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Research paper

Assessment of the Transmission's Impact on the Operational Properties of Land Vehicles

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Abstract

The article considers the issue of the transmission's effect on the operational properties of the land vehicles. The main transmission parameters are given in the work. The analysis of the possibilities of using different types of transmission is shown. The technique for selecting the number of gears and the mean value of the capacity utilization factor is considered. The technical and economic assessment is given on the basis of complex factors of the specific productivity of the road train and the prime cost of its use.

Keywords: *Electromechanical transmission; Gear ratio; Gearbox; hydromechanical transmission; hydrostatic transmission.*

1. Introduction

Transmission affects the most important performance indicators: traction-speed properties, fuel economy, environmental compatibility and off-road performance. In addition, the transmission affects the performance properties not related to movement: strength, durability, adaptability to maintenance and repair.
Transmission is a combination of aggregates and mechanisms transmitting the engine torque to the leading wheels and changing the torque and angular velocity (rotational velocity) in magnitude and direction.
A.N. Ostrovtsev [1] notes that the design of any aggregate or system of a car should be of interest to a designer primarily in terms of the features and improvement of the work processes since the performance of a car depends on them.

2. Transmission Parameters

Transmission parameters' selection is aimed at providing a car with maximum traction and economic properties for a variety of road and ground conditions. The source data for the selection of transmission parameters are: load capacity, speed, etc. (mentioned in the technical specifications), the expected road and ground conditions, the requirements for unification with mass production of cars, the (preliminary) decisions on the parameters of a propulsion, suspension, engine, as well as geometric parameters of the vehicle's off-road performance [2].
The calculation of the transmission parameters includes: the determination of the kinematic and power ranges of a car, the range of gear ratios, the choice of the transmission scheme (type of the mechanisms making up the transmission, and their placement), the number of gears, the calculation of gear ratios of all transmission units, the determination of the type and layout of the differentials.
When choosing the transmission scheme and the type of its main aggregates it is recommended to consider the following factors. Axle transmission is well combined with a conventional suspension and a framed carrier system. It is advisable for it to be used with a simple car design and when there is a need to ensure wide unification with mass production cars. H-shaped transmission is usually chosen in the development of original cars, without wide unification with mass production axle cars. It is advisable for it to be used with a conventional suspension of all wheels, a bearing body, front, and rear steered wheels. Aggregates with a variable gear ratio in both of these schemes are the gearbox and transfer gears. In the axle scheme, the main gears and final or wheel reduction gears have the constant gear ratio, in the H-shaped scheme, the final or wheel reduction gears have the constant gear ratio.

3. Selecting the Gearbox Type (Type of Transmission)

The selection of the type of a gearbox (transmission) is usually based on the following factors. Stepped gearbox with fixed rotation axes is the simplest and has the smallest dimensions and mass. Stepped planetary gearbox, switched by friction or electromagnetic devices, provides higher traction and dynamic properties. This is due to the possibility to change the gear without an interruption in torque delivery. Planetary gearboxes are recommended to be applied in remote and automatic control.
Hydromechanical gearboxes, consisting of a torque converter and stepped gearbox, provide an automatic continuous change of the gear ratio in a given range. It makes a vehicle handling easier, increases the smoothness of the torque change, increases the capacity utilization factor of the engine, which, in turn, increases the vehicle's off-road performance on soft soils and when overcoming obstacles. The torque converter also provides the ability to move at a very low speed, increases the reliability of the car when working under stressful off-road conditions. At the same time, the efficiency of hydromechanical transmissions is less than that of mechanical stepped ones, and the engine braking ability is lower. The mentioned drawbacks are significantly reduced as the torque con-

verter is automatically locked in the higher gears and under the braking condition.

Hydrostatic transmissions are not automatic but allow regulating the gear ratio continuously according to any law in a relatively wide range. The hydrostatic transmission is able to replace all transmission mechanisms and gives great layout advantages, especially for multi-linked road trains with all driving wheels, since hydraulic motors can be placed at any distance from the hydraulic pump.

Electromechanical transmissions provide an automatic stepless change of gear ratios, which significantly facilitates the handling and improves the off-road performance of the car. With the use of electromechanical transmission, it is possible to obtain an optimum torque distribution over the driving wheels in accordance with the rolling mode and the ground adhesion value. The possibility of electric motors to be placed on wheels and large layout capabilities of electromechanical transmissions make them particularly suitable for heavy road trains with all driving wheels. The disadvantages of electric transmissions are a large size and mass, low efficiency, higher cost.

