Vpliv računskega polmera obračanja goseničnega vozila na potrebno moč motorja pri obračanju

The Impact of a Fixed Kinematic Turning Radius of a Tracked Vehicle on the Engine Power required in a Turn

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V prispevku je analiziran vpliv velikosti računskega polmera obračanja na potrebno moč motorja pri obračanju in na obremenjenost drsnih elementov mehanizma obračanja goseničnega vozila, ki ima mehanizem obračanja z več izračunanimi polmeri obračanja.

Definirani so parametri, ki opisujejo relativno spremembo potrebne moči motorja pri obračanju, relativno spremembo zavorne moči na drsnih elementih mehanizma obračanja in spremembo koeficienta koristnega delovanja mehanizma obračanja v odvisnosti od relativnega polmera obračanja.

Podana so priporočila za izbiranje računskih polmerov obračanja pri projektiranju mehanizma obračanja goseničnega vozila.

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(Ključne besede: goseničarji, obračanje vozil, moči motorjev, mehanizmi obračanja)

This paper analyzes the impact of a fixed kinematic turning radius on the engine power required in a turn and on the load of friction elements in the turning mechanism of a tracked vehicle equipped with turning mechanisms that have several fixed kinematic turning radii.

Parameters describing the relative change of engine power in a turn, the relative change of braking power on the friction elements of the turning mechanism, and the change of efficiency coefficient of the turning mechanism depending on the relative turning radius are defined.

Recommendations regarding the choice of fixed kinematic turning radii, relating to the design of the turning mechanism of a tracked vehicle, are given.

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(Keywords: tracked vehicles, turning radius, engine power, turning mechanism)

0 UVOD

Obračanje goseničnega vozila se doseže s spremembo hitrosti navijanja ene v primerjavi z drugo gosenico ob hkratnem oblikovanju različnih vlečnih sil na gosenicah ([1] in [2]). Dotikalni del gosenice dobiva ob vzdolžnem pomikanju pri obračanju še dodatno bočno premikanje po tleh. Zaradi vzdolžnih pomikov gosenice se pojavljajo odpori tudi pri gibanju vozila naravnost. Bočni pomiki gosenic povzročajo dodatne odpore zaradi bočnega trenja med gosenicami in tlemi, kakor tudi zaradi gnetenja, rezanja in premikov tal pod gosenicami. Zaradi tega se pojavi pri vstopu v obračanje goseničnega vozila porast skupnih odporov gibanja, zaradi česar se poveča obremenitev pogonskega motorja in pogonskega mehanizma, posledično pa tudi poveča obremenitev posameznih elementov v mehanizmih obračanja.

Obremenitev motorja in pogonskega mehanizma goseničnega vozila ter posameznih elementov v mehanizmih za obračanje je v veliki meri

0 INTRODUCTION

Turning a tracked vehicle is accomplished by changing the winding speed of one of its tracks with respect to the other, resulting in the simultaneous occurrence of different tractive forces on each track ([1] and [2]). The leaning part of the track, apart from the longitudinal, receives an additional lateral shift during the turn. Longitudinal shifts of the tracks induce the same kind of resistances as during rectilinear motion of the vehicle. Lateral shifts induce additional resistances due to lateral friction between the tracks and the ground and due to the compression, cutting and mixing of soil under the tracks. When a turn begins, the total resistance to motion of a tracked vehicle is increased, this causes an increased load on the engine and transmission, and therefore an increased load on the elements of the turning mechanisms.

The loads on the engine and transmission of a tracked vehicle, as well as on the elements of its turning mechanisms, significantly depend on the

odvisna od velikosti t.i. računskega polmera obračanja R_p. Računski polmer obračanja je tisti polmer obračanja, ki se realizira ob fiksnem prenosnem razmerju pogonske verige do notranje gosenice in brez drsenja drsnih elementov v mehanizmih obračanja ([1] in [2]).

V tem prispevku je analiziran vpliv velikosti računskega polmera obračanja na potrebno moč motorja v obračanju in na obremenjenost drsnih elementov mehanizma obračanja na konkretnem primeru gosenečga vozila, ki ima mehanizme obračanja z več računskimi polmeri obračanja. Definirani so parametri, ki opisujejo relativno spremembo potrebne moči motorja pri obračanju, relativno spremembo zavorne sile na drsnih elementih mehanizma za obračanje in spremembo koeficienta koristnega delovanja mehanizma obračanja v odvisnosti od relativnega polmera obračanja. Na podlagi izvedene analize so dana priporočila za izbiranje računskega polmera obračanja.

