1st International Conference on
Advanced Technologies for Developing Countries
September 12-14, 2002
Slavonski Brod, Croatia

IMPACT OF THE TRACKED VEHICLE TURNING MECHANISM ON THE ENGINE POWER REQUIRED IN TURN

V. Stojković, E. Bazijanac and M. Jakopčić

Keywords: tracked vehicle, turning mechanism, engine power required in turn, relative fixed kinematic turning radius

1. Introduction

The turn of a tracked vehicle is accomplished by changing the winding speed of one of its tracks with relation to the other, with simultaneous formation of different tractive forces on each track [1,4]. During the turn of the tracked vehicle there is total increase of the movement drag, causing increased engine and transmission loads. The engine and transmission load of the tracked vehicle depend to a great extent also on the size of the so-called kinematic turning radius (R_p). The kinematic turning radius is the turning radius which is realised with fixed transmission relation of the kinematic chain to the inner tracks and without the sliding of frictional elements in the turning mechanism [1,4].

This paper analyses the impact of the type of the military vehicle turning mechanism on the engine power required for the turn and the efficiency coefficient of the turning mechanism. Parameters are defined, describing relative change in the engine power required for the turn and the change of efficiency coefficient of the turning mechanism depending on the relative turning radius. Based on the performed analysis, recommendations are provided for the selection of the kinematic turning radius.

2. Definition of Parameters for Analysing the Turn

The turn of a tracked vehicle on a horizontal surface (Figures 1 and 2) is analysed with the turning radius (R) within the interval from the turn around the inner track to the turn with a free turn radius (F_1 = 0), with the following assumptions and restrictions: the impact of the centrifugal force is neglected; the track width and the vehicle weight equal one (B = 1, G = 1).

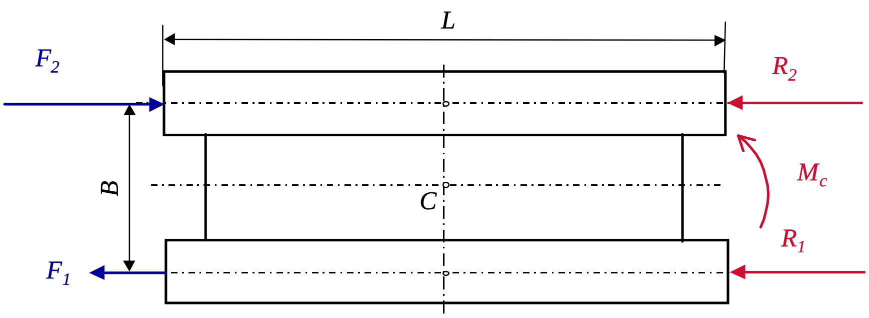

Figure 1 Forces and torques on a tracked vehicle in turn

Figure 1 shows forces and torques acting on a tracked vehicle:

R_2 = resistance to rectilinear motion on the outer tracks,

R_1 = resistance to rectilinear motion on the inner tracks,

F_2 = tractive force on the outer tracks,

F_1 = braking force on the inner tracks,

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 [1,4]:

$$R_1 = 0.5\, f\, G, \tag{1}$$

$$R_2 = 0.5\, f\, G, \tag{2}$$

$$M_c = \frac{mGL}{4}, \tag{3}$$

$$m = \frac{m_{\max}}{0.925 + 0.15\, r}, \tag{4}$$

where:

f = rectilinear motion resistance coefficient for a particular surface,

L = length of the leaning part of the tracks,

r = relative turning radius ($r = \frac{R}{B}$),

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

$m_{\max}$ = turning resistance coefficient for a particular surface at r = 0.5.

