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Defining the combat tracked vehicles turning radius in the function of dynamic characteristics

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One of the main problems emerging in the design of the combat tracked vehicles power trains is the way of matching turning radius with the straight-line motion dynamics. The procedure of determining the minimum turning radius values, depending on the vehicle straight-line motion dynamics, using two methods: the analytical method, as a newly developed one, and the graphic-analytical method, as an existing one, has been developed in this paper.

Key words: **combat vehicle, tank, armoured personnel carrier, infantry fighting vehicle, turning radius, analytical method, graphic-analytical method**

Introduction

TRACKED vehicle turning is mainly followed by simultaneous changes of track revolve speeds, tractive force values and tractive force direction on the inner track. Therefore, it is necessary that a power train contains a device or units providing a necessary ratio between the track revolve speeds, and an appropriate tractive force distribution on the tracks. In the power trains made of independent units, that function is performed by a special unit or units (there are two of them), known as the vehicle control mechanism, and in the composite power trains, as in block transmissions, that function is performed by transmission subunits or units, which the vehicle control mechanism consists of, also known as the auxiliary unit. In both cases, their function is to provide the steering control of the vehicle.

Therefore, the same term, the steering control system, will be used in the further text for both the vehicle control system and the steering control system, except in cases something is to be highly emphasized.

The vehicle turning can be obtained from the standpoint of kinematics or energy. In the first case, the kinematics of the steering control system, i.e. the vehicle control mechanism provides different gear ratios on the sprocket wheels, and also different track revolve speeds, while in the second case, a sprocket wheel (or elements connected to it), is stopped or slowed down from the inner track.

Special requirements for combat tracked vehicles are high mobility and cross-drive ability, and the achievement of high average vehicle speeds off road.

Therefore, it is necessary that the steering control system, i.e. the vehicle control mechanism provides a vehicle turning with a wide range of turning radius, at the maximum vehicle speed. This is important because turning makes most of the service life of a vehicle.

It is to provide a secure control of direction during the straight-line motion of the vehicle, especially when the vehicle drives at the maximum speed to the entering in the turn as well as after the outgoing from the turn.

On the basis of the presented requirements, it can be concluded that the best solution of the steering control system was the one providing the continual turning radius change at any engaged gear ratio in the gearbox, in a wide range and with the minimum decrease in the vehicle speed, if it is required by the conditions of movement.

That means the steering control of a combat tracked vehicle depends primarily on the applied design solution of the steering control system. However, the influence of the complete solution of the power train and the type of engine may not be ignored.

In case the vehicle has a built-in hydrodynamic transmission (HMT), with a gearbox having the hydrodynamic train, the vehicle speed decrease due to the increased turning resistance will yield automatically to the increase of the tractive force on the sprocket wheels. During the turning resistance decrease, the vehicle speed increases automatically, which significantly contributes to the increase of vehicle maneuverability.

In the same way, the turbine will act as an engine due to its high elasticity coefficient.

If the vehicle does not possess a sufficient power reserve during the straight-line motion, it will not be able to move at the same speed during the straight-line motion and the turn because the resistance is higher during the turn.

Selecting the steering control system and determining its basic parametres during the combat tracked vehicle design

As already mentioned, it is required that the steering control system of a combat tracked vehicle provides high tractive characteristics during the turn, and the total control of the straight-line motion and turning.

The question of the steering control system is solved together with the request that the vehicle, during the straight-line motion, should have high tractive and speed characteristics, and the secure control of direction, taking into consideration the category of the vehicle.

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It is known that high tractive characteristics and secure steering control during the straight-line motion and turning can provide transmissions with the gearbox, the gear ratio of which changes automatically and continually, providing the tractive force change along the ideal load hyperbola, and the steering control system with a forced regulation, providing the continual turning radius change.

In most modern combat tracked vehicles there are built-in HMTs, with the gear box having an automatic gear ratio change, the gear ratio of which changes in a wide range, and the steering control system providing the turning radius continual change by the means of the hydrostatic train.