4. Selecting the Number of Transmissions and the Mean Value of Capacity Utilization Factor

The gear ratio of the transmission is:

$D_{tr} = i_{max}/i_{min}$.

The maximum gear ratio must meet two requirements: 1) to ensure the adhesion of wheels under all intended road and ground conditions; 2) to ensure the stable movement of the car with a given low speed.

The first requirement is obvious, the second one is for the vehicles intended for the use under super-severe conditions, as well as for cars with increased requirements for maneuvering in a confined area.

If the use of a torque converter or a hydraulic clutch is assumed in the transmission, the value of i_{max} is determined without taking into account the second requirement. Since these mechanisms ensure the possibility of stable movement with an arbitrarily low speed.

Based on the first requirement, $i_{max} = \varphi_{max} G_a K_{ad} r_{k0} / M_{E\,max}\, \eta_t$. The maximum value of the coefficient of adhesion can be equal to φ_{max} = 0,8 - 1,0. A greater value is possible when overcoming solid threshold obstacles with comparatively sharp edges and with reduced air pressure in tires.

To provide the second requirement, the following inequality must be fulfilled

$i_{max} \geq 0{,}337 n_{E\,min}\, r_{k0}/\upsilon_{a\,min}$,

where $n_{E\,min}$ is the minimum possible engine crankshaft speed.

The minimum gear ratio of the transmission is usually selected considering the possibility for moving at a given maximum speed.

$i_{min} = 0{,}337 n_{E\,max}\, r_{k0}/\upsilon_{a\,max}$.

The dividing of the range of gear ratios according to the steps is carried out based on the requirements for obtaining high average speed, high traction properties, high fuel economy, good dynamic properties. For off-road traffic conditions, the first two are the main requirements.

To fulfill the first requirement, it is necessary to ensure the highest speed of movement at each section of the road, characterized by a value ψ. The maximum possible speed is defined by the expression

$\upsilon = \eta_t N_{e\,max} v/(G_a \psi + P_\omega)$,

where v is the engine capacity utilization factor.

The equation shows, that in order to obtain a possibly higher velocity, it is necessary to ensure high values of η_t and v. If to consider that the efficiency depends on the value of power used $\eta_t = v\eta_{t\,max}$, then

$$\upsilon = v^2 \eta_{t\,max} N_{e\,max}/(G_a \psi + P_\omega). \quad (1)$$

Thus, to ensure a high average speed of the car under various conditions, it is necessary, firstly, to have a high capacity utilization factor v (quadratic dependence). The value of v = 1 can only be obtained by using a continuously variable transmission. Comparative tests of cars with hydromechanical and mechanical transmissions show that the first one, despite the lower efficiency, has a higher average speed by 10-40%, while the difference is greater when the specific engine capacity is lower. With the increase in the specific engine capacity, the ratio of the capacity utilization of the car with the stepped transmission increases due to the reduction in the number of gear changes, and the speed increases accordingly. Therefore, for large engine specific capacities, the use of continuously variable transmissions is less effective.

Assuming that at each of the transmissions the car moves at a designed speed υ_c corresponding to the engine running at a frequency of n_N, and using the expression $G_a\psi + P_\omega = \psi_Д G_a$, the equation (1) can be transformed:

$$\psi_E = v^2 \eta_{t\,max} N_{e\,max}/(G_a \upsilon_c), \quad (2)$$

where $\eta_{t\,max} N_{e\,max}/G_a \upsilon_c = \psi_{E\,max}$ for each transmission is a constant value.

Consequently, at each transmission, the car is able to move at a given speed along the roads with $\psi_E = 0 - \psi_{E\,max}$, while the capacity utilization factor varies within the range v = 0...1. If the resistance exceeds $\psi_{E\,max}$, the next lower gear is shifted. As a result of the downshift, with the same specific traction force, the capacity utilization factor will decrease from 1 to 0 due to the increase in the transmission ratio and the corresponding reduction in speed. In this case, it becomes possible to increase ψ_E to $\psi_{E\,max}$ as the value of v increases from 0 to 1. While $\psi_{E\,i} = \psi_{E\,(i+1)}$

$$v_{0i} = \frac{\sqrt{\psi_{E\max(i+1)}}}{\psi_{Ei}} \quad (3)$$

By multiplying the left and right sides of equation (3) for all gears, we have

$$v_{01} v_{02} \ldots v_{0(n-1)} = 1/\sqrt{D_{ad}}. \quad (4)$$

Assume that the average value of v_{0i} is constant for all steps, then we obtain an expression for the average value of the capacity utilization factor

