Simulation model of a transport vehicle with a fixed-ratio transmission and a flywheel energy storage in case of random external action

V Korsunskiy\textsuperscript{1,2}
\textsuperscript{1}Bauman Moscow State Technical University
\textsuperscript{2}E-mail: vakors@bmstu.ru

Annotation. The purpose of this study is to evaluate the dynamic qualities and efficiency of a heavy-duty transport vehicle equipped with a mechanical transmission, using a combined power plant (CPP) with a flywheel energy storage (FES), establish an acceptable level of power reduction of the main internal combustion engine (ICE), determine the range of installed power of the FES drive and its average value of efficiency, as well as the calculation of the level of required energy capacity of the flywheel when driving in various types of terrain. A universal technique for simulating statistical modeling of external disturbances acting on the transport vehicle has been developed. Using the developed methodology, the vehicles’ characteristics have been compared according to the criterion of average speed and an assessment of the efficiency of the use of CPP has been carried out. The results obtained allow us to carry out a comparative analysis of vehicles equipped with a conventional or combined power plant with a random external action of the road, as well as optimize their parameters.

Introduction (Immediacy of the problem)
The feasibility of using a combined power plant (CPP) on transport vehicles (TVs), which uses a flywheel energy storage (FES) as an additional energy source, has been proved by a number of scientific studies: [1-7]. An example of a practical use of a device that accumulates the braking energy of a vehicle and develops great power during acceleration using a fast-rotating flywheel is a system developed by Flybrid Systems for racing cars [8].

To use CPP with FES in TVs of various weight categories, the Oak Ridge National Laboratory (ORNL) conducted a comparative analysis of their characteristics and designs [9]. Tests of heavy-duty vehicles equipped with CPP [9], conducted by a number of leading US and Western European companies, made it possible to determine the level of energy accumulated (delivered) by the flywheel equal to 2.0 kWh (7.2 MJ) with a peak (short-term) power developed by the FES drive of 150 ~ 200 kW. In addition, to date, extensive experience has been gained in the countries of the European Union in operating fast-rotating flywheels as part of CPPs used for heavy-duty urban vehicles, for example, buses with ICES and various electric vehicles: [10-13].

Statement of the problem, assumptions, design formulas and limitations
The purpose of this study is to evaluate the dynamic qualities and efficiency of heavy-duty TVs equipped with CPP and mechanical transmission, determine the acceptable level of power reduction of the main engine while maintaining the average speed of the vehicle, calculate the required energy capacity of the flywheel, the drive's installed power and its average value of efficiency when driving at random terrain.

The external environment action on TV is represented by a vector \( \vec{F} \) with random components possessing the properties of stationarity and ergodicity

\[
\vec{F} = (f_{gr}, \alpha, \phi_{ad}, \mu_{\text{max}}, h, k,)^\top, \tag{1}
\]

where \( f_{gr} \) is the coefficient of ground resistance to the rectilinear movement of the vehicle; \( \alpha \) is slope angle of the terrain profile; \( \phi_{ad} \) is adhesion factor; \( \mu_{\text{max}} \) is maximum coefficient of resistance to rotation of the vehicle; \( h \) is the height of the irregularities of the micro profile of the path; \( k \) is road curvature of the path.

An accurate description of the processes occurring in the mathematical model elements describing the transient motion of TV with CPP, considered, for example, in [14], complicates the modeling algorithm. The difficulty of solving the problem using computer due to the complexity of solving a system of nonlinear differential equations when calculating one random implementation of the vector \( \vec{F} \) minimizes the total number of possible implementations.

The level of formalization of a system of differential equations describing the motion of the TV with CPP can be various. In simulation modeling, the models that are implemented in the form of difference equations are most often used.