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 [1,4]:

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

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

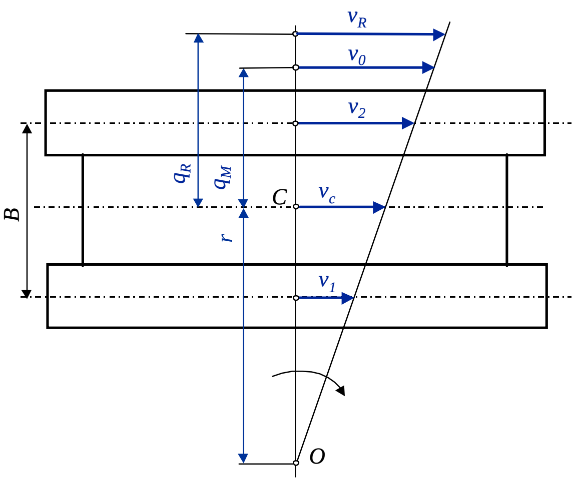

Figure 2 Plan of velocities of turning tracked vehicle

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

v_2 = velocity of tracked vehicle on the outer track,
v_1 = velocity of the tracked vehicle on the inner track,
v_c = velocity of the centre of mass of a tracked vehicle,
v_0 = velocity of the point on the vehicle which retains the same velocity in turn as in rectilinear motion,
q_M = kinematic parameter of the turning mechanism (relative co-ordinate of the point on the vehicle with velocity v_0).

Parameter q_R is introduced which satisfies the following condition [3]:

$$(R_1 + R_2)\ q_R\ B = M_c\ . \tag{7}$$

After inserting expressions (1), (2) and (3) into the expression (7), the result is:

$$q_R = \frac{mL}{4fB}. \tag{8}$$

The term of relative fixed kinematic turning radius r_p is introduced:

$$r_p = \frac{R_p}{B}. \tag{9}$$

The power required to overcome the total outer resistances in turn (P_0) is greater than the power necessary to overcome resistance in rectilinear motion (P_{pr}) [2,3]:

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

$$P_0 = P_{pr}\ \frac{q_R + r}{q_M + r}. \tag{11}$$

The engine power required to make the tracked vehicle turn (P_{mz}) is determined according to the expression [2,3]:

$$P_{mz} = (R_1 + R_2)\ v_0\ \frac{q_R + r_p}{q_M + r_p} = P_{pr}\ \frac{q_R + r_p}{q_M + r_p}. \tag{12}$$

The coefficient of the relative engine power required in turn (y_{mz}) and the efficiency coefficient of the turning mechanism (h_{mz}) are introduced in the following manner [2,3]:

$$y_{mz} = \frac{P_{mz}}{P_{pr}} = \frac{q_R + r_p}{q_M + r_p}, \tag{13}$$

$$h_{mz} = \frac{P_0}{P_{mz}} = \frac{q_R + r}{q_M + r}\ \frac{q_M + r_p}{q_R + r_p}. \tag{14}$$

Based on the previous expressions it may be concluded that the coefficient of relative engine power required in turn y_{mz} and the efficiency coefficient of the turning mechanism h_{mz} are variables depending on: motion resistance coefficient (f) of the ground on which the turn is realised; relative turning radius r ; relative fixed kinematic turning radius r_p and kinematic parameter q_M. For an ideal turning mechanism y_{mz} = 1 and h_{mz} = 1 would be achieved for every relative turning radius.

3. Impact Analysis of the Turning Mechanism

For the analysis the turning mechanisms have been selected that have the value q_M = 0.5 [5,6,7,8]. The turn of the vehicle on the horizontal sodded-grass surface is analysed with the assumed coefficients [1,4] f = 0.06 and $m_{\max}$ = 0.8. Graphical presentation of the calculation results of coefficients y_{mz} and h_{mz} for the vehicle T-34 is presented in Figures 3 and 4, for the vehicle T-55 in Figures 5 and 6, for the vehicle BVP M-80 in Figures 7 and 8 and for the vehicle M-84A in Figures 9 and 10.

