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ANALYSIS OF THE INFLUENCE OF THE INTER-WHEEL DIFFERENTIALS DESIGN ON THE RESISTANCE OF THE CAR CURVED MOTION

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Описано метод і наведені результати аналізу впливу конструкції міжколісних диференціалів і відповідних значень коефіцієнтів пропорційності для додаткового блокуючого моменту на опір криволінійному руху повнопривідного автомобіля по дорогах з твердим покриттям. Це дозволяє з урахуванням результатів дослідження ефективності різних типів міжколісних диференціалів в умовах бездоріжжя вибрати з наявних конструктивних пропозицій найбільш доцільні структури для параметричного синтезу міжколісного диференціалу з внутрішньою автоматичністю. Проведення параметричної оптимізації надасть можливість синтезувати міжколісний диференціал з внутрішньою автоматичністю, який би задовольняв вимогам до тягової прохідності і динаміки машини і водночас не перешкоджав її криволінійному руху.

В процесі моделювання було оцінено вплив структури і параметрів міжколісних диференціалів на витрати потужності, необхідні для руху із заданою швидкістю і кривизною траєкторії, а також на збільшення дійсного радіуса повороту машини.

За результатами моделювання зроблено висновок про можливість створення постійно діючого міжколісного диференціалу з внутрішньою автоматичністю на основі диференціалів, в яких ступінь блокування залежить від квадрату різниці кутових швидкостей півосей. Для цього необхідне проведення оптимізації за коефіцієнтом пропорційності блокувального моменту з урахуванням наведених у роботі обмежень і використанням описаного методу аналізу впливу конструкції міжколісних диференціалів на опір повороту. Це дозволить ефективно експлуатувати повнопривідні колісні машини військового і цивільного призначення, як у важкопрохідних дорожніх умовах, так і на дорогах з твердим покриттям. При цьому процес керування машиною не буде передбачати відволікання водія на керування режимами роботи міжколісних диференціалів, а трансмісія буде виконана без невиправданого ускладнення конструкції

Ключові слова: міжколісний диференціал, диференціал підвищеного тертя, блокувальний момент, енергоефективність, керованість

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1. Introduction

The need for inter-wheel differentials (IWD) arose immediately after the appearance of the first two-wheel drive cars. Most notably, this need was manifested during turning and driving on rough roads. The absence of IWD in such transmissions caused power circulation, unreasonably large additional loads on the drive axle and wheels, increased fuel consumption and high tire wear. With the invention of the traditional symmetric bevel gear IWD, these problems were solved. However, there were other problems associated with the phenomenon of slipping in difficult road conditions. Naturally, this challenge has found a large number of technical solutions that somehow smoothed out the problem, but did not solve it comprehensively.

The simplest and chronologically first solution to combat slipping of one of drive wheels was the complete lock of the IWD on demand (manual control). As a rule, locking is carried out by the gear or jaw coupling. In this case, the process of engaging or disengaging the coupling requires a full stop of the car. In heavy road conditions, this requires the driver to pre-engage the described device. Such a lock ensures the maximum traction power of the wheel drive, since it allows transmitting up to 100% of power to any of wheels that can take it under conditions of adhesion to the support surface. If structurally engagement or disengagement of full lock is made by means of friction coupling, this does not require stopping the vehicle to change the mode. Such a design is relatively easy to automate, but usually entails an increase in the size and weight of the differential and main gear, as well as a decrease in clearance (road clearance). The complete IWD lock must be switched off after getting on a paved or dry dirt road.

The second solution to the problem with slipping was the invention of a whole group of self-locking limited-slip differentials (SLLSD). Depending on design, these can be differentials with an additional torque depending on a load, difference in angular velocities or squared difference in angular velocities of the semiaxles.

In recent decades, design solutions have emerged in which the differential lock is controlled electronically by a given algorithm, as well as drives in which the traction control system directly controls the input torque of the drive wheel. These solutions are especially effective for electric or hydrostatic motor wheels.

However, despite the abundance of technical solutions in the field of differential wheel drive [1–8], there is no effective IWD design for military wheeled and multi-purpose four-wheel drive vehicles. The existing designs either still use complete manual lock, or are based on SLLSD that are not able to provide high cross-country ability and good handling at the same time.

Therefore, despite the rapid development of electronic control systems and individual electric drives, the development of an efficient, internally automated IWD is topical for military wheeled and multi-purpose four-wheel drive vehicles.