Among certain advantages, these transmissions have not completely pushed out of use mechanical transmissions with a gradual gear ratio change and the steering control system providing one or more turning radius in each gear ratio.

In relation to the HMT, these transmissions do not provide turning under control with any of the radius (R), differing from the calculated one (R_p), which makes the steering control of the vehicle difficult. Every turning with $R \neq R_p$, with the mechanical transmission, is followed by the slipping of the control unit friction elements. On the basis of kinematics, all steering control systems can be divided into three groups:

- The first group consists of the steering control systems where a point situated outside the vehicle maintains the straight-line motion speed during the turn ($q_k > 0$, Fig.1a);
- The second group consists of the steering control systems where the outer track maintains the straight-line motion speed during the turn ($q_k > 0$, Fig.1b); and
- The third group consists of the steering control systems where a point situated between the longitudinal axes of the inner and outer tracks maintains the straight motion speed during the turn ($q_k > 0$, Fig.1c).

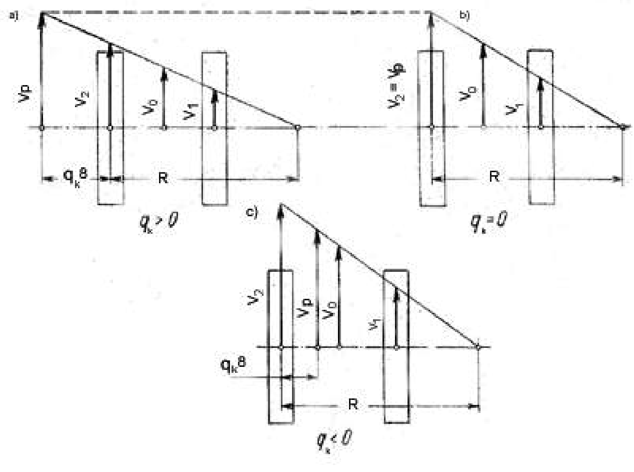

Figure 1. Diagram of the vehicle turning speed for different values of the parametre q_k

The parametre q_k is called the design parametre of the steering control system. It represents the distance between the point maintaining the straight-line motion speed and the longitudinal axis of the outer track, i.e. the point of attack of the resulting tractive force on the tracks, the force being provided by the engine.

In the first group ($q_k > 0$), the inner and outer track speeds and the vehicle gravity centre speed decrease during the turn. If the decrease of the vehicle centre speed is high (q_k has a high value), it may cause the engine muffling or stopping during the outgoing from the turn.

In the third-group of steering control systems ($q_k > 0$), the vehicle gravity centre speed decreases during the turn in relation to the straight-line motion speed when $q_k < -0.5$, while in all other cases is equal to ($q_k = -0.5$) or higher than ($0 > q_k < -0.5$) the straight-line motion speed.

Among these groups, the steering control systems with the parametre $q_k = 0$ (the outer track speed maintains the speed which the vehicle had during the straight-line motion, while the vehicle gravity centre speed of the inner track decreases) and $q_k = -0.5$ are mainly in use.

Mechanisms with side clutches, as well as single ratio, double ratio and multiratio planetary mechanisms belong to the steering control systems with $q_k = 0$, while the mechanisms based on simple, multiratio differential gears belong to the steering control systems with $q_k = -0.5$.

The following indicators can be used for general steering control system evaluation [1]:

1. Number and size of the calculated turning radius;
2. Parametre q_k characterized by the stability of turning and outgoing from the turn, engine load, control of direction during the straight-line motion;
3. Power losses due to friction, influencing the steering control system efficiency and friction elements wear;
4. Braking moment, acting upon friction elements;
5. Design solution complexity.

Based on the given expression for the required specific tractive force (f_z) during the turn:

$$f_z = f_P \frac{q_d + \rho_p}{q_k + \rho_p} \tag{1}$$

where

f_p – required specific tractive force during the straight-line motion under the same conditions as during the turn.

q_d – parametre determining the distance between the outer track axis and the point of attack of the resulting turning resistance force.