1 DEFINIRANJE PARAMETROV ZA ANALIZIRANJE OBRAČANJA

Analizirano je obračanje gosenečga vozila na vodoravni podlagi (sl. 1 in 2), ob naslednjih predpostavkah in omejitvah:
-   zanemarjeno je podrsavanje zunanje in notranje gosenice,
-   predpostavljena širina sledi gosenice je ena ($B = 1$),
-   zanemarjen je vpliv sredobežne sile,
-   predpostavljena teža vozila je ena ($G = 1$),
-   analizirano je obračanje s polmerom obračanja R v območju od obračanja okoli notranje gosenice do obračanja s prostim polmerom obračanja.

Prosti polmer obračanja je tisti, pri katerem je sila na notranji gosenici enaka nič ($F_1 = 0$). Pri obračanjih, ki so večja od prostega polmera obračanja, na notranji gosenici nastane vlečna sila, pri polmerih obračanja, ki so manjši od prostega polmera obračanja, pa se na notranji gosenici formira zavorna sila ([1] in [2]).

Na sliki 1 so prikazane sile in vrtilni momenti, ki delujejo na gosenično vozilo:
R_2 = odpor pri gibanju naravnost na zunanji gosenici,
R_1 = odpor pri gibanju naravnost na notranji gosenici,
F_2 = vlečna sila na zunanji gosenici,
F_1 = zavorna sila na notranji gosenici,
M_c = moment bočnih odporov obračanja.

Odpora pri gibanju naravnost R_1 in R_2 ter moment bočnega odpora obračanja M_c, se določajo z izrazi ([1] in [2]):

magnitude of the so-called fixed kinematic turning radius R_p. The fixed kinematic turning radius is the turning radius realized with a fixed transmission ratio of a kinematic chain to an inner track and without any sliding of the friction elements in the turning mechanism ([1] and [2]).

This paper analyzes the impact of the fixed kinematic turning radius on the engine power required in a turn and on the load of the friction elements of the turning mechanism by using the concrete example of a tracked vehicle equipped with turning mechanisms that have several fixed kinematic turning radii. The parameters that describe the relative change of the required engine power in a turn, the relative change of the braking power on the friction elements of the turning mechanism, and the change of the efficiency coefficient of the turning mechanism depending on the relative turning radius are defined. On the basis of the performed analysis, recommendations regarding the choice of fixed kinematic turning radii are given.

1 DEFINITION OF PARAMETERS FOR ANALYZING A TURN

The turn of a tracked vehicle on a horizontal surface (Figs. 1 and 2) is analyzed with the following assumptions and constraints:
-   sliding of the outer track and the inner track is neglected,
-   the distance between the tracks is assumed to be one ($B = 1$),
-   the influence of centrifugal force is neglected,
-   the vehicle's weight is assumed to be one ($G = 1$),
-   turns with turning radii R in the interval ranging from a turn around the inner track to a turn with a free turning radius are analyzed.

The free turning radius is characterized by the force on the inner track being zero ($F_1 = 0$). For turns with radii larger than the free turning radius a tractive force is induced on the inner track, while for turns with radii less than the free turning radius a braking force is induced on the inner track ([1] and [2]).

The forces and torques acting on a tracked vehicle are presented in Figure 1:
R_2 = resistance to rectilinear motion on the outer track,
R_1 = resistance to rectilinear motion on the inner track,
F_2 = tractive force on the outer track,
F_1 = braking force on the inner track,
M_c = turning resistance torque.

Resistances to rectilinear motion R_1 and R_2, and the turning resistance torque M_c are determined by the following expressions ([1] and [2]):

