Determination of the number of speeds of a mechatronic transmission of a road car based on a typical specified driving mode

Valeriy Grinin¹*, Evgeniy Shkarupelov¹, Aleksander Kartashov¹, Ruslan Gazizullin¹, and Sergey Nazarenko²

¹Bauman Moscow State Technical University, 105005 Moscow, Russia
²PJSC "KAMAZ", Transport Passage, 420000 Naberezhnye Chelny, Republic of Tatarstan, Russia

Abstract. The article discusses the problem of determining the rational number of gears for a mechatronic transmission of a truck vehicle. A review of the designs of existing mechatronic transmissions from leading manufacturers of motor vehicles was carried out, with an emphasis on analyzing the number of gears used in them. A method has been developed for determining the rational number of gears based on simulation modeling of the movement of a truck along driving cycles, synthesized on the basis of statistical data on the movement of trucks across the territory of the Russian Federation. A synthesized driving cycle and a simulation model of truck movement, designed to determine the loads on the electromechanical transmission when moving along this cycle, are considered. Based on the results of simulation modeling of the movement of a truck with a different number of gears in a mechatronic transmission, conclusions about the influence of multi-speed transmissions on the energy efficiency of electric truck vehicle were drawn. Conclusions have been formulated about the required number of gears for mechatronic transmissions of trucks moving on the roads of the Russian Federation.

1 Introduction

One of the main trends in the development of truck industry is the desire to reduce the detrimental impact products of vehicle exploitation on the environment.

The main source of pollution during vehicle operation is the chemical compounds present in exhaust gases. The greatest harm from exhaust emissions is observed in large cities with heavy traffic, as well as on federal highways used for intercity truck transportation [1].

The most effective measure to reduce exhaust emissions is the conversion of trucks to electric traction. According to the concept for the development of production and use of electric vehicles in the Russian Federation until 2030, the number of Russian electric vehicles produced on the domestic market should be at least 10 percent of the total production of vehicles. Consequently, the production of the most advanced electric vehicles requires the development of modern technical solutions in the development of electric traction drives of...
vehicles. The most widespread in the designs of electric-powered trucks are mechatronic transmissions. The mechatronic transmission is drive axle with a built-in traction motor, gearbox, axle differential and traction inverter located in one body. The use of mechatronic transmissions makes it possible to completely refuse the application of internal combustion engines in the design of trucks.

A well-known problem of electric transport is the small power reserve of cars of this type, therefore, the greatest achieving energy efficiency of mechatronic transmissions is the main task in the design. When choosing the kinematic scheme of a vehicle’s mechatronic transmission, the most pressing question is the number of gears required to ensure the traction and speed indicators.

In the first approximation, the number of gears and the corresponding gear ratios of mechatronic transmissions are selected according to traction-dynamic calculations from the condition of ensuring intensive acceleration of the car and the fullest use of the power of the propulsion system. Nevertheless, this method of determining the number of gears does not allow us to assess the energy efficiency of the transmission and the motor speed from one or the other gear, which, as a rule, in conditions of small layout space and complexity of the constructive implementation of a mechatronic transmission with a selected suboptimal number of gears can lead to a decrease in the power reserve of a truck, the reliability of a mechatronic transmission, an increase in mass-overall indicators of the transmission of the vehicle.

To assess the most commonly used number of gears in designs, an analysis of the kinematic diagrams of mechatronic transmissions of trucks from leading global manufacturers, such as ZF, ArvinMeritor, Allison Transmissions, AxleTech, AVL, Dana, Magna, GKN, Schaeffler, and also an overview of Chinese manufacturers: Shaanxi HanDe Axle, Ningde Contemporary Electric Technology, Hangzhou Contemporary e-Drive Technology, Rawsun Technology (Shantou), Longyan Michael Machinery, BROGEN, was carried out. The dependence of the number of gears on the frequency of occurrence in the considered mechatronic transmissions of trucks is shown in Figure 1.