In relative form, the formalized motion equation of the center of mass of TV with CPP appears as follows:

\[
\frac{d\vec{V}_c}{dt} = \frac{g}{\delta} (f_{dv} \pm f_m - f_c), \tag{2}
\]

where \( \vec{V}_c \) is the velocity vector of the center of mass of TV; \( t \) is time; \( f_{dv} \) is specific thrust of ICE realized in the soil, taking into account losses in the transmission and propulsion of the TV; \( f_m \) is the specific traction force developed by the FES in the soil, taking into account losses in the drive [15], the transmission (in case the drive is connected to the drive wheels through the transmission) and the propulsion; \( f_c \) is total coefficient of resistance during curvilinear motion of the TV; \( \delta \) is coefficient of conditional mass increment of the TV; \( g \) is gravity acceleration. A negative value \( f_m \) means that the FES is in charge mode.

The total coefficient of resistance to the rectilinear motion of the TV is determined by the known relationship

\[
f_{st} = f_{gr} (\cos \alpha + \sin \alpha). \tag{3}
\]

When driving along a random path, the external action exerted on the TV, determined by the right-hand side of equation (2), is taken into account by the relationship of the total coefficient of resistance to movement of the vehicle \( f_{st} \) and travelS. For this purpose, it is necessary to set the distribution function of the coefficient \( f_{st} \) along the path, i.e., \( \Phi_x(f_{st}) \). Graphs \( \Phi_x(f_{st}) \) for two types of terrain are shown in Fig. 1.

To simplify the solution of the problem, we assume that the components \( f_{gr} \) and \( k \) of the random vector \( \vec{F} \) specified by formula (1) have constant values on the intervals of the path \( \Delta S \). The value \( \Delta S \) determines the degree of detail of the road. The choice of a small value \( \Delta S \) makes it possible to take into account disturbances that affect the dynamic loading of the transmission units and the vehicle’s cushioning system more than the average speed.
According to equation (2), it is possible to determine the acceleration of TV with CPP on each section $\Delta S$ of the road using the traction characteristic (Fig. 2).

To construct the traction characteristic of the TV with a general power plant (GPP), it is necessary to have an experimental external characteristic of the ICE (i.e., the relationship between the torque and speed and control parameter), the values of the general gear ratios $i_\alpha$ of the transmission in various gears in the gear box, the drive wheel radius $R_{dw}$, and also calculated or experimental relationships of the overall efficiency $\eta_o$ or losses vs load and speed of the vehicle [16]. For vehicles with CPP, you also need to know the maximum power transmitted by the FES drive, as well as the relationship of the losses in the drive vs the load and speed.

The critical speed at which the sidewise skidding of the vehicle begins is found from the formula

$$V_{cr} = V_c \leq \sqrt{\frac{\mu_{max}}{k}},$$

where $\mu$ is the real curvature of the TV movement, which differs from the road curvature $k_r$ and depends, in particular, on the type of steering mechanism.

The traction characteristic of TV with GPP is based on the experimental points obtained, for example, using spline interpolation.

A similar characteristic for TV with CPP differs by a constant value within the speed change in the $k$-th gear in the box by $f_{mk max}$ (see Fig. 2). The limit value $f_{mk max}$ is limited by the strength of the hydrostatic (or power transmission) and mechanical branches of the FES drive, i.e. maximum torque (or maximum power) transmitted by the drive. In the case of using frameless double-sided synchronous DC motors in the adjustable drive branch, power losses can be taken into account according to the method described in [17].

In the calculations, the assumption is made that switching to the highest gear occurs when the TV reaches the maximum speed value in this gear. That is, with $V_{c1 max}$ = $V_{k-1 max}$, the gear box changes from $(k-1)$-th to $k$-th gear (see Fig. 2).

In contrast to that proposed in [18], the control algorithm of the vehicle is chosen so that when switching from the $(k-1)$-th to $k$-th gear, the following conditions are satisfied:
where $V_{k-1\text{max}}$ is the maximum speed of the car in (k-1)-th gear in the gearbox;

\[ V_{\text{perm}i} \] - permissible speed of the vehicle on the i-th section of the road;

\[ f_{d k\text{max}} \] and \[ f_{m k\text{max}} \] - maximum values of specific thrust forces developed, respectively, by ICE and FES on propulsion devices when gearbox is engaged in the k-th gear (see Fig. 2);

\[ S_i \] and \[ \Delta L \] - the current value of the path and the section of the road visually assessed by the driver;

\[ E_{mri} \] and \[ E_{mm\text{ax}} \] - the current and maximum energy value (maximum energy capacity) of the flywheel energy storage.

