A method for selecting parameters of the electromechanical transmission of an industrial tractor

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Abstract. The article describes a method for selecting the torque — speed characteristics of the drive motors and parameters of the electromechanical transmission of an industrial tractor. In order to provide the maximum performance of the tractor the transmission developed in the research has two gears: transportation range and technological range. The distinctive feature of the transmission is the use of the ZK tank steering system developed by the researchers of BMSTU (G.I. Zaychik, M.A. Kreynes, M.K. Kristi). The authors have studied the impact of the design parameter of the planetary gear sets of the ZK mechanism on the tractor performance. The torques and drive motor rotor speeds providing required performance in a turn as a functions of the planetary gear design parameter have been calculated. Also the time of the tractor full turn around its center of mass and around one braked track when the transmission operates in the transport and technological ranges has been found. An finally, the impact of the tractor transmission parameters on the maximum tractor speed at manoeuvring has been analysed. The results of the analysis were used for selection of the electric motor output power providing a comfortable work for the tractor operator.

1. Introduction

An industrial tractor is used for performing working operations involving high tractive efforts [1]. However the tractor operation efficiency depends not only on the speed of the working maneuvers execution but also on the speed of its transfer between the sites in the transport mode.

This requires the tractor transmission to have at least two gears: a working gear and a transport gear. Besides, when the transmission is operating in each gear, the tractor should be able to move without shifting gears and to turn smoothly in order to provide its operation by drivers with lower professional qualification and to increase the working efficiency.

One of the ways to provide such qualities of the industrial tractor is to use the electromechanical transmission. In the simplest case, the smooth turn of the tracked vehicle can be provided by the use of an individual electric drive of its driving wheels. But in this case during turning in the transport gear, the torque of the drive sprocket of the outside track is not sufficient, which forces the driver to lower the tractor speed and shift into the working gear. In the course of this problem solution, different transmission layouts for tracked vehicles have been analyzed [2], [3], [4], [5], [6].

The solution that meets the specified requirements is the use of the ZK steering system (developed by the researchers G.I. Zaychik, M.A. Kreynes and M.K. Kristi from the BMSTU) providing a smooth turnig and increased tractive performance of the tracked vehicle [7].

The electromechanical transmission to be installed inside the frame of the commercially available tractors should be extremely compact. Figure 1 shows a kinematic diagram of the tractor transmission intended to solve this problem. The kinematic diagram contains identical gear sets which provide a
high reduction in the working gear and the functionality of the ZK steering system during transportation operations. Switching between the modes is carried out by means of four control elements (since it is assumed that there will be no switching between the ranges while driving). Cam clutches can be used as the control mechanisms.

![Kinematic diagram of the tractor transmission.](image)

**Figure 1.** Kinematic diagram of the tractor transmission.

When the transmission is in the working gear, the brakes T1 and T2 are locked (see figure 1). The motors individually drive the wheels. The gear sets of the ZK steering system are disengaged and operate as the gear sets of the planetary gear train with the stopped central gear (the gear ratio of such a planetary gear is $k_{zd}+1$).

Turning of the tracked vehicle is provided by the high reduction on each track and with the help of the stopping brakes Ts.

When the transmission is in the transport gear, the clutches $M_1$ and $M_2$ are locked (see figure 1) and provide the reduction required for the straight-line motion. When the tractor is turning, the tractive effort on the tracks is provided by the ZK steering system (with the respective decrease in the tractor speed).

2. **Selection of the main parameters of the transmission**

In order to reduce the requirements for the accuracy of the gear manufacturing and accordingly reduce the cost of the transmission, the speed of the drive motor shaft should be limited by the value of 4000 rpm. In this case, the required gear ratio of the transmission top gear (the transport gear) is determined by the maximum speed of the tractor:

$$i_{tr} = \frac{n_{max}}{V_{max}},$$

(1)

where $n_{max}$ is the maximum speed of the motor shaft; $r_k$ is the radius of the drive wheel; $V_{max}$ is the maximum speed of the tractor.

When the transmission operates in the transport gear in case of a straight-line motion, all the elements of the ZK mechanism rotate together without torque reduction. Hence, the transmission gear
ratio in the transport gear is defined by the gear ratio of the final reduction gear $i_{fr}$ and motor reductor ratio $i_{em}$ (see figure 1).