![Fig. 1. Application of the number of gears in foreign mechatronic transmissions of truck vehicles](image)

The diagram shows that there is no consensus on the required number of gears for mechatronic transmissions among global automakers: in foreign mechatronic transmissions, one or two gears are most common, three and four are less common, the tendency to use five or more gears is currently in open sources not traceable.
The main uncertainty lies in the fact that today there is no unambiguous method that can reliably determine the required number of gears of a mechatronic transmission. Classical methods for determining the number of gears for a transmission with an internal combustion engine are not applicable for calculating electromechanical transmissions, since the traction motor has a wider range of rotation speeds than an internal combustion engine and is capable of operating in short-term overload modes, allowing it to achieve the required traction and speed indicators using fewer gears.

Frequently, in the case of insufficient power of the traction electric motor to ensure the required traction and speed characteristics of the vehicle, a more effective solution is to use a more powerful and large-sized traction electric motor instead of complicating the design by increasing the number of gears.

The absence of an established methodology for determining the rational number of gears can lead to an incorrect choice of traction motors used, an increase in the mass-dimensional parameters and a decrease in the efficiency of the transmission.

During designing mechatronic transmissions of trucks intended for operation on the roads of the territory of the Russian Federation, characterized by a long length and high permissible speeds, the task of improving the energy efficiency of the mechatronic transmission and determining the required number of gears becomes one of the most important [2, 3]. The lack of consensus among leading manufacturers and the use of their production experience does not allow us to reliably determine the required number of gears for mechatronic transmissions. To obtain the most accurate results when choosing the required number of gears, it is advisable to use vehicle driving cycles synthesized on the basis of statistical data on the movement of trucks moving on the territory of the Russian Federation.

In order to create similar driving cycles for vehicles, natural tests and the collection of statistical data on the movement parameters of PJSC "KAMAZ" production trucks with an axle load of 11.5 tons were carried out [4-6]. In the current study, a driving cycle synthesized using the k-means clustering method was applied to determine the rational number of gear shifts [7-9]. The synthesis of the driving cycle involved refining the average road gradients and vehicle accelerations. The resulting cycle, which allows for a more accurate assessment of the movement of truck vehicles, encompasses various conditions and driving modes both in the city and on highways in the Russian Federation.

The object of study in this work is a medium-tonnage truck KAMAZ "KOMPAS 12", shown in Figure 2, having a 4x2 wheel formula, with a total weight of 11.9 tons, and an electromechanical transmission designed for an axle load of 8 tons.

![Fig. 2. Electric truck vehicles KAMAZ «KOMPAS 12»](image)
2 Mathematical model of a truck with an electromechanical transmission.

An imitation model of the movement of a truck vehicle has been developed for the analysis of vehicle motion, calculation of transmission parameters necessary for selecting an optimal number of gears, and further analysis of the obtained data. The main advantages of using an imitation model include the ability to quickly change the parameters of the mechatronic transmission and the vehicle as a whole, simulate movement based on vehicle driving cycles, analyze the performance of the gear shifting system, and assess the efficiency of the traction electric motor in various modes under different loads. This allows determining the optimal number of gears and analyzing the energy efficiency of the electromechanical transmission based on specified criteria.

During the creation of the model, the following assumptions were made:
- the model only considers straight-line motion of the vehicle.
- the model is based on a simple scheme of planar motion of a truck, called the "bicycle" scheme.
- damping and elastic elements of the suspension are not taken into account. The basis of the simulation model is a mathematical model of the flat motion of a wheel vehicle with a rear drive axle. The calculation scheme of the motion of the KAMAZ "KOMPASS 12" electric truck is presented in Figure 3.

![Fig. 3. Calculation scheme for the movement of a wheeled vehicle: $P_w$ – force of air resistance; $P_a$ – inertia force of a wheeled vehicle; $R_d$ – forces of interaction of wheels with the support base; $R_{c1}, R_{c2}$ – normal reactions at the point of contact of the wheels with the support surface; $M_{fr}, M_{f2}$ – Rolling resistance moment; $M_k$ – Torque applied to the rear axle; $\alpha$ – elevation angle; $X_k$ – distance from the center of mass to the front axle of the truck; $h_k$ - height of the center of mass of the truck; $L$ – wheelbase of a truck vehicle; $G$- total weight of the machine](image)