The permissible speed of the vehicle on the i-th section of the road is determined by the formula

\[ V_{\text{perm}i} = \min(V_{cri}, V_{\text{cush}i}) \] (6)

where \[ V_{cri} \] is the critical speed of the TV in the i-th section of the road, determined by formula (4) for random values $\mu_{max,i}$ generated by the computer and real curvature $k_i$ and their distribution functions along the path; \[ V_{\text{cush}i} \] - TV speed on the i-th section of the road, limited by the cushioning system and determined for random values $h_i$ generated by the computer and obtained using the speed characteristics of the cushioning system $h(V)$ and the given distribution function $\Phi_s(h)$ for height of the irregularities along the path [19].

Switching from the k-th gear to the (k–1)-th gear occurs when the TV reaches the minimum speed in this gear $V_{k\text{min}} = V_{k-1\text{max}}$ in the following cases (Fig. 2):

a) when the minimum speed in the k-th gear is greater than the permissible speed, i.e. at $V_{k\text{min}} > V_{\text{perm}i}$;

b) under condition $f_{d k\text{max}} + f_{m k\text{max}} < f_{ci}$.

Since the traction characteristic of the vehicle (Fig. 2) is non-linear in nature, in order to take the movement equally variable, it is necessary to refine the current values $f_{d k}$ and $f_{m k}$ even when the vehicle passes sections of the road less than the value set at the beginning of the calculation of the constant value $\Delta S$.

The above assumptions allow us to proceed to the determination of the vehicle with CPP parameters without solving a system of nonlinear differential equations.

The movement of the vehicle with CPP at a speed $V_i$ when the k-th gear is engaged on the i-th road section will be described in a generic form by the system:

\[
\begin{align*}
\frac{dV_i}{dt} &= \frac{g}{\delta_k}(f_{d k} - f_{m k}), \\
\Delta f_{d k} &= \psi_1(M_{d k}; n_{d k}; u_{d k}), \\
\Delta f_{m k} &= \psi_2(M_{m k}; n_{d k}; u_{f k}), \\
\Delta E_{m} &= \psi_3(f_{m k}; n_{d k}; u_{f k}), \\
V_i &\leq V_{\text{perm}i},
\end{align*}
\] (7)
where $\delta_k$ is the coefficient of the conditional increment of the vehicle mass in the $k$-th gear; $\Delta f_{d\delta}$ and $\Delta f_{mi}$ - total losses in the transmission and propulsion, as well as losses in the FES drive; $M_{ni}$ and $M_{dwi}$ are the torques on the FES drive shaft and on the driving wheels; $n_{dvi}$ and $n_{dwi}$ - rotational speed of the ICE shaft and driving wheels; $\Delta E_{ni}$ - change in FES energy on the i-th road section; $u_{dv}$ and $u_{fu}$ - control parameters of the ICE and FES drive.

Quantitative values of $\Delta f_{dvi}$ and $\Delta f_{mi}$ depend on the type of transmission and drive FES. For comparative calculations, the values of $\Delta f_{dvi}$ and $\Delta f_{mi}$ are assumed constant on each gear in the gearbox.

**Analysis of the simulation results.** Quantitative calculations were obtained for a vehicle with a general ICE, as well as for a TV with a CPP with a mechanical transmission, the kinematic scheme and numerical characteristics of which are given in [20].

The maximum speed of TV adopted in the calculations was $V_{\text{max}} = 16.91$ m/s (61 km/h). In the calculations, the section of the road $\Delta L$ estimated by the driver ranged from 20 to 100 m.

The relative energy capacity $E_o$ of the flywheel energy storage was determined by the formula

$$E_o = \frac{E_{m\text{max}}}{E_{v\text{max}}},$$

(8)

where $E_{v\text{max}}$ is the kinetic energy of the TV at maximum speed.

The maximum installed power of the FES hydrostatic-mechanical drive adopted in the calculations was 258 kW.