2. Literature review and problem statement

The issues of power distribution between wheels and axles in modern cars are given much attention in scientific and technical literature. There is a large number of monographs and educational literature, for example [1–3], describing standard structures of differentials and approaches to their design. Newer design solutions can be found in the review publications [4–7] and fundamental monograph [8]. If in [4–7] practical information is presented on the design of differentials, their advantages and drawbacks, sometimes about the elements of calculation of locking coefficients, applicability in cars, in [8] the scientific approach to the problem is fully developed. In [8], aspects of influence of the front, rear or four-wheel drive and, as a consequence, insufficient, redundant or neutral handling on the applicability of differentials are discussed in detail.

In the series of papers [9–14], an attempt was made to systematize methods and approaches to power distribution between drive wheels in four-wheel drive cars. Thus, in the similar works [9, 11, 13, 14], the efficiency of the basic methods of power distribution between the drive wheels of four-wheel drive vehicles is evaluated: disconnection of drive axles, locking of IWD and inter-axle differentials, braking of the slipping wheel. The main approaches were changes in the transmission structure (possibility to disconnect drive axles) in different road conditions and slowing down of the slipping wheel. However, the change in the transmission structure due to the disconnection of axles leads to a significant change in handling, and slowing down of the slipping wheel by braking off-road leads to additional power losses and intense heating and wear of braking mechanisms. Similar papers [10, 12] consider similar issues, but only in terms of power consumption, without taking into account vehicle handling.

In [15, 16], the options for improving the car dynamics due to various semi-active differentials [15] or freewheel clutches [16] with controlled lock are considered. However, such technical solutions complicate the design and reduce transmission reliability as a whole.

In many works devoted to hybrid and electric transmission cars, for example [17, 18], the issues of power distribution between drive wheels are efficiently solved, but only for an individual electric drive. Unfortunately, it is not possible to apply these techniques to mechanical transmissions.

A fairly large number of publications [19–22] are devoted to the influence of various IWD designs on car handling and stability, as well as methods of modeling this problem. However, for example in [20], the effect of the abstract locking factor on the car stability when going into turn is investigated without reference to the IWD structure. And the majority of other publications, for example, absolutely identical [19, 21, 22], investigate not so much the possibility of reducing the IWD influence on handling, as the effectiveness of combining conventional ABS and ASR active safety systems with controlled SLLSD.

In [23], the author came closest to the problem considered in the proposed work. The author considered the possibility of constructing an IWD based on hydrostatic lock with the dependence of the locking torque on the squared difference in angular velocities of the wheels. However, the author conducted this analysis only for the rear non-steerable axle and improperly simplified the design scheme. This is confirmed by the quote from [23]: «In view of the fact that the actual circular path curvature was set without regard to the steering angle, side reactions and wheel slip angles were not determined».

The considered method was described in [24], but it was focused on the use of piston pumps, which inadequately increased design complexity and cost. The solution described in [25] assumed joint operation of a simple and compact gerotor pump and clutch plate pack closed by pressure generated by this pump. However, the underdeveloped technology for manufacturing cycloidal gear sets and relatively rapid wear of frictional surfaces somewhat worsen the performance of this rather effective technical solution.

The authors of the presented work in [26–28] began a series of publications on scientific support for the development of IWD with hydrostatic locking without friction discs for military and multipurpose four-wheel drive vehicles. Thus, kinematic [26] and power requirements [27] to IWD based on SLLSD are considered, taking into account the operation peculiarities of these vehicles. In [28], the analysis of implementability of the obtained requirements on the basis of standard internal gear pumps was made.

For the final choice of IWD design and parameters, an assessment of their impact on the efficiency and handling of the four-wheel drive vehicle during curved motion is required, which was not found in the desired statement in the literature.

3. The aim and objectives of the study

The aim of the work is to study the influence of IWD design on the curved motion resistance of the four-wheel drive car on paved roads.

To achieve the aim, the following objectives were set:

– to form criteria for assessing the curved motion resistance of the four-wheel drive car by the parameters of fuel consumption and increase in the actual turning radius;

– to develop a mathematical model for registration of the stated criteria of estimation by numerical experiment;

– to obtain numerical dependences of relative power consumption and increase in the actual turning radius due to the locking torque of self-locking IWD of various designs and with different proportionality factors of the locking torque;

– to work out recommendations for further optimization of the search for effective IWD designs.

4. Formation of criteria and formulation of the problem of numerical simulation of the curved motion of the four-wheel drive vehicle

In the process of problem statement, the following criteria were used:

1) To determine the power losses caused by the locking torque in the curved motion, the power required for the motion with a given constant velocity and theoretical turning radius was used. In this case, the theoretical turning radius was determined by the average steering angle, and motion was considered on a horizontal, even and dry asphalt road. Further, the relative excess of this power in comparison with the power necessary for the linear vehicle motion under similar conditions was used.