$$v_{0av} = 1/D_{ad}^{\frac{1}{2(n-1)}}. \quad (5)$$

By using equation (5), it is possible to determine the required number of transmissions according to the given value of the capacity utilization factor or solve the inverse problem. It should be considered that as the number of transmissions increases, the capacity utilization factor increases too, which should lead to an increase in the average speed, but it complicates the gearbox design and makes it difficult to control it since the driver is unable to switch too many gears in time. The optimum number of transmissions depends on the range of gear ratios and the specific power of the engine. For modern cars, the optimal number of gears is 3-4, for trucks is 4-6 with a range of 6-11, for tow cars and special-purpose vehicles with a large range of gear ratios, the number of

gears reaches 8-12. The average capacity factor values in the range of 0.80 – 0.85.
After selecting the number of gears and the average value of the capacity factor, adjust the values for each gear. The values of v_0 are increased for running gears and are reduced for not running, while condition (4) [2] must be fulfilled.

5. Technical and Economic Assessment

One of the first issues to be solved when designing a vehicle is the choice of engine and transmission parameters that provide the best performance.
Two main parameters are identified: engine maximum brake power and maximum speed in the top gear. An attempt was made to find their optimal combination for the line-haul trains. These parameters are not interrelated, but the performance of the vehicle largely depends on their consistency [3].
When compiling variants for calculation, the well-known methods of the mathematical theory of experimental design for the purposes of computer calculations were used. The calculations were performed using the CFE matrix (complete factor experiment) of type 3^2, which involves the variation of two factors at three levels.
The basic level of the gear ratio of the leading axles is $U_0 = 5.8$. At the adopted as unchanged parameters of the engine and transmission, it allows developing V_a = 95.38 km/h. The upper level at a speed V_a = 75 km/h corresponds to $U_0 = 7.3$. The lower level of the gear ratio equals U_0 = 4,3 at V_a = 128.6 km/h. The interval of factor variation was 7.3 - 5.8 = 1.5. Checking 5.8 – 4.3 = 1,5.
As the main level of specific power is assumed N_{sp} = 5.9 kW/t. With a variation range of 1.5, the lower and upper levels equal 4.4 and 7.4 kW/t respectively.
As a result of the implementation of the CFE matrix type 3^2, second-order regression equations were obtained. Where the optimization parameters are the average speed or average fuel consumption resulting from the mathematical modeling of the movement of road trains: three-axle or two-axle tractor units when driving on a low-hilly or hilly highway [3].
The problem of determining the optimal combination of the specific power and the gear ratio of the leading axles has a unique solution only when a vehicle is moving along a low-hilly road with a general speed limit of V_a = 70 km/h for cars with a gear ratio of the leading axles less than five. When the increase in specific power leads to an increase in the average speed while reducing the average fuel consumption. In all other cases, the increase in specific power causes an increase in both average speed and fuel consumption and a separate assessment according to the average speed and average fuel consumption does not give an unambiguous answer to the issue posed.
It is necessary to move from a technical area to the technical and economic area and address the methods of integrated assessment, based on the interconnected consideration of high-speed, fuel and cost factors and allow obtaining not only a qualitative but also a quantitative evaluation of the road train variants. The specific performance of the road train and the specific prime cost of its use recommended by RTM 37.031.007-78 were used as complex indicators.
The specific performance of the Wsp road train characterizes its performance efficiency, taking into account the speed and corresponding fuel consumption, and is defined as the ratio of hourly output (t.km/h) to fuel consumption per 100 km (l/100 km). The Wsp index comprehensively reflects the most important technical and economic requirements: labor productivity increase and fuel and energy resource saving.
The specific cost of using the road train S_{sp} is a technical-economic indicator which is an equivalent to the national economic costs and is represented as the ratio of the total hourly production and operation costs to the hourly productivity of the road train. This complex indicator allows choosing the optimal values of the specific power and gear ratio of the leading axles taking into account the cost factor.
For dump trucks, as well as for line-haul trains, it is recommended to choose the optimal power on the basis of a feasibility study. In this case, the criterion for the optimal variant is the total national economy expenditures of Z_{ne} for carrying out a given volume of transport work for a certain time.
The results of the calculations are shown in Fig. 1.