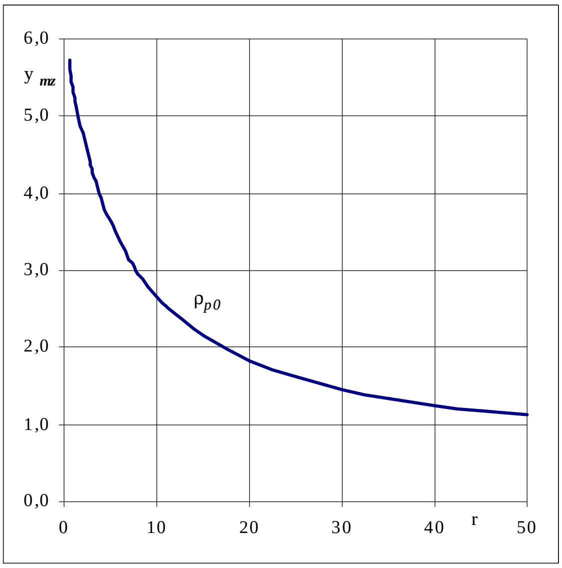

Figure 3 Coefficient of relative engine power required in turn for T-34

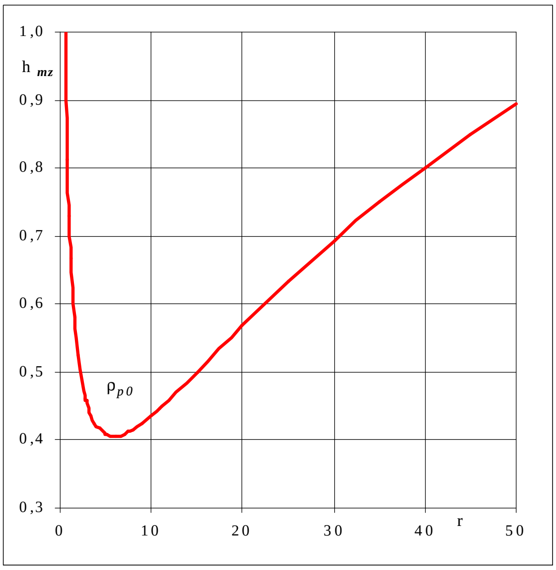

Figure 4 Efficiency coefficient of turning mechanism for T-34

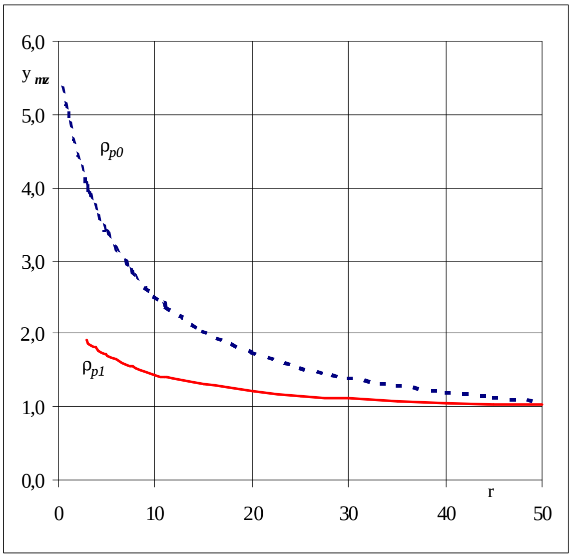

Figure 5 Coefficient of relative engine power required in turn for T-55

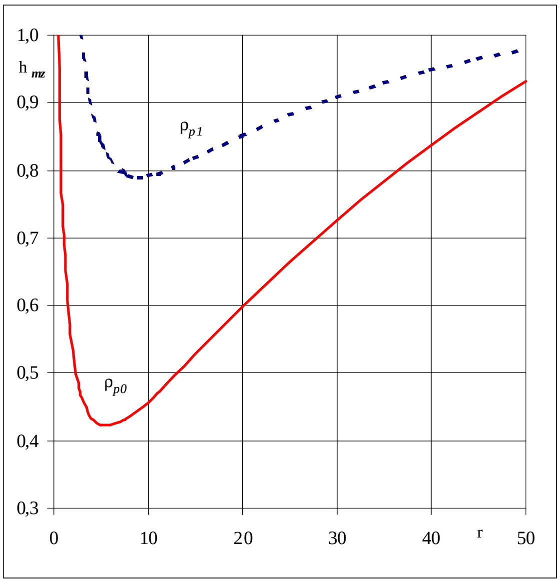

Figure 6 Efficiency coefficient of turning mechanism for T-55

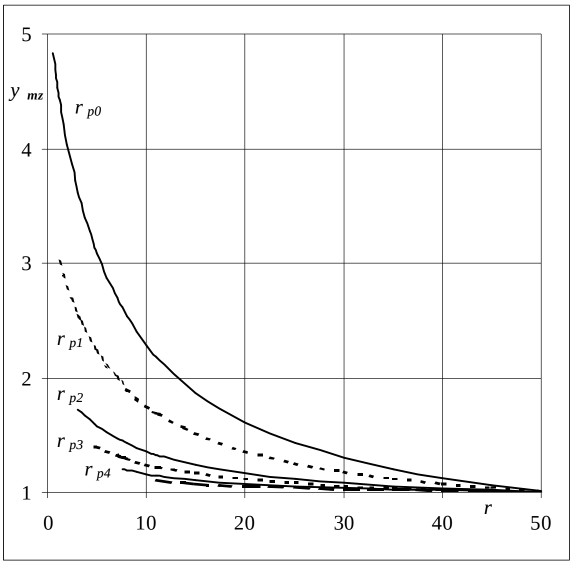

Figure 7 Coefficient of relative engine power required in turn for BVP-M80

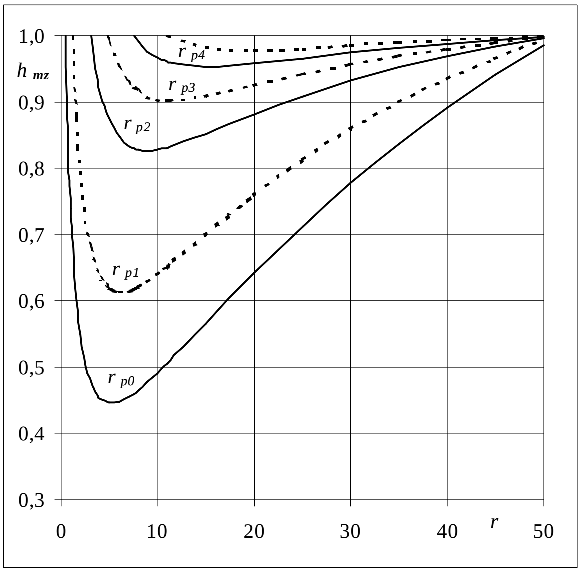

Figure 8 Efficiency coefficient of turning mechanism for BVP-M80