As it is known, in the average operation conditions, the need for a tractive effort to weight ratio to be higher than 0.4 is very unlikely [8]. Then, the required maximum torque of the motor is defined in the following way:

$$M_{max} = \frac{mg \cdot D \cdot r_k}{i_{fr} \cdot 2\eta_{fr} \eta_{gar}}$$

where $M_{max}$ is the motor maximum torque; $m$ is the gross mass of the tractor; $g$ is the acceleration of gravity; $\eta_{fr}$ is the mechanical efficiency of the transmission; $\eta_{gar}$ is the mechanical efficiency of the track. $D$ is the tractive effort to weight ratio of the tractor.

For any planetary gear set with negative gear ratio an important parameter is the value $k_{zk} = \frac{Z_{BCK}}{Z_{MCK}}$, where $Z_{BCK}$ is the number of teeth of the ring gear, $Z_{MCK}$ is the number of teeth of the sun gear. Recommended values of the ratio $k_{zk}$ for such a planetary gear set are selected in the range from 4/3 to 4 [9]. The tractive effort to weight ratio and maximum speed of the tractor in the first gear as functions of the ratio $k_{zk}$ of the ZK mechanism planetary gear sets can be calculated in the following way:

$$D_t = \frac{M_{max} \cdot i_{fr} \cdot 2 \cdot \eta_{fr} \cdot \eta_{gar} \cdot (k_{zk} + 1)}{m \cdot g \cdot r_k} = D_s \cdot (k_{zk} + 1),$$

$$V_{max}^{n} = \frac{\pi \cdot n_{max} \cdot r_k}{30 \cdot i_{fr} \cdot (k_{zk} + 1)} = \frac{V_{max}}{(k_{zk} + 1)},$$

where $D_s$ is the tractive effort to weight ratio in the first gear; $V_{max}^{n}$ is the maximum speed of the tractor in the first gear.

Let us consider an industrial tractor as an example. The tractor specifications are given in table 1.

| Parameter                                      | Value   |
|------------------------------------------------|---------|
| Mass, kg                                       | 41500   |
| Maximum tractive effort to weight ratio in the transport gear, minimum | 0.4     |
| Maximum speed in the transport gear, km/h      | 20      |
| Drive sprocket radius, m                       | 0.434   |
| Wheel base, m                                  | 3.3     |
| Track width, m                                 | 2.2     |
| Height of the center of mass, m                | 1.8     |
| Maximum tractive effort to weight ratio in the working gear, minimum | 0.9     |
| Maximum speed in the working gear, km/h, minimum | 3       |
| Maximum mechanical efficiency of the transmission | 0.922  |

The tractor characteristics calculated by formulae 1 and 2 are shown in table 2.

Thus (according to formulas 3 and 4), for the considered tractor, the range of permissible values of the ratio $k_{zk}$ provides a tractive effort to weight ratio from 0.933 to 2, and maximum speeds from 8.6 km/h to 4.0 km/h in the transport mode.
Table 2. Calculated characteristics of the industrial tractor.

| Parameter                              | Value  |
|----------------------------------------|--------|
| Maximum required torque of the drive motor, Nm | 1250   |
| Maximum rotor speed of the drive motor, rpm | 4000   |
| Gear ratio of the transport gear        | 32.76  |

In order to calculate performance characteristics of the tractor we should use the drive sprocket required torque/tractor speed — tractor turning radius curves. In a general case this kind of analysis should be performed taking into account the characteristics of the road irregularities [10] and with the use of the appropriate model of the track shoe — road interaction [11, 12], for instance the model based on the friction ellipse concept [13]. In this study, to reduce the amount of input data, it is advisable to use a simpler solution of this problem [14, 15, 16].

The most suitable method is the one described in [16]. The method consists of the following steps (for the case when the tractor moves in the most severe road conditions — on a turfy soil).

1. Calculation of the theoretical turning radius \( R_t \):

\[
R_t = R_f \cdot \frac{B}{L \cdot k},
\]

where \( B \) is the track width; \( L \) is the wheel base; \( k \) ratio of the actual turning radius to the theoretical turning radius for the given tracked vehicle; \( R_f \) is the actual turning radius.

2. Calculation of the coefficient of lateral resistance \( \mu \) by the A. O. Nikitin formula [17]:

\[
\mu = \frac{\mu_{\text{max}} R_f}{0.925 + 0.15 \frac{R_f}{B}},
\]

where \( \mu_{\text{max}} \) is the maximum coefficient of lateral resistance on the given terrain (in the case of the turfy soil \( \mu_{\text{max}} = 0.85 \) [18]).

3. Calculation of the required track speeds \( V_2 \) and \( V_1 \) of the outside and inside tracks:

\[
V_2 = \frac{V}{1 + \frac{B}{2R_s}}; \quad V_1 = \frac{V}{1 - \frac{B}{2R_t}},
\]

where \( V \) is the speed of the tractor center of mass.