Mathematical model of truck is described by the following system of equation (1) and (2):

\[
\begin{align*}
    m \cdot \ddot{V} &= \sum_{i=1}^{2} R_{xi} - G \cdot \sin \alpha - P_w \\
    J_{ki} \cdot \ddot{\omega}_i &= M_k - M_k^f
\end{align*}
\]

where $m$ - weight of the truck vehicle; $\ddot{V}$ - longitudinal acceleration of the center of mass of the truck; $J_{ki}$ - the moment of inertia of the $i$-th wheel; $\ddot{\omega}_i$ - angular acceleration of the $i$-th
wheel; \( M_{f_i} \) - rolling resistance moment of the \( i \)-th wheel; \( M_i^f \) - the moment of resistance to wheel rotation caused by hysteresis losses during rolling and interaction with the support surface

\[
\begin{align*}
R_{z1} + R_{z2} &= G \cdot \cos \alpha; \\
R_{z2} \cdot L &= (P_a + P_w + G \cdot \sin \alpha) \cdot h_c - G \cdot X_k \cdot \cos \alpha + \sum_{i=1}^{2} M_{f_i}
\end{align*}
\]

(2)

The slipping coefficient \( S_{si} \) is determined by the formula (3)

\[
S_{si} = \left| \frac{w_i \cdot r_k - V}{\max(w_i \cdot r_k, V)} \right|
\]

(3)

where \( r_k \) – rolling radius of the wheel in the driving mode; \( w_i \) – angular velocity of the \( i \)-th wheel.

The sliding friction coefficient \( \mu_s \) is calculated by the formula (4)

\[
\mu_s = \mu_{s_{\alpha_{\max}}} \cdot \left( 1 - e^{-\frac{S_{si}}{S_0}} \right) \cdot \left( 1 - e^{-\frac{S_{si}}{S_1}} \right)
\]

(4)

where \( S_0 \) и \( S_1 \) -- constants defining the shape of the curve; \( \mu_{s_{\alpha_{\max}}} \) - coefficient of friction of full sliding for a given angle of rotation.

The initial data of the car used in the simulation of motion in the developed simulation model are presented in Table 1.

<table>
<thead>
<tr>
<th>Name of the parameter</th>
<th>Meaning</th>
</tr>
</thead>
<tbody>
<tr>
<td>Gross vehicle weight, kg</td>
<td>12200</td>
</tr>
<tr>
<td>Frontal area of the vehicle, m²</td>
<td>4.62</td>
</tr>
<tr>
<td>Rolling radius, m</td>
<td>0.843</td>
</tr>
<tr>
<td>Vehicle track, m</td>
<td>1,927</td>
</tr>
<tr>
<td>Vehicle height, m</td>
<td>3.0</td>
</tr>
<tr>
<td>Vehicle base, m</td>
<td>4.70</td>
</tr>
<tr>
<td>Vehicle center of mass distance to the front axle, m</td>
<td>2.35</td>
</tr>
<tr>
<td>Center of gravity height, m</td>
<td>0.90</td>
</tr>
<tr>
<td>Maximum power of the traction motor, kW</td>
<td>143</td>
</tr>
<tr>
<td>Rated torque of the traction motor at a frequency of 2000 rpm, N * m</td>
<td>310</td>
</tr>
<tr>
<td>Road-ground conditions:</td>
<td></td>
</tr>
<tr>
<td>- Rolling resistance coefficient</td>
<td>0.018</td>
</tr>
<tr>
<td>- Maximum coefficient of adhesion</td>
<td>0.7</td>
</tr>
<tr>
<td>- Constants defining the shape of the curve S0/S1</td>
<td>0.1/0.2</td>
</tr>
</tbody>
</table>

This study considers one, two, three, and four-stage mechatronic transmissions. Traction calculations were conducted for each transmission, resulting in preliminary gear ratios and a dynamic factor that determines the traction-speed performance of the vehicle. It is worth noting that in order to achieve equivalent traction-dynamic properties for the "KOMPAS 12" truck using a single-speed transmission, a more powerful and larger traction electric motor
is required compared to the one currently used (Table 1). The maximum power of such a traction electric motor used in a mechatronic transmission with a single gear corresponds to 290 kW.