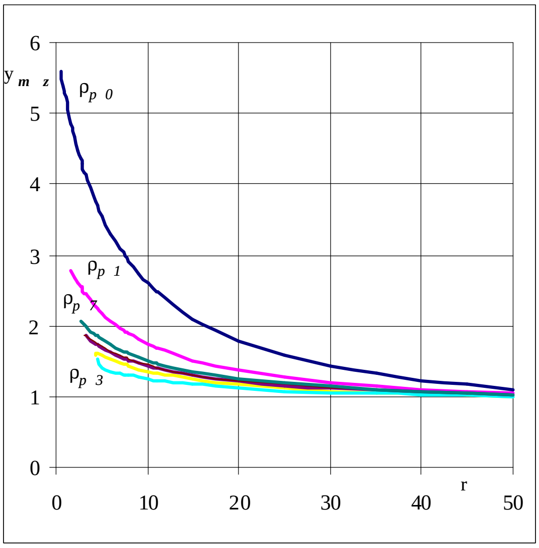

Figure 9 Coefficient of relative engine power required in turn for M-84

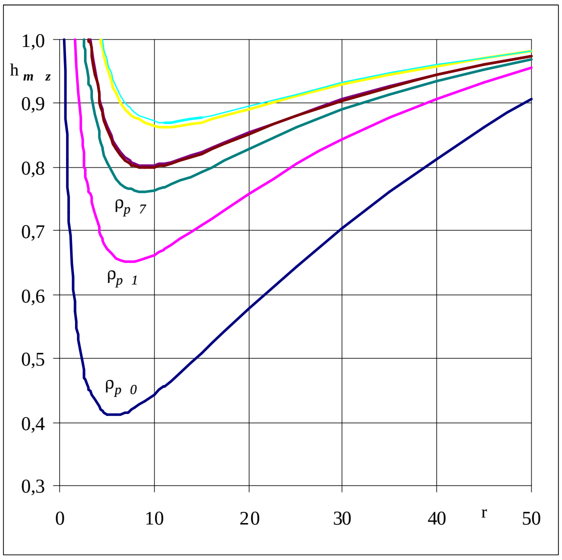

Figure 10 Efficiency coefficient of turning mechanism for M-84

Analysis of the obtained results shows that:

– the increase of relative fixed kinematic turning radius affects the decrease of the coefficient of relative engine power required in turn y_{mz} with increase in the efficiency coefficient of the turning mechanism h_{mz} for relative turning radii r which are greater than the relative fixed kinematic turning radius r_{pi} for single transmission levels,