P_p – relative calculated turning radius:

$$\rho_p = \frac{R_p}{B}$$

R_p – calculated turning radius

B – track trails width (data given in Tables 1 and 2)

It is not difficult to conclude that the specific tractive force can be affected by the parametres P_p and q_k. With the increase of P_p and q_k, the required specific tractive force during the turn decreases. In order to reduce the required specific tractive force during the turn, the turning radius decrease is mainly adopted as the most acceptable solution, which leads to the worsening of vehicle maneuverability.

On the basis of the speed diagram (Fig.1), the ratio of the vehicle gravity centre speed (V_o) to the vehicle speed during the straight-line motion (V_p) preceeding the entering in the turn can be derived

$$\frac{V_0}{V_p} = \frac{\rho - 0,5}{\rho + q_k} \tag{2}$$

As seen in (2), by decreasing the relative turning radius and increasing the parametre q_k, the vehicle gravity centre speed is decreased. This decrease is maximal when $p=1$, i.e. when the turn is performed around the track. In that case, by the steering control system with the parametre $q_k > 0$, the vehicle gravity centre speed suddenly decreases, while in the steering control system with q_k =-0.5 the vehicle gravity centre speed is not decreased if the vehicle has a power reserve sufficient to overcome the increased resistances.

By decreasing the vehicle gravity centre speed, the tractive force required for turning decreases, while during the turn with small radius it suddenly increases.

The tractive force required for turning is higher in the differential-type system (q_k =0.5). In order to avoid a sudden increase of the required specific tractive force during the turn and stopping of the engine, it is recommended that the turning radius in the differential-type system is larger than the vehicle track trails width (p>1) and in the derived power trains it ranges within the limits p=2÷3.

The steering control systems with blocked one degree of freedom, provide a secure control of direction during the straight-line motion. The systems where q_k =0 belong to this group.

The differential-type steering control systems (q_k =-0,5), where one degree of freedom is disengaged due to tracks/ground contact, do not provide the secure control of direction during the straight-line motion. Because of possible different kinds of resistance to the tracks, most frequently while the vehicle moves on a rough terrain (slippery road, bad surface terrain, muddy road, etc.), a spontaneous turning or a turning off a direction can occur.

In modern tracked vehicles, the steering control systems providing one turning radius and one turn around the track are superseded. These systems were applied in the vehicles developing low average and maximum speeds ($V_{\max}$≤40÷50 km/h). Modern steering control systems provide the continual turning radius change (HMT) or one or more turning radii in each gear ratio, including also the turn around the track (mechanical or HMT).

Terrain conditions typical for combat action and, at the same time, the most difficult conditions for turning should be taken [2] for the calculation. The maximum turning resistance coefficient $\mu_{\max}$ =0.8÷0.85 and the rolling resistance coefficient f =0.07÷0.085 correspond to these conditions.

In order to provide the acceleration of the vehicle on such a terrain, it is necessary that the tractive force provided by the engine (F_M) is higher than the rolling resistance (F_R =fG).

The acceleration of the vehicle (a) during the straight-line motion is determined by the following expression [1]

$$a = \frac{g}{\delta}(f_M - f_U) \tag{3}$$

where

g – acceleration due to gravity;

δ – coefficient of rotation mass increment influence. It can be determined experimentally or on the basis of the empirical formulae from [2] and [4] for

- tanks

$$\delta = 1,2 + 0,002 i_{Ui}^2$$

- infantry fighting vehicles (IFVs) and armoured personnel carriers (APCs)

$$\delta = 1,2 + 0,0015 i_{Ui}^2$$

i_{Ui} – total gear ratio in the engaged gear ratio from the engine to the sprocket wheels;

f_M – specific tractive force during the straight-line motion, being provided by the engine [1]

$$f_M = \frac{360 P_M \eta_T \eta_g}{GV} \tag{4}$$

f_U – total rolling resistance coefficient

$$f_U = \sin\alpha + f\cos\alpha \tag{5}$$

α – angle of gradient;

P_M – engine power in kW;

η_T – power train efficiency from the engine to the sprocket wheels;

η_g – track efficiency. According to the expression in [2]

$$\eta_g = 0.95 - 0.005V$$

V – vehicle speed during the entering in the turn in km/h;

G – vehicle weight in daN.