The dynamic factor $D$ was estimated according to the following formula:

$$D = \frac{P_T}{G} = \frac{P_e - P_w}{G},$$

(5)

where $P_T$ – free traction force on the wheels of the car; $P_e$ – circumferential force on wheels.

The results of the traction calculation are shown in Table 2 and Figures 4-7.

**Table 2.** Electromechanical transmission parameters

<table>
<thead>
<tr>
<th>Number of transmission gears</th>
<th>Gear ratio</th>
<th>Efficiency factor, %</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>I</td>
<td>12,5</td>
</tr>
<tr>
<td></td>
<td>II</td>
<td>–</td>
</tr>
<tr>
<td></td>
<td>III</td>
<td>–</td>
</tr>
<tr>
<td>Two</td>
<td>I</td>
<td>26,5</td>
</tr>
<tr>
<td></td>
<td>II</td>
<td>14</td>
</tr>
<tr>
<td></td>
<td>III</td>
<td>–</td>
</tr>
<tr>
<td>Three</td>
<td>I</td>
<td>26,5</td>
</tr>
<tr>
<td></td>
<td>II</td>
<td>17,5</td>
</tr>
<tr>
<td></td>
<td>III</td>
<td>–</td>
</tr>
<tr>
<td>Four</td>
<td>I</td>
<td>26,5</td>
</tr>
<tr>
<td></td>
<td>II</td>
<td>20,5</td>
</tr>
<tr>
<td></td>
<td>III</td>
<td>11,5</td>
</tr>
</tbody>
</table>

![Fig. 4. Dynamic factor of the «KOMPAS 12» truck with reducer](image-url)
Fig. 5. Dynamic factor of the «KOMPAS 12» truck with a two-speed gearbox

Fig. 6. Dynamic factor of the «KOMPAS 12» truck with a three-speed gearbox

Fig. 7. Dynamic factor of the «KOMPAS 12» truck with a four-speed gearbox
As a result of traction-dynamic calculations, transmission ratios were selected that can provide the necessary traction forces and speed indicators for each considered gear set. This ensures:
- maximum driving speed of 100...120 km/h;
- stable movement at 70 km/h during prolonged uphill drive, corresponding to 5%;
- ability to drive on higher gears during short uphill drives, corresponding to 13%;
- ability to overcome a critical uphill slope corresponding to 29%.

In addition, large areas of transmission overlap implemented on a dynamic factor (Figure 5-7) allow for a reduction in switching frequency, increased driving comfort, and efficient adaptation of gear shifts based on load.

However, at this stage, it is only possible to preliminarily estimate the rational number of gears, as the calculation of the traction-dynamic properties of the vehicle does not take into account the frequency of gear shifts and does not allow to assess the performance of the powertrain throughout a mixed driving route.

The gear shift maps are used for the operation of the transmission gearbox [10-15]. A shift map is a set of curves used by the gearbox to select the appropriate gear for the current vehicle speed and corresponding position of the accelerator pedal within a given time interval of the driving cycle. The following assumptions were made during the development of the shift maps:
- the graphical construction of the shift maps is based on the calibration and tuning experience of engineers in a heuristic mode;
- the shift map model takes into account only sequential gear shifting.

The optimization criterion for the graphical representation of maps was aimed at minimizing the number of shifts and improving the transmission efficiency for individual gears throughout the driving cycle. As a result of the iterative tuning process, shift maps for two-, three-, and four-speed electromechanical transmissions of the truck vehicle were developed and are presented in Figures 8-10.

![Gear shift map for two-speed transmission](image)

*Fig. 8. Gear shift map for two-speed transmission*
Fig. 9. Gear shift map for three-speed transmission

Fig. 10. Gear shift map for four-speed transmission

The optimized driving cycle of a truck, synthesized using the micro-trip method with k-means clustering [5], applied for modeling the motion of a truck with an electromechanical transmission, is shown in Figure 11.
The selection of a rational number of gears was based on several criteria obtained during the simulation modeling of the movement of an electric truck along the considered driving cycle: the energy efficiency of the traction electric motor, the optimal distribution of the vehicle's movement time on different gears throughout the entire driving cycle, the number and frequency of gear shifts.