$$R_1 = 0{,}5\,f\,G \tag{1},$$

$$R_2 = 0{,}5\,f\,G \tag{2},$$

$$M_c = \frac{\mu G L}{4} \tag{3},$$
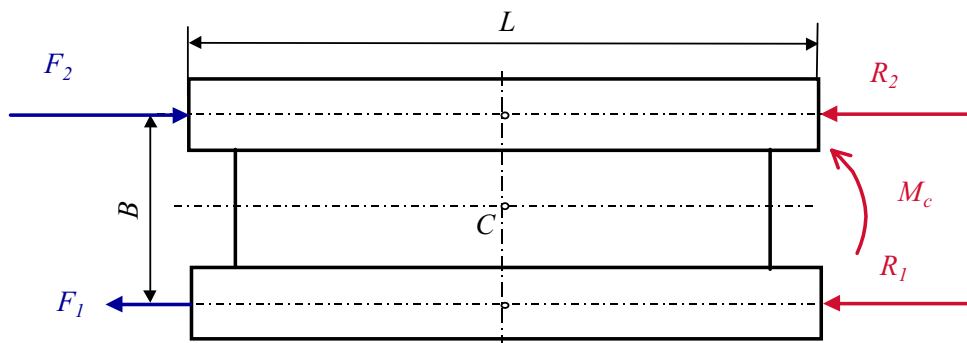
Sl.1. *Sile in momenti, ki delujejo pri obračanju gos:eničnega vozila*
Fig. 1. *Forces and torques on a tracked vehicle in a turn*

$$\mu = \frac{\mu_{maks}}{0,925 + 0,15\rho} \tag{4},$$

kjer so:

f = koeficient odpora pri gibanju naravnost na določeni podlagi,

ρ = relativni polmer obračanja ($\rho = R/B$),

μ = koeficient odpora obračanja na določeni podlagi in za določeni relativni polmer obračanja,

μ_{maks} = koeficient odpora obračanja na določeni podlagi pri $\rho = 0,5$,

L = dolžina dotikalnega dela gosenice.

Vlečna sila na zunanji gosenici F_2 in zavorna sila na notranji gosenici F_1 lahko določimo z izrazoma ([1] in [2]):

where:

f = the rectilinear motion resistance coefficient for a particular surface,

ρ = the relative turning radius ($\rho = R/B$),

μ = the turning resistance coefficient for a particular surface and a particular relative turning radius,

μ_{max} = the turning resistance coefficient for a particular surface at $\rho = 0.5$,

L = the length of the leaning part of a track.

The tractive force on the outer track F_2 and the braking force on the inner track F_1 can be determined by the following expressions ([1] and [2]):

$$F_2 = R_2 + \frac{M_c}{B} \tag{5},$$

$$F_1 = -R_1 + \frac{M_c}{B} \tag{6}.$$

Na sliki 2 so prikazane hitrosti vozila pri obračanju in relativni polmeri obračanja gos';eničnega vozila:

v_2 = hitrost gos eničnega vozila na zunanji gosenici,

v_1 = hitrost goseničnega vozila na notranji gosenici,

Figure 2 presents the velocities and relative turning radii of a turning tracked vehicle:

v_2 = the velocity of a tracked vehicle on the outer track,

v_1 = the velocity of a tracked vehicle on the inner track,
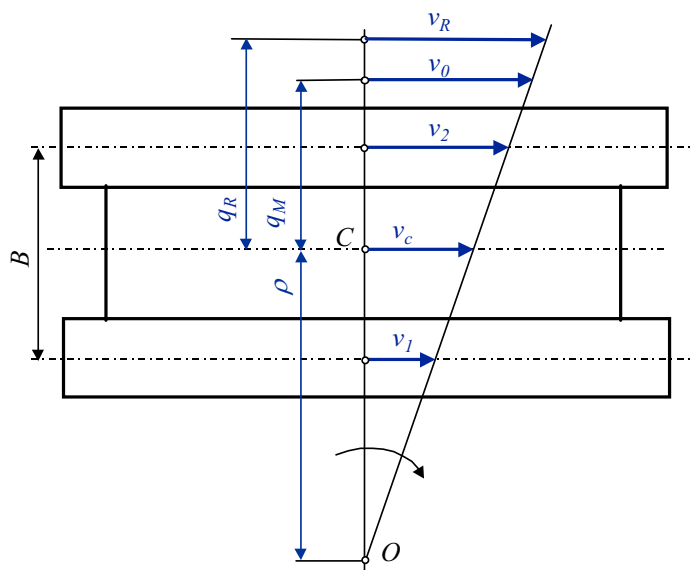
Sl.2. *Načrt hitrosti goseničnega vozila pri obračanju*
Fig. 2. *Velocity plan of a turning tracked vehicle*

v_c = hitrost težišča gosemičnega vozila,
v_0 = hitrost točke vozila, ki ohranja enako hitrost pri obračanju in pri gibanju naravnost,
q_M = kinematski parameter mehanizma obračanja (relativna koordinata točke vozila, ki ima hitrost v_0).