Based on expression (3), the specific tractive force during the straight-line motion which is to be provided by the engine in order to realize acceleration a, can be determined:

$$f_M = a\frac{\delta}{g} + f_U \tag{6}$$

The calculated turning radius for certain terrain conditions is determined regarding the requirement that the vehicle is to perform a uniform turn. This requirement will be fulfilled if

$$f_M = f_Z \tag{7}$$

Based on the expression in [1], the following expressions are derived for the specific tractive force required for turning (f_Z) for

- symmetrical drive

$$f_{Z_{SP}} = \frac{f_{Z_2}\rho_p - f_{Z_1}(\rho_p - 1)\eta_R}{\rho_p} \tag{8}$$

- non-symmetrical drive

$$f_{Z_{NP}} = \frac{f_{Z_2}\rho_p - f_{Z_1}(\rho_p - 1)\eta_R}{\rho_p - 0,5} \tag{9}$$

where:

f_{Z1} and f_{Z2} – required specific tractive force which has to be performed on the inner or outer track

$$f_{Z_1} = \frac{f}{2} - \frac{\mu L}{4B} \tag{10}$$

$$f_{Z_2} = \frac{f}{2} + \frac{\mu L}{4B} \tag{11}$$

μ - turning resistance coefficient. According to [1]

$$\mu = \frac{\mu_{\max}}{0,85 + 0,15\rho}$$

L– length of track on ground (data given in Tables 1 and 2);

P– relative turning radius

$$\rho = \frac{R}{B}$$

R – turning radius;

η_R – recuperation efficiency (of the train from the inner to the outer track).

Table 1

Tanks	m [t]	L [mm]	B [mm]	L/B	GB [s^{-2}]
AMX-30	36	4120	2530	1.63	24.8
MERKAVA1	56	4520	2760	1.63	27
MERKAVA2	60	4780	2760	1.73	27
M1 ABRAMS	56	4650	2840	1.63	27.86
M-60	48.98	4235	2921	1.45	28.65
M-48	47.1	4000	2921	1.37	28.65
M-48 SUPER	53	4000	2921	1.37	28.65
M-47	46.1	3911	2794	1.4	27.4
M4 SHERMAN	31	3733	2108	1.77	20.67
LEOPARD 2	55	4945	2785	1.77	27.3
LEOPARD1	40	4236	2700	1.578	26.48
T-80	43	4270	2804	1.52	27.5
T-72	41	4270	2790	1.53	27.36
T-62	40	4160	2640	1.57	25.89
T-55	36	3840	2640	1.454	25.89
T-54	36.36	3840	2640	1.454	25.89
T-34	32	3850	2450	1.57	24
Pz-68	38	4220	2590	1.629	25.4
Pz-71	38	4130	2590	1.594	25.4
CHALLENGER	62	4800	2720	1.76	26.68
CHIEFTAIN Mk3, Mk5	54.1	4800	2718	1.76	26.68
VICKERS VALIANT	43.6	4470	2700	1.655	26.48
VICKERS Mk3	39.1	4280	2533	1.689	24.84
VICKERS Mk1	38.6	4280	2533	1.689	24.84
CENTURION	62	4570	2690	1.76	26.68

Table 2

APC and IFV	m [t]	L [mm]	B [mm]	L/B	GB [s^{-2}]
STEYR 4k 7FA	14.8	3192	2120	1.5	20.79
VCI	27	3900	2620	1.48	25.7
BMP-1	12.6	3460	2550	1.356	25
BMP-2	-	3912	2439	1.6	23.9
MCV-80	-	3816	2540	1.5	24.9
VCC-80	-	3792	2450	1.55	24
AMX 10P	15	3012	2159	1.39	21.1
XM2	21.3	3911	2440	1.6	23.9
MARDER	28.2	3900	2620	1.48	25.7
M 113	11.1	2667	2159	1.23	21.1
IFV M80	13.85	3300	2526	1.3	24.78
APC M60	10.7	2940	2380	1.23	23.34
Typ 73	13.5	3380	2500	1.55	20.1
Typ Su 60	11.8	3180	2050	1.55	20.1

If expressions (8) and (9) are applied on a conditioned steering control system (the steering control system providing all turning radii as the calculated ones) we should put ρ instead of ρ_p.