4. Calculation of the required tractive efforts \( P_2 \) and \( P_1 \) on the outside and inside tracks [8]:

\[
P_2 = \frac{m \cdot g}{2} \left[ f_{gr} \left( 1 + \frac{2V^2 \cdot H_z}{B \cdot R_f \cdot g} \right) + \frac{\mu \cdot L}{2B} \left( 1 - \left( \frac{V}{V_{kr}} \right)^4 \right) \right] + \frac{m \cdot V^4 \cdot L}{4 \cdot R_f^3 \cdot \mu \cdot g};
\]

\[
P_1 = \frac{m \cdot g}{2} \left[ f_{gr} \left( 1 - \frac{2V^2 \cdot H_z}{B \cdot R_f \cdot g} \right) - \frac{\mu \cdot L}{2B} \left( 1 - \left( \frac{V}{V_{kr}} \right)^4 \right) \right] + \frac{m \cdot V^4 \cdot L}{4 \cdot R_f^3 \cdot \mu \cdot g},
\]

where \( f_{gr} \) is the tractors straight-line motion resistance coefficient for the given soil (in the case of a turfy soil \( f_{gr} = 0.07 \) [18]); \( H_z \) is the height of the tractor center of mass; \( V_{kr} \) is the skidding critical speed \( (V_{kr} = \sqrt{\mu_{\text{max}} \cdot g \cdot R_f}) \).

5. Calculation of the mechanical efficiency \( \eta_2 \) and \( \eta_1 \) of the outside and inside tracks [19]:

\[
\eta_2 = 0.95 - 0.005 \cdot 3.6V_2; \quad \eta_1 = 0.95 - 0.005 \cdot 3.6V_1.
\]
6. Calculation of the required power $N_2$ and $N_1$ on the sprockets of the outside and inside tracks:

$$N_2 = \frac{P_2 \cdot V_2}{\eta_2}; \quad N_1 = \begin{cases} P_1 \cdot V_1 \cdot \eta_1, & \text{if } P_1 < 0 \\ P_1 \cdot V_1 / \eta_1, & \text{if } P_1 > 0 \end{cases}.$$  

(11)

7. Calculation of the required drive torques $M_2$ and $M_1$ on the sprockets of the outside and inside tracks:

$$M_2 = \frac{P_2 \cdot r_k}{\eta_2}; \quad M_1 = \begin{cases} P_1 \cdot r_k \cdot \eta_1, & \text{if } P_1 < 0 \\ P_1 \cdot r_k / \eta_1, & \text{if } P_1 > 0 \end{cases}.$$  

(12)

Then, based on the data taken from the given characteristics the required torque — speed curve in the transport gear is calculated. The calculation is performed according to the following method:

1. Calculation of the rotation speeds $\omega_2$ and $\omega_1$ for the outside and inside tracks:

$$\omega_2 = \frac{V_2}{r_2}; \quad \omega_1 = \frac{V_1}{r_1}.$$  

(13)

2. Calculation of the rotation speeds $\omega_{e,2}$ and $\omega_{e,1}$ for the drive motors of the outside and inside tracks:

$$\omega_{e,2} = \left((1 + k_{e,2} \cdot \omega_2 - k_{s,2} \cdot \omega_1) \cdot \omega_2\right) \cdot \frac{1}{\eta_{e,2}}; \quad \omega_{e,1} = \left((1 + k_{e,1} \cdot \omega_2 - k_{s,1} \cdot \omega_1) \cdot \omega_1\right) \cdot \frac{1}{\eta_{e,1}}.$$  

(14)

3. Calculation of the required torques $M_{e,2}$ and $M_{e,1}$ of the drive motors of the outside and inside tracks:

$$M_{e,2} = \left(\frac{1 + k_{s,2} \cdot M_2}{1 + 2k_{s,2}} \cdot \frac{1}{\eta_{e,2}}\right) + \left(\frac{k_{s,2} \cdot M_1}{1 + 2k_{s,2}} \cdot \frac{1}{\eta_{e,1}}\right);$$

$$M_{e,1} = \left(\frac{1 + k_{s,1} \cdot M_1}{1 + 2k_{s,1}} \cdot \frac{1}{\eta_{e,1}}\right) + \left(\frac{k_{s,1} \cdot M_2}{1 + 2k_{s,1}} \cdot \frac{1}{\eta_{e,2}}\right).$$  

(15)

4. Calculation of the required torques $N_{e,2}$ and $N_{e,1}$ of the drive motors of the outside and inside tracks:

$$N_{e,2} = M_{e,2} \cdot \omega_{e,2}; \quad N_{e,1} = M_{e,1} \cdot \omega_{e,1}.$$  

(16)

In addition, using the required speeds of the tracks the time spent on the full turn of the tractor around its vertical axis $T$ can be calculated:

$$T = \frac{2 \pi \cdot B}{V_2 - V_1}.$$  

(17)

Figures 2–4 show the torque — speed and output power — speed curves of the drive motor at different turning radii and tractor speed 10 km/h in transport gear on a turfy soil.