2) To determine the effect of the locking torque on the actual turning radius, the calculated value of the actual turning radius of the vehicle was used, taking into account the tire slip in the contact spot when moving in the same modes. Further, the relative excess of this radius for SLLSD relative to the theoretical turning radius, calculated without the wheel slip, was used for comparison.

For research and comparison, the following were chosen:

– open standard symmetric bevel gear IWD;

– completely locked IWD;

– IWD, in which locking torque is proportional to the load (torque transmitted by the differential);

– IWD, in which locking torque is proportional to the difference in angular velocities of the semiaxles;

– IWD, in which locking torque is proportional to the squared difference in angular velocities of the semiaxles.

The research was conducted for the BTR-4 8×8 wheeled armored personnel carrier with the constant four-wheel differential drive.

To solve the problem, provisions of the theory of car motion using the mathematical apparatus of numerical integration of the system of second-order differential equations by the Runge-Kutta method with a constant step were used.

Under the conditions of the numerical experiment, the vehicle in the linear motion was accelerated to a given speed due to a smooth increase in the power input. After that, the steering wheels rotated to a certain angle for 2 seconds, and then the vehicle moved along a circular trajectory, taking into account the wheel slip. Provided that the vehicle reached a constant speed of the curved motion, fixing of the power input and actual turning radius was made taking into account the wheel slip. Stable motion at a given speed was reached by gradually increasing the power at the input of the interaxle differential in the transfer case (Fig. 1). Acceleration thus occurred continuously while maintaining a constant four-wheel differential drive. The interaxle differential in the transfer case and through axles were the standard symmetric bevel gear differentials without any locking devices.

The tire in contact with the road was modeled taking into account the coefficients of elastic and inelastic deformation resistance in the radial, lateral and longitudinal directions. Redistribution of support reactions under the wheels took into account the action of longitudinal and lateral forces of inertia in the curved motion. The wheel tilt in the vertical plane from the suspension, steering and roll of the vehicle was not considered in the analysis of its contact with the road. All shafts and gears were assumed to be absolutely rigid and high-frequency torsional oscillations in the transmission were not considered.

For calculations, the necessary data were taken on the example of the BTR-4 wheeled armored personnel carrier, namely:

– vehicle weight m=24 t;

– static wheel radius at normal tire pressure R_w=0.525 m;

– moment of inertia of the wheel with the wheel reduction gear and semiaxle, reduced to the axle shaft gear of IWD $I_{l(r)}$=2.54 kg·m^2;

– coefficient of wheel adhesion to asphalt concrete pavement in the longitudinal and transverse directions φ_m=0.8;

– average coefficient of resistance to the motion on asphalt concrete pavement for all-terrain vehicle tires at a nominal pressure f_m=0.02.

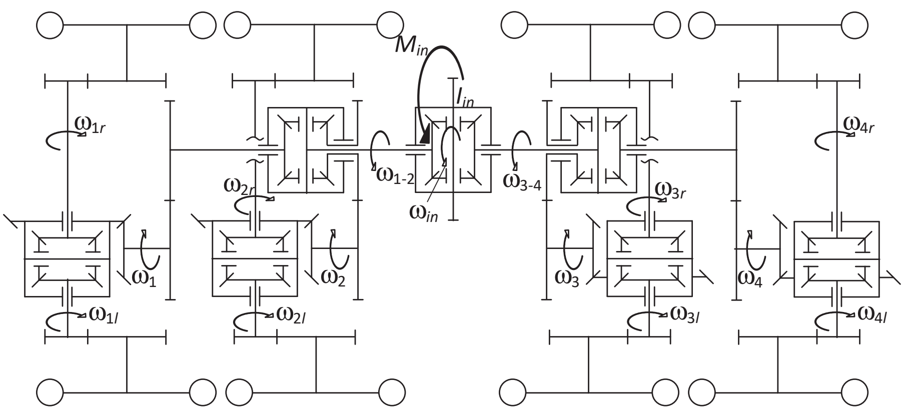

Fig. 1. Kinematic diagram of the BTR-4 wheeled armored personnel carrier transmission

5. Structure of the mathematical model of the curved motion of the four-wheel drive vehicle

The mathematical model was developed on the basis of the D'Alembert principle (Newton's second law), consisting of 39 differential equations with 29 generalized coordinates and 10 generalized velocities (Fig. 1, 2). The generalized velocities were chosen instead of coordinates when there was no need for motion calculations.