While defining turning radius, we should pay particular attention to the relation between the radius and the speed at which the vehicle enters in the turn.

It is well-known that the tracked vehicle turning may be followed by a partial or complete skidding.

The speed at which the partial skidding occurs during the turn is determined by the expression given in [2] for

– symmetrical drive

$$V_P \geq \sqrt{\mu_{\max} gB(\rho - 0.5)} \tag{12}$$

– non-symmetrical drive

$$V_P \geq \frac{\rho}{\rho - 0.5}\sqrt{\mu_{\max} gB(\rho - 0.5)} \tag{13}$$

and during the complete skidding it is determined by the expression given in [2] for

– symmetrical drive

$$V_P \geq \sqrt{\mu_{\max} gB(\rho - 0.5)} \tag{14}$$

– non-symmetrical drive

$$V_P \geq \frac{\rho}{\rho - 0.5}\sqrt{\mu_{\max} gB(\rho - 0.5)} \tag{15}$$

The analytical method for determining the turning radius

This is a new method and it should represent certain contribution to the theory of tracked vehicle turning. The procedure of determining the turning radius according to this method is the following:

Starting from the equality given in (7), by equating expressions for f_M (6) and f_Z (8) or (9), a final expression for calculating the relative turning radius in the form of quadratic equation is obtained for

– symmetrical drive

$$a\rho^2 + b\rho + c = 0 \tag{16}$$

– non-symmetrical drive

$$a_1\rho^2 + b_1\rho + c_1 = 0 \tag{17}$$

where the coefficients are

$$a = 0.6f_M - 0.3f_U(1+\eta_R)$$

$$b = 3.1f_M - f_U(1.7+1.4\eta_R) - \mu_{\max}\frac{L}{B}(1-\eta_R)$$

$$c = 1.7(f_U - f_M) - \mu_{\max}\frac{L}{B}$$

$$a_1 = 0.6f_M - 0.3f_U(1+\eta_R)$$

$$b_1 = 3.4f_M - f_U(1.7+1.4\eta_R) - \mu_{\max}\frac{L}{B}(1-\eta_R)$$

$$c_1 = \eta_R(1.7f_U - \mu_{\max}\frac{L}{B})$$

By solving equations (16) and (17) by ρ, the relative turning radius for the given calculated conditions is obtained.

While selecting turning radii for combat tracked vehicles, it is also necessary to take into consideration dynamic vehicle characteristics during the straight-line motion, as well as competent terrain conditions.

In order to determine turning radii in some gear ratios, the previous determination of speed at which the vehicle enters in the turn is necessary.

In this case, while determining the speed, it is started from the expression for the specific tractive force during the straight-line motion, the force being obtained with the power corresponding to the rated engine speed, i.e. from the ideal load hyperbola equation, as follows [1]:

$$f_p = \frac{360P_{M\max}\eta_g\eta_T}{GV} \tag{18}$$

where $P_{M\max}$ – engine power at the rated speed

Since η_g is the vehicle speed function, the final expression for determining V is obtained by introducing the substitution for η_g in expression (18) and equating f_P and f_M

$$V=\frac{342P_{M\max}\eta_T}{f_M G+1.8P_{M\max}\eta_T} \tag{19}$$

The speed V represents the straight-line abscissa, $\rho=f(V)$, which can be written as a straight-line equation through two points. For symmetrical drives, those are points 0 (0,0.5) and 0 (V,ρ_0), and for non-symmetrical drives 0 (0,0) and (V,ρ_0). The straight-line equation is obtained by the substitution of these values. When $V_{i\max}$ is introduced in the straight-line equation instead of an unknown abscissa, the expressions for the relative turning radius calculation are obtaned for

- symmetrical drive

$$\rho=\frac{\rho_0-0.5}{V}V_{i\max}+0.5 \tag{20}$$

- non-symmetrical drive

$$\rho=\frac{\rho_0}{V}V_{i\max} \tag{21}$$

where $V_{i\max}$ is the maximum vehicle speed which can be developed in the gear ratio i.