Uvedimo parameter q_R, ki zadovoljuje naslednji pogoj [3]:

v_c = the velocity of the center of mass of a tracked vehicle,
v_0 = the velocity of a point on the vehicle which retains the same velocity in a turn as in rectilinear motion,
q_M = the kinematic parameter of the turning mechanism (relative coordinate of a point on the vehicle with velocity v_0).

Let us introduce parameter q_R, satisfying the following condition [3]:

$$(R_1 + R_2) \cdot q_R \cdot B = M_c \qquad (7).$$

Z uporabo izraza (7) uvajamo povezanost med momentom odpora obračanju M_c in momentom vsote odporov pri gibanju naravnost $(R_1 + R_2) \cdot B$. Po uvrstitvah izrazov (1), (2), in (3) v izraz (7) dobimo:

Expression (7) describes the correlation between the turning resistance torque M_c and the torque of the sum of resistances to rectilinear motion $(R_1 + R_2) \cdot B$. Inserting (1), (2) and (3) into (7) results in the following expression:

$$q_R = \frac{\mu L}{4 f B} \qquad (8).$$

Uvedemo pojem relativnega računskega polmera obračanja ρ_p:

The relative fixed kinematic turning radius ρ_p is defined as:

$$\rho_p = \frac{R_p}{B} \qquad (9).$$

Pri obračanju gosemičnega vozila, razen odporov pri gibanju naravnost $(R_1 + R_2)$, obstajajo tudi dodatni bočni odpori, ki oblikujejo moment odpora obračanja M_c. Potrebna moč za obvladovanje skupnih zunanjih odporov obračanja (P_0) je večja od potrebne moči za premagovanje odporov pri gibanju naravnost (P_{pr}) ([3] in [4]):

Besides resistances to rectilinear motion $(R_1 + R_2)$, there are additional lateral resistances, which form the turning resistance torque M_c in the turning movement of a tracked vehicle. The power required for overcoming the total external resistances in a turn (P_0) is larger than the power required for overcoming the resistance in rectilinear motion (P_{pr}) ([3] and [4]):

$$P_{pr} = (R_1 + R_2) \, v_0 \qquad (10),$$

$$P_0 = (R_1 + R_2) v_0 \frac{q_R + \rho}{q_M + \rho} \qquad (11),$$

$$P_0 = P_{pr} \frac{q_R + \rho}{q_M + \rho} \qquad (12).$$

Potrebna moč motorja za izvedbo obračanja gosemičnega vozila (P_{mz}) se določi z izrazom ([3] in [4]):

The engine power required for accomplishing the turn of a tracked vehicle (P_{mz}) is determined by the following expressions ([3] and [4]):

$$P_{mz} = (R_1 + R_2) v_0 \frac{q_R + \rho_p}{q_M + \rho_p} \qquad (13),$$

$$P_{mz} = P_{pr} \frac{q_R + \rho_p}{q_M + \rho_p} \qquad (14).$$

Pri obračanju gosemičnega vozila s polmeri obračanja, ki so različni od računskih, se na drsnih elementih mehanizma za obračanje sprošča moč drsenja (P_k), ki se določa z izrazom ([3] in [4]):

The power of sliding (P_k) that is spent on the friction elements of the turning mechanism during the turns of a tracked vehicle at turning radii different from the fixed kinematic turning radius is determined by the following expression ([3] and [4]):

$$P_k = (R_1 + R_2) v_0 \frac{q_R - q_M}{q_M + \rho_p} \frac{\rho - \rho_p}{q_M + \rho} \qquad (15),$$

$$P_k = P_{pr} \frac{q_R - q_M}{q_M + \rho_p} \frac{\rho - \rho_p}{q_M + \rho} \qquad (16).$$

Uvedemo koeficient relativne potrebne moči motorja pri obračanju (ψ_{mz}), koeficient relativne moči

The coefficient of relative engine power required in a turn (ψ_{mz}), the coefficient of relative

drsenja v drsnih elementih mehanizma za obračanje ψ_k in koeficient koristnega delovanja mehanizma obračanja η_{mz}, in sicer ([3] in [4]):

power of sliding in the friction elements of the turning mechanism ψ_k, and the efficiency coefficient of the turning mechanism η_{mz} are defined as ([3] and [4]):

$$\psi_{mz} = \frac{P_{mz}}{P_{pr}} \tag{17},$$

$$\psi_{mz} = \frac{q_R + \rho_p}{q_M + \rho_p} \tag{18},$$

$$\psi_k = \frac{P_k}{P_{pr}} \tag{19},$$

$$\psi_k = \frac{q_R - q_M}{q_M + \rho_p} \frac{\rho - \rho_p}{q_M + \rho} \tag{20},$$

$$\eta_{mz} = \frac{P_0}{P_{mz}} \tag{21},$$

$$\eta_{mz} = \frac{q_R + \rho}{q_M + \rho} \frac{q_M + \rho_p}{q_R + \rho_p} \tag{22}.$$