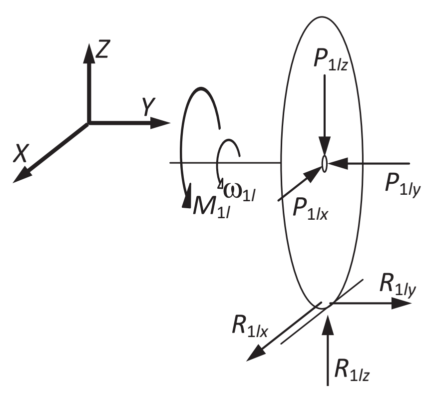

Fig. 2. Design scheme of wheel-road interaction

List of generalized coordinates and velocities:

1) V_{XM} – vehicle speed along the longitudinal axle associated with the moving coordinate system with the origin in the vehicle center of gravity (+forward);

2) V_{YM} – vehicle speed along the transverse axle associated with the moving coordinate system with the origin in the vehicle center of gravity (+left);

3) φ – steering angle (azimuth) of the vehicle relative to the vertical axle (+counterclockwise);

4) $\Delta\omega_{12-34}$ – difference in angular velocities of the front and rear axles at the output of the interaxle differential in the transfer case (+front axles are faster);

5) $\Delta\omega_{1-2}$ – difference in angular velocities of the first and second axles at the output of the interaxle differential in the second through axle (+first axle is faster);

6) $\Delta\omega_{3-4}$ – difference in angular velocities of the third and fourth axles at the output of the interaxle differential in the third through axle (+third axle is faster);

7) $\Delta\omega_{1l-1r}$ – difference in angular velocities of the left and right semiaxles of the first axle at the IWD output in the first axle (+left semiaxle is faster);

8) $\Delta\omega_{2l-2r}$ – difference in angular velocity of the left and right semiaxles of the second axle at the IWD output in the second axle (+left semiaxle is faster);

9) $\Delta\omega_{3l-3r}$ – difference in angular velocities of the left and right semiaxles of the third axle at the IWD output in the third axle (+left semiaxle is faster);

10) $\Delta\omega_{4l-4r}$ – difference in angular velocities of the left and right semiaxles of the fourth axle at the IWD output in the fourth axle (+left semiaxle is faster);

11) ω_{in} – speed of the input link of the interaxle differential in the transfer case;

12)–19) Δx_{ij} – longitudinal deformations of all tires (+wheel hub lags behind);

20)–27) Δy_{ij} – lateral deformations of all tires (+wheel hub to the right);

28)–35) Δz_{ij} – deformation of all suspensions (+compression);

36) α_M – trim angle of the vehicle body relative to the horizontal surface (+nose down);

37) β_M – roll angle of the vehicle body relative to the horizontal surface (+right side down);

38) X – vehicle motion along the x-axis of the fixed coordinate system;

39) Y – vehicle motion along the y-axis of the fixed coordinate system.

The differential equations describing the accelerated motion of these masses:

$$\frac{dV_{XM}}{dt}=\frac{\sum_{1}^{8}R_{ijX}-R_W}{m}, \tag{1}$$

where m is the vehicle weight; R_w is the air resistance $R_w=k_wF(V_{XM})^2$; R_{ijX} is the projection of the resulting force of wheel-road interaction on the longitudinal axle of the vehicle $R_{ijX}=(P_{Dij}-P_{fij})\cos\gamma_{ij}-P_{ijY}\sin\gamma_{ij}$,where in turn P_{Dij} is the wheel traction power, calculated as a function of the longitudinal tire deformation on each of the wheels and the maximum adhesion coefficient $P_{Dij}=f(\Delta x_{ij}, \varphi)$; P_{fij} is the force of wheel rolling resistance, calculated as a product of the motion resistance coefficient f and normal reaction of the road under the wheel R_{ijZ}; P_{ijY} is the wheel lateral force; γ_{ij} is the steering angle (for nonsteerable wheels $\gamma_{ij}=0$).

$$\frac{dV_{YM}}{dt}=\frac{\sum_{1}^{8}R_{ijY}}{m}, \tag{2}$$

where R_{ijY} is the projection of the resulting force of wheel-road interaction on the transverse axle of the vehicle $R_{ijY}=(P_{Dij}-P_{fij})\sin\gamma_{ij}+P_{ijY}\cos\gamma_{ij}$.