![Figure 2](image-url)  

**Figure 2.** Motor torque — tractor turning radius curves at the tractor speed $V = 10$ km/h.
These results show that with the decrease in the parameter $k_{zk}$ of the ZK planetary gear sets. During its operation the tractor makes turns with small radii including full turn about a stopped track and full turn around the tractor center of mass (when this maneuver is possible). In this regard, it is advisable to consider the impact of the parameter on the operation of the transmission when performing such maneuvers in the transport and working gears. The required parameters of the drive motor when the tractor pivots about its stopped track are calculated by the formulae (5 - 17). For the case of the tractor pivoting around its center of mass the speeds $V_1$ and $V_2$ of the tracks and the required tractive efforts $P_1$ and $P_2$ on the tracks are calculated according to the following method. The speed of the drive motors was set to its maximum value $\left[ n_{1,2} = 4000 \text{rpm} \right]$.

1. Calculation of the required track speeds $V_1$ and $V_2$ of the outside and inside tracks during pivoting around the tractor center of mass:

$$\omega_2 = \left( \frac{1 + k_{zk}}{1 + 2k_{zk}} \cdot \omega_{v2} + \frac{k_{zk}}{1 + 2k_{zk}} \cdot \omega_{v1} \right) \frac{1}{l_i}$$

2. Calculation of the required tractive efforts $P_1$ and $P_2$ on the outside and inside tracks during pivoting around the tractor center of mass:

$$\omega_1 = \left( \frac{1 + k_{zk}}{1 + 2k_{zk}} \cdot \omega_{v1} + \frac{k_{zk}}{1 + 2k_{zk}} \cdot \omega_{v2} \right) \frac{1}{l_i}$$

$$V_2 = \omega_2 \cdot n_k; \quad V_1 = \omega_1 \cdot n_k.$$
$$P_2 = mg \cdot \left( -0.5 f_{gr} + \frac{\mu_{\text{max}} \cdot L}{4 \cdot B} \right); \quad R = mg \cdot \left( 0.5 f_{gr} - \frac{\mu_{\text{max}} \cdot L}{4 \cdot B} \right).$$  \tag{21}$$

The calculation results are shown in tables 3–6.

**Table 3.** Parameters of the tractor pivoting about its stopped track in the transport gear.

| Parameter | $k_z$ | 1.33 | 2.00 | 3.00 | 4.00 |
|-----------|-------|------|------|------|------|
| Required torque $M_{e1}$ of the drive motor of the inner track, Nm | -178.6 | -43.7 | 42.0 | 95.0 |
| Required torque $M_{e2}$ of the drive motor of the outer track, Nm | 785.2 | 636.8 | 542.1 | 483.2 |
| Required output power $N_{e1}$ of the drive motor of the inner track, kW | 22.3 | 6.5 | -6.9 | -16.7 |
| Required output power $N_{e2}$ of the drive motor of the outer track, kW | 172.0 | 139.5 | 118.9 | 105.9 |
| Duration $t$ of the tractor full turn, s | 11.1 | 14.6 | 19.0 | 23.8 |

**Table 4.** Parameters of the tractor pivoting about its stopped track in the working gear.

| Parameter | $k_z$ | 1.33 | 2.00 | 3.00 | 4.00 |
|-----------|-------|------|------|------|------|
| Required torque $M_{e1}$ of the drive motor of the inner track, Nm | -626.7 | -473.7 | -364.5 | -291.5 |
| Required torque $M_{e2}$ of the drive motor of the outer track, Nm | 886.0 | 665.7 | 510.2 | 407.0 |
| Required output power $N_{e1}$ of the drive motor of the inner track, kW | 0 | 0 | 0 | 0 |
| Required output power $N_{e2}$ of the drive motor of the outer track, kW | 185.7 | 139.7 | 107.0 | 85.2 |
| Duration $t$ of the tractor full turn, s | 11.6 | 15.3 | 19.9 | 24.9 |