Na podlagi gornjih izrazov lahko povzamemo, da se koeficient relativne potrebne moči motorja pri obračanju ψ_{mz}, koeficient relativne moči podrsavanja v drsnih elementih mehanizma obračanja ψ_k in koeficient koristnega delovanja mehanizma obračanja η_{mz} spreminjajo v odvisnosti od:
− koeficienta odpora gibanja f na tleh, kjer se obrača,
− relativnega polmera obračanja ρ,
− relativnega proračunskega polmera obračanja ρ_p in
− parametra gibanja q_M.

Idealen mehanizem obračanja bi bil tisti, pri katerem bi za vsak relativni polmer obračanja dosegli $\psi_{mz} = 1$, $\psi_k = 0$ in $\eta_{mz} = 1$. Vrednosti koeficientov ψ_{mz}, ψ_k in η_{mz} za realni mehanizem obračanja pričajo o stopnji uspešnosti projektnega oblikovanja mehanizma obračanja z vidika smotrne rabe moči motorja, a posredno tudi o njegovem vplivu na porabo pogonskega goriva.

These expressions show that the coefficient of relative engine power required in a turn ψ_{mz}, the coefficient of relative power of sliding in the friction elements of the turning mechanism ψ_k, and the efficiency coefficient of the turning mechanism η_{mz} depend on:
− the rectilinear motion resistance coefficient f for a particular surface,
− the relative turning radius ρ,
− the relative fixed kinematic turning radius ρ_p,
− the kinematic parameter q_M.

An ideal turning mechanism would achieve $\psi_{mz} = 1$, $\psi_k = 0$ and $\eta_{mz} = 1$ for every turning radius. For real turning mechanisms the values of the coefficients ψ_{mz}, ψ_k and η_{mz} indicate the quality of design, regarding rational use of engine power, and indirectly influence fuel consumption.

2 ANALIZA VPLIVA VELIKOSTI RAČUNSKEGA POLMERA OBRAČANJA

Za izvajanje analize je izbran mehanizem obračanja na goseničnem vozilu BVP - M80, katerega shema gibanja [5] je prikazana na sliki 3. Prikazan mehanizem obračanja sodi v skupino t.i. pogonskih prenosov v bloku (TUB), pri katerih je pogonsko in konstrukcijsko združena funkcija menjanja stopnje prenosa v menjalniku in menjanje hitrosti obračanja pogonskih koles pri obračanju goseničnega vozila. Ta pogonski prenos v bloku je sestavljen iz menjalnika (MJ), zbirnih diferencialov (SD_2 in SD_1), pomožnih drsnih spojk (ϕ_{n2} in ϕ_{n1}), pomožnih drsnih zavor (T_{n2} in T_{n1}) in drsnih zavor za zaustavljanje (T_{02} in T_{01}).

S sprožitvijo drsne zavore T_0 v TUB do notranje gosenice nastane najmanjši relativni računski polmer obračanja $\rho_{p0} = 0,5$. S popolnim proženjem pomožne drsne spojnice ϕ_n v TUB do notranje gosenice nastane še po en relativni računski polmer obračanja v vsaki stopnji prenosa ($\rho_{p1} = 1,2$; $\rho_{p2} = 3,1$; $\rho_{p3} = 4,8$; $\rho_{p4} = 7,5$; $\rho_{p5} = 11,1$). Obračanje s preostalimi polmeri obračanja

2 AN IMPACT ANALYSIS FOR A FIXED KINEMATIC TURNING RADIUS

The analysis is conducted on the turning mechanism of the BVP-M80 tracked vehicle. Its kinematic scheme [5] is presented in Figure 3. This turning mechanism belongs to the category of the so-called block transmissions (BT). In block transmissions the functions of changing gears in a gearbox and of changing the rotation speed of the propulsion wheels during a turn are kinematically and constructionally unified. The presented block transmission consists of a gearbox (MJ), summing differentials (SD_2 and SD_1), auxiliary friction clutches (ϕ_{n2} and ϕ_{n1}), auxiliary friction brakes (T_{n2} and T_{n1}), and halting friction brakes (T_{02} and T_{01}).