$$\frac{d^2\varphi}{d\varphi^2}=\frac{\sum_{1}^{8}\left(R_{ijX}(\pm b/2)+R_{ijY}l_i\right)}{I_Z}, \tag{3}$$

where $\pm b/2$ is half the wheel gauge with «+» for the right side wheels and with «–» for the left side wheels; l_i is the distance from the vehicle center of gravity to each of the axles taking into account the sign; I_Z is the vehicle moment of inertia when turning relative to the vertical axle.

$$\frac{d\Delta\omega_{12-34}}{dt}=\frac{(M_{12}-M_{34})\eta_{bg}^2}{I_{12}+I_{34}}, \tag{4}$$

where M_{12}, M_{34} are the torques on the output cardan shafts of the front and rear axles, respectively, I_{12}, I_{34} are the moments of inertia of the front and rear axles, respectively, reduced to the output differential gears, η_{bg} is bevel gear efficiency $\eta_{bg}\approx 0.95...0.96$.

$$\frac{d\Delta\omega_{1-2}}{dt}=\frac{(M_1-M_2)\eta_{bg}^2}{I_1+I_2}, \tag{5}$$

where M_1, M_2 are the torques on the output cardan shafts of the first and second axles, respectively, I_1, I_2 are the moments of inertia of the first and second axles, respectively, reduced to the output gears of the interaxle differential on the second through axle.

$$\frac{d\Delta\omega_{3-4}}{dt}=\frac{(M_3-M_4)\eta_{bg}^2}{I_3+I_4}, \tag{6}$$

where M_3, M_4 are torques on the output cardan shafts of the third and fourth axles, respectively, I_3, I_4 are the moments of inertia of the third and fourth axles, respectively, reduced to the output gears of the interaxle differential on the third through axle.

$$\frac{d\Delta\omega_i}{dt}=\frac{(M_{il}-M_{ir})\eta_{bg}^2-M_{ifr}}{I_{il}+I_{ir}}, \tag{7}$$

where M_{il}, M_{ir} are the torques on the semiaxles of the left and right wheels of the i-th axle; I_{il}, I_{ir} are the moments of inertia of the left and right wheels of the i-th axle, respectively, along with the wheel reduction gears; M_{ifr} is the additional locking torque, the value of which depends on IWD type and design. Equation (7) with a change of i from 1 to 4 describes operation with 7 to 10 generalized velocities.

For the IWD, whose locking degree depends on the load:

$$M_{ifr}=M_{fr0}+k_M(M_{il}+M_{ir}),$$

where M_{fr0} is the initial, constant value of the additional locking torque, k_M is the proportionality factor.

For the IWD, whose locking degree depends on the difference or squared difference in angular velocities of the semiaxles $M_{ifr}=k_\omega\Delta\omega$ or $M_{ifr}=k_\omega 2(\Delta\omega)^2$, depending on IWD type, where k_ω and $k_\omega 2$ are the corresponding proportionality factors.

$$\frac{d\omega_{in}}{dt}=\frac{M_{in}-M_{12}-M_{34}}{I_{in}+I_{12}+I_{34}}, \tag{8}$$

where M_{in} is the input (drive) torque on the interaxle differential housing in the transfer case, calculated through the specified constant drive power $M_{in}=N_{in}/\omega_{in}$.

$$\frac{d\Delta x_{ij}}{dt}=V_{\omega ij}-V_{Xij}\cos\gamma_{ij}-V_{Yij}\sin\gamma_{ij}, \tag{9}$$

where $V_{\omega ij}$ is the linear velocity of the tire at the center of contact with the road, calculated by the formula $V_{\omega ij}=\omega_{ij}r_{wij}$, where in turn ω_{ij} is the angular velocity of the wheel, r_{wij} is the actual rolling radius of the corresponding wheel; V_{Xij}, V_{Yij} are the linear velocities of the corresponding wheel hub with the vehicle in projections on the longitudinal and lateral axes, taking into account vehicle rotation relative to the vertical axle. Equation (9) with a change of i from 1 to 4 and j from 1 to 2 describes operation with 12 to 19 generalized displacements (deformations).