When the values for ρ_o and V are determined by expressions (16) and (19) for symmetrical drives, i.e. by (17) and (19) for non-symmetrical drives, and the maximum speeds are introduced in some gear ratios instead of $V_{i\max}$, on the basis of expressions (20) and (21), the values for turning radii in some gear ratios are obtained.

The graphic method for determining the turning radius

Turning radii can also be determined by using the graphic method from [1], the procedure of which is much longer. In order to apply this method, it is necessary to start from some concrete data. A tank that weighs 43000 daN, with an engine of 735 kW at 2000 min^{-1}, where L/B =1.52, η_R =0.7, η_T =0.9, with built-in side gearboxes (Fig.2), obtaining four gear ratios for going ahead, with the following gear ratios:

- The first gear ratio i_I = 16.82
- The second gear ratio i_{II} = 8.312
- The third gear ratio i_{III} = 5.62
- The fourth gear ratio i_{IV} = 3.85

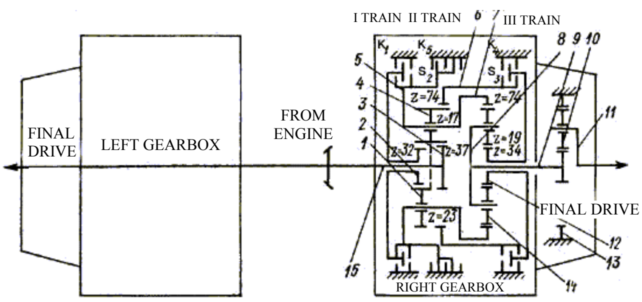

Figure 2. Kinematic scheme of the power train with side gearboxes

and an IFV that weighs G =13600 daN, with an engine of 238 kW at n=2500 min^{-1}, where L/B =1.3, η_R =0.7, η_T=0.92, with a built-in block transmission (Fig.3) obtaining five gear ratios for going ahead and the same number of turning radii with the following total gear ratios:

- The first gear ratio i_I = 25.6
- The second gear ratio i_{II} = 12
- The third gear ratio i_{III} = 8.1
- The fourth gear ratio i_{IV} = 5.46
- The fifth gear ratio i_V = 3.74

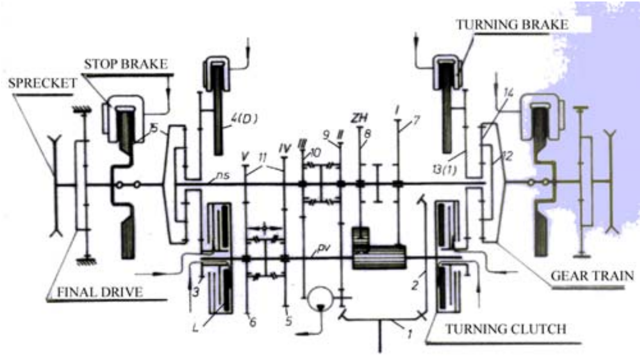

Figure 3. Kinematic scheme of the block transmission

In both cases, it is assumed that the vehicles are used under the same conditions where f =0.08, $\mu_{\max}$ =0.8, α =0^0, and the vehicle acceleration is a=0.4 m/s^{-2}. δ=1.4 will be adopted for the tank and δ =1.25 for the IFV.

The operation processes are the following. First of all, it is necessary to determine f_M, according to expression (6) for

- tank

$$f_M=a\frac{\delta}{g}+f_U=0.4\frac{1,4}{9.81}+0.08=0.137$$

- IFV

$$f_M=0.4\frac{1.25}{9.81}+0.08=0.13$$

In case of the power train design phase, when turning radii are unknown, it is necessary to determine the specific tractive force for turning, the force being provided by the conditioned steering control system (the steering control system providing all turning radii as the calculated ones) according to expression (8) in the case of symmetrical drive, i.e. according to expression (9) in the case of non-symmetrical drive.

