation mode. The moment of gear changing and gear number was determined according to gear change control pattern putting in AGCS operation algorithm, i.e. did not depend on the driver. It is necessary to mark, that on specific productivity this version is worse than version «High qualification driver» on 7,6 %, however from the point of fuel economy – has an advantage on 5,8 % (see tab. 2).




**a**




**b**




**c**

**Figure 3. The graphs of driving simulation on a stage:**
**a** – in a command mode («Unskilled driver»); **b** – in an automatic mode («Automat»);
**c** – in a command mode («High qualification driver»)

## Conclusion

The using of the real-time simulation model in practice for an efficiency estimation of the gear change control pattern is shown on the considered example. In the long term there is a capability essentially to improve AGCS operation efficiency in an automatic mode («Automat»), having programmed in gear-shift algorithm the optimum gear change control patterns. For this purpose it's necessary to solve a task of GCCP synthesis by a method of a multicriteria parametric optimization, that will require, alongside with the simulation model, development of complex optimization model.

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