The energy efficiency assessment was carried out according to the perfect useful work of the traction motor $A_e$:

$$A_e = \int_0^t N_e \cdot dt,$$

where $N_e$ – the power of the electric motor, which is calculated by the formula:

$$N_e = M_e \cdot \omega_e,$$

where $M_e$ – torque on the rotor shaft of the electric motor; $\omega_e$ – angular velocity of rotation of the rotor shaft.

The frequency of gear shifts can be visually estimated by the graphical representation of gearbox operation in the driving cycle section shown in Figures 12-14. The results of the calculation of criteria for assessing the rational number of gears are presented in Table 3.
Fig. 12. Graphic representation of the operation of a two-speed gearbox in the considered section of the driving cycle

Fig. 13. Graphic representation of the operation of a three-speed gearbox in the considered section of the driving cycle
Table 3. Results of calculating criteria for assessing the rational number of gears

<table>
<thead>
<tr>
<th>Name of the parameter</th>
<th>One gear</th>
<th>Two gear</th>
<th>Three gear</th>
<th>Four gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>Completed work $A_e$, kJ</td>
<td>578760</td>
<td>581622</td>
<td>587664</td>
<td>593865</td>
</tr>
<tr>
<td>Number of switches</td>
<td>–</td>
<td>136</td>
<td>272</td>
<td>492</td>
</tr>
<tr>
<td>Distribution of the driving time of an electric truck on separate gears, %</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>I</td>
<td>100</td>
<td>4.9</td>
<td>2.3</td>
<td>1.8</td>
</tr>
<tr>
<td>II</td>
<td>–</td>
<td>95.1</td>
<td>24.7</td>
<td>7.2</td>
</tr>
<tr>
<td>III</td>
<td>–</td>
<td>–</td>
<td>73</td>
<td>34</td>
</tr>
<tr>
<td>IV</td>
<td>–</td>
<td>–</td>
<td>–</td>
<td>57</td>
</tr>
</tbody>
</table>

The obtained data clearly indicate that increasing the number of gear shifts in the transmission results in an enhance in the amount of energy consumed for overcoming the distance by more than 2.5% for a multi-gear transmission compared to a single gear, which negatively affects the energy efficiency of the car and leads to a decrease in the driving range. It can also be noticed that in a mechanical transmission with a four-speed gearbox, there is a much larger number and frequency of gear shifts compared to two- and three-speed gearboxes, which is accompanied by additional losses in the shifting mechanisms that are not taken into account in this model.

When considering the results obtained from simulating the movement of an electric truck on the optimized driving cycle and modeling the operation of the gearbox using shift maps, it can be observed that each gear of the multi-speed mechatronic transmission corresponds to an optimal time distribution, and all gears are used efficiently.

4 Conclusion

According to the data obtained from the rational choice study on the number of gears in a mechatronic transmission using a driving cycle, the following conclusions have been formulated:
1. Having a large number of gears in a mechatronic transmission is not an optimal solution, as it leads to a higher number of control elements in the gearbox, gear meshes, and bearing supports. Additionally, increasing the number of gears results in more frequent shifting, leading to increased energy consumption in overcoming the route. Often, this leads to a reduction in the lifespan and increased wear of transmission components, negatively impacting energy efficiency, vibrational load, metal consumption, and transmission reliability.

2. It has been found that when a truck electric vehicle with a multispeed transmission drives on the evaluated driving cycle, higher gears are predominantly used. However, for electric trucks driving in both city conditions with low speeds and high, irregular loads, as well as on highways characterized by high speeds, it is advisable to use a single- or dual-speed electromechanical transmission that possesses sufficient traction and speed properties for mixed routes.

3. The use of a single-speed transmission requires selecting a more powerful traction electric motor to achieve the required traction and dynamic properties of the considered truck vehicle.

4. It has been established that the application of a two-speed mechatronic transmission is the most rational when using a low-power electric motor and the need to implement the required traction and speed properties.

Acknowledgements

This research work was financially supported by the Ministry of Science and Higher Education of the Russian Federation under agreement No. 075-11-2022-015 dated 07.04.2022 with PJSC "KAMAZ" under the complex project "Creation of high-tech production of mechatronic transmissions for promising KAMAZ trucks and buses with electrical energy storage devices and hydrogen fuel cells" in terms of R&D and technological works.

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