Longitudinal deformations of all tires Δx_{ij} allow, with a known tire elasticity in the longitudinal direction, determining the traction power P_{Dij}, limiting its value to the corresponding maximum force of wheel adhesion to the road.

$$\frac{d\Delta y_{ij}}{dt}=V_{LDij}+V_{Xij}\sin\gamma_{ij}-V_{Yij}\cos\gamma_{ij}, \tag{10}$$

where V_{LDij} is the velocity of the side wheel slip without slipping, due to rolling in the lateral tire deformation Δy_{ij}. Equation (10) with a change of i from 1 to 4 and j from 1 to 2 describes operation with 20 to 27 generalized displacements (deformations).

$$\frac{d^2\Delta z_{ij}}{dt^2}=\frac{R_{ijz}-P_{ijz}}{m_w}, \tag{11}$$

where R_{ijz} is the vertical force acting on the wheel hub from the tire side and calculated by the formula:

$$R_{ijz}=(r_{w0}-r_{wij})c_{rw}-\Delta x_{ij}k_{cxw}-\Delta y_{ij}k_{cyw},$$

where in turn r_{w0}, r_{wij} are free and actual wheel radii; c_{rw} is the radial tire stiffness; k_{cxw} and k_{cyw} are the coefficients of influence of the longitudinal and lateral tire deformation on the rolling radius; P_{ijz} is the reduced vertical force acting on the wheel hub from the suspension side and calculated by the formula:

$$P_{ijz}=\Delta z_{ij}c_s+(d\Delta z_{ij}/dt)k_s,$$

where in turn c_s is the reduced suspension stiffness; k_s is the reduced suspension damping factor; m_w is the wheel mass with the wheel reduction gear and part of independent suspension. Equation (11) with a change of i from 1 to 4 and j from 1 to 2 describes operation with 28 to 35 generalized displacements (deformations).

$$\frac{d^2\alpha}{dt^2}=\frac{\sum_1^8\left(P_{ijz}l_i\right)}{I_Y}, \tag{12}$$

where I_Y is the vehicle moment of inertia relative to the transverse axle passing through the center of gravity.

$$\frac{d^2\beta}{dt^2}=\frac{\sum_1^8\left(\pm P_{ijz}b/2\right)}{I_X}, \tag{13}$$

where I_X is the vehicle moment of inertia relative to the longitudinal axle passing through the center of gravity; «+» for left suspensions, «–» for right suspensions.

$$\frac{dX}{dt}=V_{XM}\cos\varphi-V_{YM}\sin\varphi, \tag{14}$$

where φ is the azimuth of vehicle motion in the fixed coordinate system, calculated by integrating the equation (3).

$$\frac{dY}{dt}=V_{YM}\cos\varphi+V_{XM}\sin\varphi. \tag{15}$$

The described mathematical model is implemented in the Borland Delphi environment.

6. Results of the comparative analysis of the influence of the design and parameters of the inter-wheel differential on handling

The numerical experiment was carried out using the described model, which allowed obtaining the following information array:

– power consumption in the curved motion with a given speed and theoretical turning radius;

– actual turning radius of the vehicle, taking into account the wheel slip, associated with the turning resistance of differentials of various designs.

Fig. 3 shows the graphical dependencies of the relative increase in power consumption of the armored personnel carrier. They are constructed with the average steering angle of 5° (Fig. 3, *a*), 15° (Fig. 3, *b*) and 25° (Fig. 3, *c*) depending on speed for different IWD. Relative power losses were calculated in relation to power losses in the linear motion of the armored personnel carrier at a similar speed.

Fig. 4 shows the graphical dependences of the relative increase in the actual turning radius of the armored personnel carrier. They are constructed with the average steering angle of 5° (Fig. 4, *a*), 15° (Fig. 4, *b*) and 25° (Fig. 4, *c*) depending on speed for the selected IWD. The relative increase in the actual turning radius was calculated in relation to the theoretical turning radius of the armored personnel carrier without taking into account the wheel slip in contact with the road.

In Fig. 3, 4, the corresponding colors and abbreviations indicate:

– open – standard bevel gear IWD without any positive locking means;

– w10, w20, w40 – IWD, in which locking degree depends on the wheel speed difference with the proportionality factors k_ω of 10, 20 and 40;

– qw05, qw10, qw15 and qw20 – IWD, in which locking degree depends on the squared wheel speed difference with the proportionality factors $k_\omega 2$ of 5, 10, 15 and 20;

– M02, M04, M06 – IWD, in which locking degree depends on the load with the proportionality factors k_M of 0.2, 0.4 and 0.6.