By activating a halting friction brake T_0 in the BT to the inner track, the minimum relative fixed kinematic turning radius of $\rho_{p0} = 0.5$ is realized. By fully activating an auxiliary friction clutch ϕ_n in the BT to the inner track, yet another relative fixed kinematic turning radius is realized for each gear ($\rho_{p1} = 1.2$; $\rho_{p2} = 3.1$; $\rho_{p3} = 4.8$; $\rho_{p4} = 7.5$; $\rho_{p5} = 11.1$). Turns
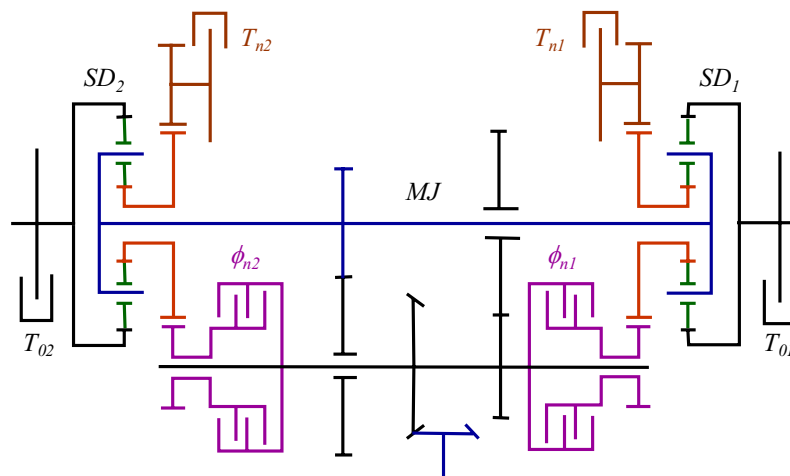
Sl. 3. *Shema gibanja mehanizma obračanja gosenčnega vozila BVP M-80*
Fig. 3. *Kinematic scheme of the turning mechanism of the BVP-M80 tracked vehicle*

nastane z drsenjem enega od drsnih elementov v TUB (ϕ_n ali T_n ali T_0).

Za gosenično vozilo BVP M-80 (5) so znane vrednosti [5] $q_M = 0,5$ in $L/B = 1,3$. Analiziramo obračanje z vozilom na vodoravni s travo pokriti podlagi z osvojenima koeficientoma ([1] in [2]) $f = 0,06$ in $\mu_{max} = 0,8$. Grafični prikaz rezultatov računanja koeficientov ψ_{mz}, ψ_k in η_{mz}, za izbrano vozilo je podan na slikah 4, 5 in 6.

Pri analizi dobljenih rezultatov je opazno naslednje:
– večanje relativnega računskega polmera obračanja vpliva na zmanjševanje koeficientov relativne potrebne moči motorja pri obračanju ψ_{mz} in koeficienta relativne zavorne moči ψ_k, ob povečanju koeficienta koristnega delovanja mehanizma obračanja η_{mz} za relativne polmere

with other turning radii are realized by sliding one of the friction elements in the BT (ϕ_n or T_n or T_0).

For the BVP M-80 tracked vehicle, the following values are known [5]: $q_M = 0.5$ and $L/B = 1.3$. The turn of the vehicle on a horizontal sodded-grass surface is analyzed with the assumed coefficients ([1] and [2]) $f = 0.06$ and $\mu_{max} = 0.8$. Graphs of the coefficients ψ_{mz}, ψ_k and η_{mz} are calculated for the chosen vehicle and presented in Figures 4, 5 and 6.