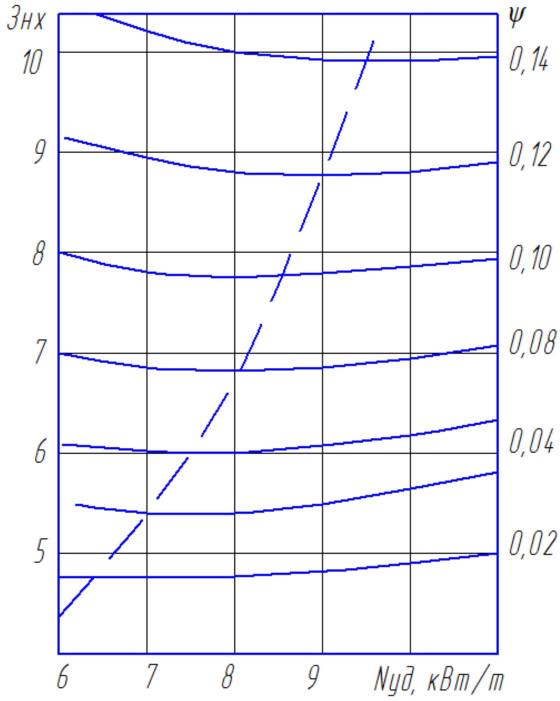

Fig. 1: National economy expenditures, depending on the average road resistance and specific power

The graphs show that for easy operating conditions characterized by the coefficient ψ_{av} = 0,02-0,04, the optimum specific power is within the range of 5.9-7.4 kW/t and then gradually increases with the complication of road conditions.

6. Conclusions

The described technical and economic efficiency depending on the change of a particular parameter of the car is estimated by its impact only on the vehicle under study, and the change in the traffic parameters of other vehicles participating in the transport stream is not taken into account. As the traffic intensity on the road increases, the car is interfered by the overtaken cars or itself becomes interference to the passing cars [3].
The specification of the load regime for the design of the vehicle's transmission and the verification of the reliability of its elements consists of a number of steps. When performing deterministic and probabilistic calculations, the main ones are:
1) Determination of the relative mileage of the car on various transmissions;
2) Determination of load distribution curves for individual transmissions and a common curve;
3) Determination of limits and laws of load variation under different operating conditions;
4) Determination of load mode parameters for an individual unit or part;
5) Determination of the values of maximum dynamic loads.
When performing a deterministic calculation, step 3 is eliminated.

7. Final Conclusion

The most powerful tool for increasing the efficiency of road transport is increasing its bearing capacity. With the limits of permissible loads on the axle of a vehicle, its bearing capacity can be significantly increased only by increasing the number of axles. This fact explains the increasingly widespread use of multiaxial multi-bearing vehicles and road trains [4].
This problem is particularly acute in the Russian Federation, where the majority of roads are designed for an axial load of 6 tons. Therefore, KamAZ vehicles are designed with a 6x4 axle configuration. Countries with a developed road network with an allowable axle load of 10 tons or more use vehicles and tow cars of road trains with an axle configuration of 4x2. Multi-axle vehicles have a rather complex transmission, which leads to a kinematic mismatch between the geometric constraints of power flow.
The kinematic discrepancy, depending on the causes of its occurrence, can be divided into three groups:
1) Kinematic discrepancy of constructive and operational origin, due to the condition and design of the vehicle. It includes discrepancy caused by the different radii of rolling of the wheels, the inequality of gear inches, etc.;
2) Kinematic discrepancy due to the curvilinear motion of the vehicle in the horizontal plane;
3) Kinematic discrepancy caused by the profile of the road, road macro-friction, obstacles.
The magnitude of the kinematic discrepancy of constructive-operational origin can be determined by the formula

$$\chi_{mk} = \left(1 - \frac{r_k^c i_m}{r_k^c i_k}\right) 100\%,$$

where r_k^c, r_m^c - radii of rolling of wheels in a conducted mode (M_k = 0); i_k, i_m - gear ratios of wheel drives.
The difference between the radii of rolling of wheels varies within the tolerance for the manufacture of tires, and depending on the size of the wheels equals 8-15 mm. The maximum value of the kinematic discrepancy will reach 1.5-2% [3, 5].
Increasing the number of transmissions with traditional methods drastically increases the complexity and metal consumption of the units. The new methods for the designing of transmission aggregates are necessary, which will allow creating competitive, patent-protected multistage aggregates of small metal capacity and meeting the requirements [4].

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