Figure 3 shows the relationships of the TV average speed $V_{av}$ and the ICE load factor $\bar{z}_{dv}$ vs the relative energy capacity $E_o$ determined by the formula (8).

Figure 4 shows the curves of changes in the vehicle average speed $V_{av}$ depending on the limitation $V_{\text{perm}} = \text{const}$.

**Figure 3.** 1 – ICE loading $\bar{z}_{dv}$; 2 – TV average speed $V_{av}$.

**Figure 4.** Graphs of the TV average speed for various values of the speed limit: 1 - TV with CPP; 2 - TV with ICE.
Figure 5 illustrates graphs of changes in the speed of TV with CPP, TV with a conventional ICE, the current energy value of the flywheel energy storage $E_m$ and the total coefficient of resistance to rectilinear movement $f_{st}$ when it is linearly interpolated along the path $S$ (a fragment of unsteady TV movement over mild terrain with a length of 1500 m with mean value along the path $m_s(f_{st}) = 0.104$ and dispersion $D_s(f_{st}) = 0.00728$). The relative energy capacity of the flywheel was $E_0 = 1(E_{m_{max}} = 5.71 \text{ MJ})$ with the mean value of the FES drive efficiency along the path $m_s(\eta_m) = 0.8$. At the beginning of the TV with CPP movement, the flywheel was not charged.

To determine the possibility of reducing the installed power of ICE in the case of using CPP, it is necessary to carry out a comparative calculation of vehicles with CPP and conventional ICE with the equality of their average speeds (i.e., at $V_{av\ CPP} = V_{av\ ICE}$).

Fig. 6 shows the curves of changes $V_s\ rpm$ depending on $E_o$ for various values of installed power and the mean values of the flywheel drive efficiency $m_s(\eta_m)$ along the way when the vehicle is moving in mild terrain. The value of ICE free power in the calculation was 383 kW (80% of the nominal). An analysis of the curves shows that only with high mean values of the drive efficiency ($m_s(\eta_m) = 0.9$) and with the installation power of the FES drive $N_{max}$ equal to 440 kW, a flywheel with a relative energy capacity $E_0 = 1.5$ (i.e., at $E_{m_{max}} = 2.475 \text{ kW-h}$) provides the same average speeds for a conventional vehicle with a nominal ICE power of 485 kW and for a TV with CPP with ICE power reduced by 20%.

Figure 5. Graphs of changes in the TV parameters along the path: 1 - speed of TV with CPP; 2 - speed of TV with ICE; 3 - FES energy; 4 - coefficient of total resistance to rectilinear motion
Conclusions
The performed statistical analysis of the possibility of using CPP in TVs with a fixed-ratio transmission allows us to draw the following conclusions:

The use of CPP containing ICE and flywheel energy storage (FES), allows to increase the average speed of the TV by an average of \(10\%\)–\(20\%\) depending on the mean values of the FES drive efficiency, its installed power and the type of terrain (road).

Direct speed limitations when driving on the road allow us to get an additional \(5\%\) increase in average speed due to energy recovery during TV braking.

While maintaining the same average speed of a vehicle moving on mild terrain, it is possible to use ICE with a power reduced by \(10\%\) of the nominal value for TV with CPP. In this case, the maximum energy capacity of the FES should be at least \(2\text{ kW•h}\), and the mean value of the FES drive efficiency along the path \(m_s(\eta_m)\) should be at least \(0.8\).

A decrease in the installation power of the main ICE of the vehicle will, among other things, contribute to increased fuel economy.