**Table 5.** Parameters of the tractor pivoting about its center of mass in the transport gear.

| Parameter | $k_z$ | 1.33 | 2.00 | 3.00 | 4.00 |
|-----------|-------|------|------|------|------|
| Required torque $M_{e2}(-M_{e1})$ of the drive motor of the outer (inner) track, Nm | 454.0 | 319.3 | 234.1 | 181.5 |
| Required output power $N_{e2}(-N_{e1})$ of the drive motor of the outer (inner) track, kW | 190.2 | 133.7 | 98.1 | 76.0 |
| Duration $t$ of the tractor full turn, s | 4.55 | 6.41 | 8.7 | 11.2 |

**Table 6.** Parameters of the tractor pivoting about its center of mass in the working gear.

| Parameter | $k_z$ | 1.33 | 2.00 | 3.00 | 4.00 |
|-----------|-------|------|------|------|------|
| Required torque $M_{e2}(-M_{e1})$ of the drive motor of the outer (inner) track, Nm | 725.5 | 542.6 | 414.4 | 329.7 |
| Required output power $N_{e2}(-N_{e1})$ of the drive motor of the outer (inner) track, kW | 303.9 | 227.3 | 173.6 | 138.1 |
| Duration $t$ of the tractor full turn, s | 2.89 | 3.88 | 5.0 | 6.2 |
3. In contrast to the rotation with a large turning radius, in the case when the tractor turns around its center of mass and around its stopped track, the parameter $k_z$ has an impact on the required power of the drive motor. The results given in tables 2–5 show that with the increase in the parameter required power and torque of the drive motor decrease and the duration of the tractor full turn rises. When the parameter $k_z$ rises from 1.33 to 4, the duration of the tractor full turn increases by the factor of 2.22 at the average.

### 3. Calculation of the required output power of the drive motors

The maximum power of the drive motor should provide turning maneuvers at the tractor maximum speed limited by comfort conditions of the driver. It is known that the comfort limit of the lateral acceleration $a_y$ is 0.05g [20]. Then, the tractor maximum speed in a turn (limited by the driver comfort) can be calculated as:

$$V_{pr} = \sqrt{a_y \cdot R_f} = \sqrt{0.05g \cdot R_f}.$$ (22)

Next, by the method of successive approximations the maximum power of the drive motor providing maneuvering at a given tractor speed can be calculated (see figure 5).

![Figure 5. Relationship between the tractor maximum speed and its turning radius](image)

Thus, the required power of the drive motor of the tractor in question is 415 kW (the power to weight ratio is 10 kW / t). Besides, in order to minimize the mass and overall dimensions of the drive motor and to provide the maximum efficiency of the steering system (mechanical energy recuperation between the tracks at low turning radii) based on the data in figures 2–5 the value of the parameter $k_z$ should be 2.

Tables 2 and 7 as well as figure 6 show the calculated performance characteristics of the tractor for the selected parameters of the electromechanical transmission.

| Parameter                                      | Value   |
|------------------------------------------------|---------|
| Maximum tractive effort to weight ratio in the working gear | 1.232   |
| Maximum speed in the the working gear, km/h  | 6.5     |
| Working gear ratio                             | 98.27   |
| Duration $t$ of the tractor full turn around its center of mass in the transport gear, s | 6.4     |
| Duration $t$ of the tractor full turn around its center of mass in the working gear, s | 3.9     |
| Duration $t$ of the tractor full turn around its stopped track in the transport gear, s | 14.6    |
| Duration $t$ of the tractor full turn around its stopped track in the working gear, s | 15.3    |
The research results have shown that in order to provide the maximum operation efficiency of the tractor with the limitations imposed by the driver comfort conditions there is no need to increase the power to weight ratio of the tractor drive motors higher than the value of 10 kW/t.

4. Conclusion
The proposed layout of the electromechanical transmission is mostly suitable for the industrial tractors. The transmission layout features planetary gear sets as reduction units when operating in the working gear and the use of the same planetary gear sets as the elements of the ZK steering system in the transport gear. This approach is advantageous in terms of the tractor systems packaging and can provide better performance characteristics for the existing tractors.

The method described in the article has allowed giving recommendations the main parameters of the proposed transmission: parameter kzk of the ZK planetary gear sets is 2, maximum tractive effort to weight ratio in the working gear is 1.2, maximum tractive effort to weight ratio in the transport gear is 0.4.

The research results have shown that in order to provide the maximum operation efficiency of the tractor with the limitations imposed by the driver comfort conditions there is no need to increase the power to weight ratio of the tractor drive motors higher than the value of 10 kW/t.

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