Analysis of the results shows that:
– with the increase of the relative fixed kinematic turning radius, the coefficient of relative engine power required in a turn ψ_{mz} decreases, as does the coefficient of the relative power of sliding in the friction elements of the turning mechanism ψ_k, while the efficiency coefficient of the turning mechanism η_{mz} increases for relative turning radii
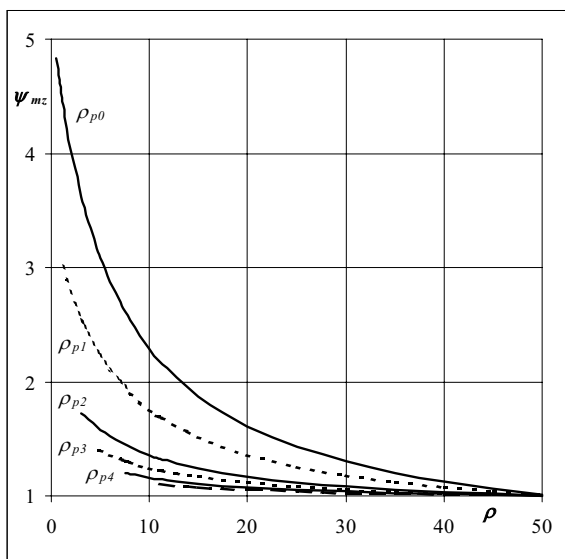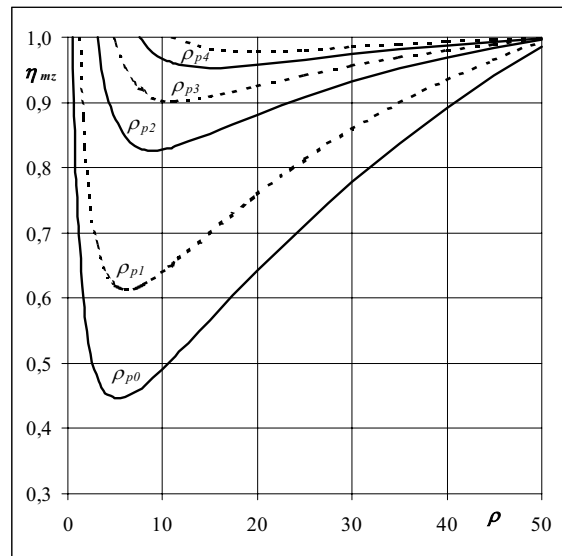

Sl. 4. *Koeficient relativne potrebne moči motorja pri obračanju*
Fig. 4. *Coefficient of relative engine power required in a turn*
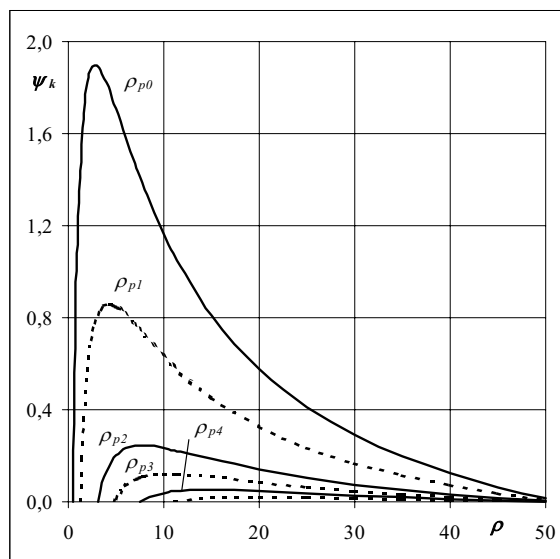
Sl. 5. *Koeficient relativne moči drsenja v mehanizmu obračanja*
Fig. 5. *Coefficient of relative power of sliding in the turning mechanism*

Sl. 6. *Koeficient koristnega dela mehanizma obračanja*
Fig. 6. *Efficiency coefficient of the turning mechanism*

obračanja ρ, ki so večji od relativnega računskega polmera obračanja za posamezno stopnjo prenosa,

– zmanjševanje koeficientov ψ_{mz} in ψ_k ter povečanje koeficienta η_{mz} je posebno izraženo v območju relativnega računskega polmera $\rho_{pi} \in \{1,0; 5,0\}$,

– za intervale računskih polmerov $\rho_{pi} \in \{5,1; 15,0\}$ je gibanje ugodnih vrednosti koeficientov ψ_{mz}, ψ_k in η_{mz} majhne intenzivnosti,

– obstoj več relativnih računskih polmerov obračanja, ki se medsebojno razlikujejo po velikosti, omogoča prehod obračanja goseničnih vozil z višje ravni (ρ_{pi}) na sosednjo nižjo raven (ρ_{pi-1}) v primeru, ko je treba izvesti obračanje z relativnim polmerom, ki je manjši od ρ_{pi} in večji od $\rho_{p0} = 0,5$,

– prehod obračanja goseničnega vozila z višje ravni (ρ_{pi}) na sosednjo nižjo raven (ρ_{pi-1}) v primeru, ko je treba izvesti obračanje z relativnim polmerom, manjšim od (ρ_{pi}), a večjim od $\rho_{p0} = 0,5$, zmanjšuje obremenitev pogonskega motorja in drsnih elementov mehanizma obračanja.