References
[1] Genta G. Kinetic energy storage: theory and practice of advance flywheel systems. Butterworth–Heinemann Ltd, 1985. – 374 p.
[2] Hebner R., Beno J., Walls W. Flywheel batteries come around again. IEEE Spectrum, 2002, vol. 39, no. 4, pp. 46–51.
[3] The design of an engine-flywheel hybrid drive system for a passenger car / Schilke N., DeHart A., Hewko L. et al. Proceedings of The Institution of Mechanical Engineers, Part D: Journal of Transport Engineering. 1986, vol. 200 (44), pp. 231-248.
[4] Kharade R. Regenerative Braking in Automobiles. International Journal of Engineering and Techniques, 2017, vol. 3, issue 6, pp.216–221.
[5] Thoolen F. Development of an advanced high speed flywheel energy storage system.Eindhoven TechnischeUniversiteit, Eindhoven, 1993. doi: 10.6100/IR406829.
[6] Flywheel Energy Storage for Automotive Applications / M. Hedlund, J. Lundin, J. Santiago et al. Energies, 2015, no 8, pp. 10636–10663. doi:10.3390/en81010636.
[7] Dhand A., Pullen K. Review of flywheel based internal combustion engine hybrid vehicles. International Journal of Automotive Technology, 2013, no 10, vol.14, issue 5, pp. 797–804.
[8] Tinham B. Breakthrough in Ricardo Kinergy high-speed flywheel technology. Transport Engineer. 23.08.2011. Available at: URL: http://www.transportengineer.org.uk/transport-engineer-news/breakthrough-ricardo-kinergy-flywheel-technology/36277/ (accessed 31 May 2019).

[9] Hansen J., O’Kain D. An Assessment of Flywheel High Power Energy Storage Technology for Hybrid vehicles. Reports. December 2011, ORNL/TV-2010-280.

[10] Principle, Design and Experimental Validation of a Flywheel-Battery Hybrid Source for Heavy-Duty Electric Vehicles / O. Briat, J.M. Vinassa, W. Lajnef et al. IET Electric Power Applications, vol. 1, issue 5, September 2007, pp.665–674. doi:10.1049/iet-epa:20060458.

[11] Low Cost Flywheel Energy Storage for a Fuel Cell Powered Transit Bus / C. Hearn, M. Flynn, M. Lewis et al. Vehicle Power and Propulsion Conference, Sept. 9 – 12, 2007. VPPC 2007. IEEE. doi: 10.1109/VPPC.2007.4544239.

[12] Ogasa M. Application of energy storage technologies for electric railway vehicles – examples with hybrid electric railway vehicles. IEE Transactions on Electrical and Electronic Engineering, 2010, no. 5(3), pp. 304–311. doi:10.1002/tee.20534.

[13] Model-based comparison of hybrid propulsion systems for railway diesel multiple units / Schmid S., Ebrahimi K., Pezouvanis A. et al. International Journal of Rail Transportation, 2017, vol. 6, issue 1, pp. 16–37.

[14] Korsunskiy V.A. [Mathematical model of hydrostatic-mechanical drive for the flywheel energy storage]. Trudy NAMI, 2017, no. 4 (271), pp. 38–45. (In Russian).

[15] Abrahamsson J., Bernhoff H. Magnetic bearings in kinetic energy storage systems for vehicular applications. Journal of Electrical Systems. 2011, vol. 7, no. 2, pp. 225-236.

[16] Kotiev G.O., Miroshnichenko A.V., Stadukhin A.A., Kositsyn B.B. / Determination of mechanical characteristics of high-speed tracked vehicles traction motor with individual drive wheels (2019). Journal of Physics: Conference Series, 1177 (1), article № 012058.

[17] Analysis on operational power and eddy current losses for applying coreless double-sided permanent magnet synchronous motor/generator to high-power flywheel energy storage system / Jang S.M., Park Ji.H., You D.J. et al. Journal of Applied Physics. 2009. vol. 105, no 7. pp. 07F116.

[18] Shen S.; Veldpaus F.E. Analysis and control of a flywheel hybrid vehicular powertrain. IEEE Transaction on Control Systems Technology, 2004, vol. 12, pp. 645–660.

[19] Methods for road microprofile statistical data transformation / Evgeniy Sarach, George Kotiev and Sergey Beketov. MATEC Web of Conferences. 2018, v.224, article № 04009.

[20] Korsunskiy V.A. Mathematical model of transport vehicle with combined power installation, flywheel energy storage and stepped transmission. Journal of advanced research in technical science, 2019, no. 14, vol. 2, pp. 78–88. https://doi.org/10.26160/2474-5901-2019-14-190-196.