– the decrease of coefficient y_{mz} and increase of coefficient h_{mz} is especially pronounced in the interval of relative fixed kinematic turning radius $r_{pi} \in \{1.0; 5.0\}$,

– for intervals of relative fixed kinematic turning radius $r_{pi} \in \{5.1; 15.0\}$ the trend of favourable values of coefficients y_{mz} and h_{mz} is more moderate,

- the existence of several relative fixed kinematic turning radii that are different in magnitude, allows transitioning of the turn by the tracked vehicle from a higher level (r_{pi}) to the nearest lower level (r_{pi-1}) for the case when turn needs to be performed with relative radius smaller than r_{pi}, and greater than r_{p0} = 0.5,
- transitioning of the turn made by a tracked vehicle from a higher level (r_{pi}) to the nearest lower level (r_{pi-1}) in case when turn needs to be performed with a relative radius smaller than r_{pi}, and greater than r_{p0} = 0.5, decreases the engine load and the load on friction elements of the turning mechanism.

4. Conclusion

The decrease in turning radius during the turn of a tracked vehicle leads to increased turning resistances and increased load on the engine and transmission devices of the vehicle. Major increase in the engine load results in turn with relative radii $r \leq 30$, and especially with relative radii $r \leq 15$. The existing problem can be overcome by designing the turning mechanism which apart from the minimal relative fixed kinematic turning radius r_{p0} = 0.5 has one or more relative fixed kinematic turning radii $r_{pi} \geq 0.5$.

The existence of the relative fixed kinematic turning radius greater than r_{p0} = 0.5 results in decrease of the engine load and transmission elements for the case of turns with turning radii equal or greater than the relative fixed kinematic turning radius. The coefficients of relative engine power required in a turn y_{mz} as well as efficiency coefficients of the turning mechanism h_{mz} have favourable values for those turning radii.

Favourable trend of the values of coefficients y_{mz} and h_{mz} is especially pronounced for the relative fixed kinematic turning radii in the interval $r_{pi} \in$ { 1.0; 5.0}. In the intervals of relative fixed kinematic turning radius $r_{pi} \in$ { 5.1; 15.0} the trend of favourable values of coefficients y_{mz} and h_{mz} is of more moderate intensity. In the range of turns with relative fixed kinematic turning radii greater than r_{pi} = 15, coefficients y_{mz} and h_{mz} do not change significantly.

When designing new transmissions of tracked vehicles, it is desirable to realise one relative fixed kinematic turning radius r_{p0} = 0.5 and several relative fixed kinematic turning radii in the interval $r_{pi} \in$ { 1.0; 15.0}.

References

[1] Farobin, J. E.: ’’Teoria povorota transportnih mašin’’, Mašinostroenie, Leningrad, 1970
[2] Krasnenkov V. I., Vašec, A. D.: ’’Proektirovanie planetarnih mehanizmov transportnih mašin’’, Mašinostroenie, Moskva, 1986
[3] Nosov, N. A., ...: ’’Rasčet i konstruirovanie guseničnih mašin’’, Mašinostroenie, Leningrad, 1972
[4] Zabavnikov, N. A.: ’’Osnovi teorii transportnih guseničnih mašin’’, Mašinostroenie, Moscow, 1975
[5] ***, Borbeno vozilo pešadije BVP M-80A, TU-I, SSNO, Beograd, 1989
[6] ***, Tenk T-34, TU-I, SSNO, Beograd, 1966
[7] ***, Tenk T-55, TU-I, SSNO, Beograd, 1966
[8] ***, Tenk M-84A, TU-I, SSNO, Beograd, 1988

Vjekoslav Stojković, MORH- Institut za obrambene studije, istraživanja i razvoj, Bijenička cesta 46, 10000 Zagreb, Croatia, vstojkovic@morh.hr
Ernest Bazijanac, Fakultet prometnih znanosti, Vukelićeva 4, 10000 Zagreb, Croatia, baza@fpz.hr
Mirko Jakopčić, GS OS RH – Hrvatsko vojno učilište ’’Petar Zrinski’’, Ilica 256b, 10000 Zagreb, Croatia, mjakopc@yahoo.com