3 SKLEP

Pri izvajanju manevra obračanja goseničnega vozila se z zmanjševanjem polmera obračanja pojavljajo povečani odpori obračanja in povečane obremenitve pogonskega motorja in sklopov v prenosnem mehanizmu za gibanje vozila.

Obstoj relativnih računskih polmerov obračanja, ki so večji od $\rho_{p0} = 0,5$, prispeva k zmanjševanju obremenitev pogonskega motorja in elementov v prenosih v primeru obračanja s polmeri obračanja, ki so enaki ali večji od relativnega računskega polmera obračanja. Koeficienti relativne potrebne moči motorja pri obračanju (ψ_{mz}), relativne zavorne moči (ψ_k) ter koeficient koristnega delovanja mehanizma obračanja (η_{mz}) imajo ugodne vrednosti za to območje obračanja.

ρ that are larger than the relative fixed kinematic turning radius ρ_{pi} for the particular gear,

– the decrease of ψ_{mz} and ψ_k and the increase of η_{mz} are particularly pronounced in the interval of the relative fixed kinematic turning radii $\rho_{pi} \in \{1.0; 5.0\}$,

– in the interval of relative fixed kinematic turning radii $\rho_{pi} \in \{5.1; 15.0\}$ the favorable trend of values of the coefficients ψ_{mz}, ψ_k and η_{mz} is more moderate,

– the existence of several relative fixed kinematic turning radii that are different in magnitude allows transitioning from a higher (ρ_{pi}) to the nearest lower level (ρ_{pi-1}), for the case when a turn with relative radius less than ρ_{pi} and larger than $\rho_{p0} = 0.5$ is required,

– transitioning from a higher (ρ_{pi}) to the nearest lower level (ρ_{pi-1}), for the case when a turn with a relative radius less than ρ_{pi} and larger than $\rho_{p0} = 0.5$ is required, decreases the engine load and the load on the friction elements of the turning mechanism.

3 CONCLUSION

The decrease in turning radius during the turn of a tracked vehicle leads to increased turning resistances, an increased engine load and an increased load on the transmission devices of the vehicle.

The existence of relative fixed kinematic turning radii larger than $\rho_{p0} = 0.5$ enables a decrease in the engine load and the load on the transmission elements for the cases of turns with turning radii greater than or equal to the relative fixed kinematic turning radius. The coefficient of relative engine power required in a turn (ψ_{mz}), the coefficient of relative braking power (ψ_k), and the efficiency coefficient of the turning mechanism (η_{mz}) have favorable values for those turning radii.

Ugodno gibanje vrednosti koeficientov ψ_{mz}, ψ_k in η_{mz} je posebej izraženo pri relativnih računskih polmerih obračanja v območju $\rho_{pi} \in \{1,0;\ 5,0\}$. Za intervale relativnih računskih polmerov $\rho_{pi} \in \{5,1;\ 15,0\}$ je gibanje ugodnih vrednosti koeficientov ψ_{mz}, ψ_k in η_{mz} umirjene intenzivnosti. V področju obračanja, v katerem bi se uporabili relativni računski polmeri obračanja, ki so večji od $\rho_{pi} = 15$, se koeficienti ψ_{mz}, ψ_k in η_{mz} le malo spreminjajo.

Ob projektiranju novih pogonskih prenosov za gosenična vozila je zato priporočljivo uporabiti en računski polmer obračanja $\rho_{p0} = 0,5$ in več relativnih računskih polmerov obračanja v območju $\rho_{pi} \in \{\ 1,0;\ 15,0\}$.

A favorable trend of values of the coefficients ψ_{mz}, ψ_k and η_{mz} is particularly pronounced for relative fixed kinematic turning radii in the interval $\rho_{pi} \in \{1.0;\ 5.0\}$. In the interval of relative fixed kinematic turning radii $\rho_{pi} \in \{\ 5.1;\ 15.0\}$ the favorable trend of values of ψ_{mz}, ψ_k and η_{mz} is more moderate. In the range of turns with relative fixed kinematic turning radii larger than $\rho_{pi} = 15$, the coefficients ψ_{mz}, ψ_k and η_{mz} do not change significantly.

When designing the transmission of a tracked vehicle it is desirable to realize one fixed kinematic turning radius $\rho_{p0} = 0.5$ and several other fixed kinematic turning radii in the interval $\rho_{pi} \in \{\ 1.0;\ 15.0\}$.

4 LITERATURA
4 REFERENCES

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