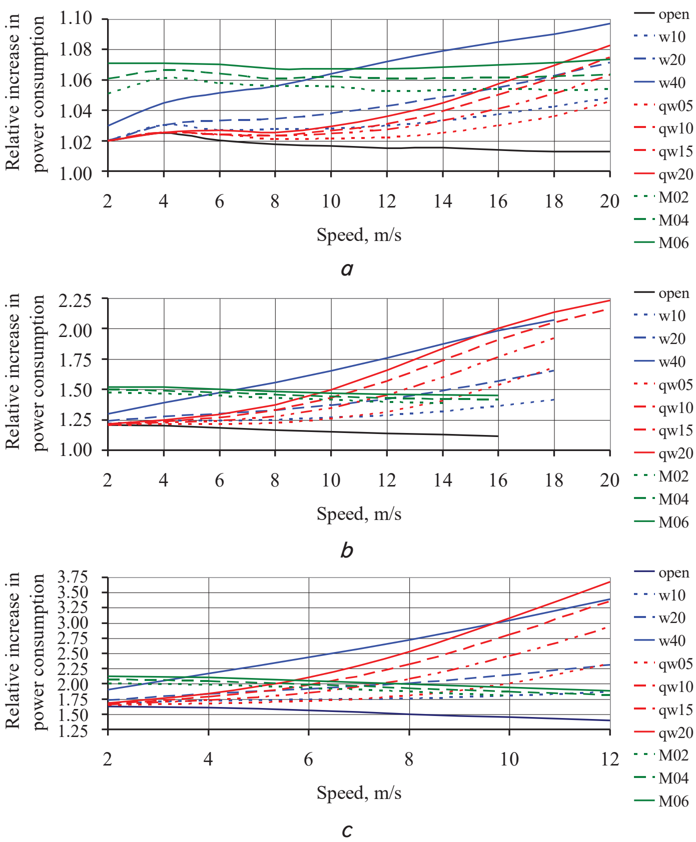

Fig. 3. Dependence of the relative increase in power consumption on speed for the average steering angle: *a* – 5°; *b* – 15°; *c* – 25°

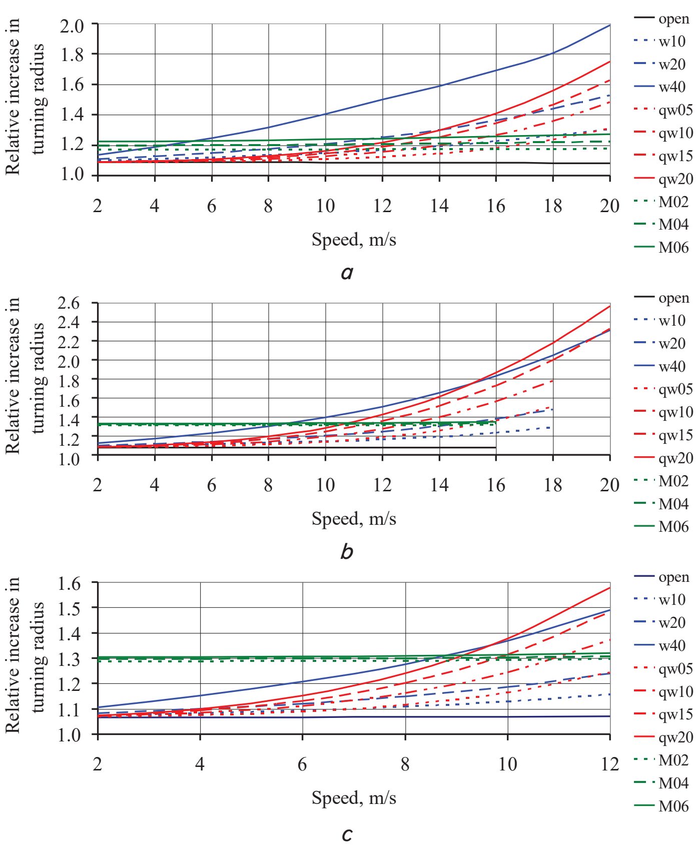

Fig. 4. Dependence of the relative increase in the actual turning radius on speed for the average steering angle: *a* – 5°; *b* – 15°; *c* – 25°

Due to the fact that the indicators of the locked IWD are substantially higher than those of other IWD types, this does not allow them to be displayed on a single graph, so these indicators are not shown in Fig. 3, 4.

7. Discussion of the results of the comparative analysis of the effect of the inter-wheel differentials design on the resistance of the car curved motion

Functional requirements to IWD of military and multi-purpose four-wheel drive vehicles are contradictory. On the one hand, IWD must ensure safe and dynamic vehicle motion off-road with a significant difference between the coefficients of wheel adhesion to the ground along the sides. On the other hand, IWD should not interfere with the curved motion increasing costs and actual turning radius. This problem can be solved in two ways. The first is that when moving under different road conditions, we change the IWD structure or significantly change control parameters within the constant structure. For example, in heavy road conditions, we engage the IWD lock manually or by means of automation. The second method is to attempt to appropriately select the proportionality factor for the locking torque by means of parametric optimization within the selected IWD structure.

When solving the problem in the second way, it should be remembered that to ensure the cross-country ability, the completely locked IWD has the best performance. However, in the curved motion on paved roads, such an IWD without unlocking has poor power and handling parameters. Therefore, for the analysis of the influence of the considered IWD on the curved motion resistance, the values of proportionality factors were selected, which provide the working area of the main functional parameter – cross-country ability. At the same time, the minimum values of the factors provide the minimum allowable parameters of dynamics and cross-country ability, and the maximum – high level of dynamics and cross-country ability, respectively, for military equipment.

Unlocked standard (open) bevel gear differentials and completely locked differentials were considered only to determine the boundaries of the best and worst handling indicators.

The graphs in Fig. 3 illustrate power consumption in the curved motion of the 8×8 armored personnel carrier relative to consumption in the linear motion under the same conditions. Depending on the steering angle, even for the standard (open) IWD, this consumption is increased from 1.5 % to 43 %. The corresponding increase in power losses for the completely locked IWD with a maximum steering angle is 4.61 times on paved roads.

From the calculations of the relative increase in the actual turning radius (Fig. 4), it can be seen that even for the standard (open) IWD when moving on dry asphalt concrete, the theoretical turning radius is increased from 4.5 % to 8.2 %. Depending on the steering angle, this is due to the wheel slip even without the use of additional locking torque.

The corresponding increase in the turning radius for the completely locked IWD with the steering angle of 5° is 25 times. Such a large increase in the turning radius is characteristic only for small steering angles, when the eight-wheeled vehicle due to the locked IWD to the steering angle of 5–8° continues the almost linear motion.

With the selected proportionality factors for small steering angles (up to 8°), in the first three quarters of the speed range, IWD have the advantage, in which locking degree depends on the squared velocity difference of the semiaxles. This advantage can be traced also energetically by the degree of turning resistance and actual turning radius increase.

For large steering angles in the first half of the safe speed range, IWD also have an advantage, in which locking degree depends on the squared velocity difference of the semiaxles. But at near breakdown speeds, for any steering angles, IWD are ahead, in which locking degree depends on the load.

According to the results of the analysis, the following recommendations can be formulated for further optimization of the search for effective IWD designs:

– for military vehicles whose task is to overcome artificial and natural obstacles, due to the possibility of bottom-out of one or more drive wheels, it is necessary to use either complete-locking IWD, or IWD, in which locking degree depends on the squared velocity difference of the semiaxles;

– for multi-purpose four-wheel drive vehicles, IWD can be considered, in which locking degree depends on the load, provided that it ensures good manufacturability, service life and reliability;

– for civil trucks and cars with increased cross-country ability, it is expedient to use standard (open) bevel gear IWD with automatic slip regulation, based on partial braking of the slipping wheel;

– when performing parametric optimization for any type of IWD, it is necessary to focus not only on a specific model of a car or military wheeled vehicle, but also on features of the foreseen operation conditions.

The research did not affect such an aspect as the effect of IWD design and parameters on car stability during acceleration and curved motion on a slippery road. This direction is planned as a task for further research.

7. Conclusions

1. For the analysis of the curved motion resistance of the four-wheel drive car, the following criteria are sufficiently informative:

– relative increase in power consumption in the motion with a predetermined constant speed and theoretical turning radius, in comparison with power consumption in the linear motion;

– relative increase in the actual turning radius of the vehicle, taking into account the tire slip in the contact spot, in comparison with the theoretical turning radius, calculated without taking into account the slip.

2. The developed mathematical model was verified by comparing the results of the calculations and earlier tests performed by the developers of the BTR-4 armored personnel carrier. By the parameter of the actual turning radius on the asphalt concrete pavement, the discrepancy between the calculation and experiment in the entire range did not exceed 7 %. This allows predicting the effective use of the developed mathematical model with the parametric optimization necessary for the synthesis of IWD with the required characteristics.

3. By the degree of influence on the curved motion resistance, the most rational among the three considered SLLSD are the dependences characterizing the operation of IWD, in which locking degree depends on the squared difference in angular velocities of the semiaxles and on the load. However, IWD, in which locking degree depends on the load, are not able to provide vehicle motion in case of the bottom-out of one or more drive wheels.

4. On the basis of IWD, in which locking degree depends on the squared difference in angular velocities of the semiaxles, it is possible to create the required internally automated IWD based on the technological gear pump without the use of friction discs.

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