In Table 3, the values in the cases of symetric and non-symmetric drive have been calculated.

Vehicle	Tank				IFV			
Relative turning radius ρ	f_{Z1}	f_{Z2}	Symmetrical drive	Non-symmetrical drive	f_{Z1}	f_{Z2}	Symmetrical drive	Non-symme-trical drive
			f_{ZSP}	f_{ZNP}			f_{ZSP}	f_{ZNP}
1	0.264	0.344	0.686	0.344	0.22	0.3	0.6	0.3
2	0.244	0.304	0.30	0.225	0.181	0.261	0.263	0.197
5	0.15	0.23	0.164	0.148	0.122	0.202	0.148	0.133
10	0.089	0.169	0.1188	0.1129	0.070	0.150	0.11	0.105
20	0.038	0.118	0.095	0.0927	0.027	0.107	0.091	0.089
30	0.0166	0.0966	0.0867	0.0853	0.0084	0.088	0.0837	0.0823
40	0.004	0.084	0.0848	0.083	0.0016	0.078	0.07788	0.0769
50	-0.0035	0.0764	0.0796	0.0788	-0.0033	0.0766	0.0773	0.0766

When the values for f_Z are known, the specific tractive force curves required for turning are drawn. Since in both cases there are transmissions with non-symmetrical drive, the data for f_{ZNP} will be taken during the drawing. In Fig.4, the specific tractive force curves are shown in the coordinate system (ρ, f_M, i.e. f_Z). If we draw horizontal lines from the point f_M =0.137 (for tanks), i.e. f_M =0.13 (for IFVs) in the intersection with the specific tractive force hyperbola for turning, the points (the ordinates of which represent relative turning radii) will be obtained.

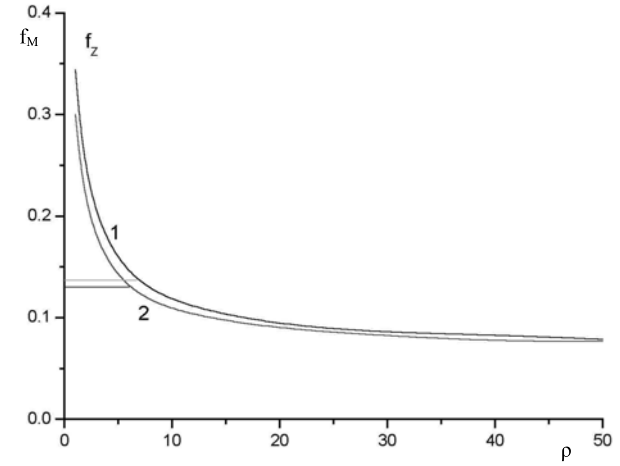

Figure 4. Diagram of the specific tractive force change during the turn: 1 – tank, 2 – IFV

The obtained values from the diagram are ρ =6.76 for the tank, and ρ =6.09 for the IFV.

The same values determined by the analytical method are: ρ =5.85 for the tank, and ρ=5.41 for the IFV. The difference between the values obtained graphically and analytically occurs due to the correction of the hyperbola.

In order to determine the speed at which the vehicle turns with this radius, it is necessary to draw an ideal load hyperbola. Its points will be determined on the basis of expression (18). The obtained values for the tank and IFV for the above mentioned data, are given in Table 5.

Table 5

Speed before entering in the turn V [km/h]	Specific tractive force during the straight-line motion f_p	
	Tank	IFV
3	1.591	1.642
5	0.944	0.974
10	0.459	0.474
20	0.217	0.223
30	0.136	0.140
40	0.0957	0.0988
50	0.0715	0.073
60	0.0553	0.057
70	0.0437	0.0451
80	0.0351	0.0362

When the ideal load hyperbolae 1 and 2 (Fig.5) are drawn from the abscissae f_M = 0.137, i.e. f_M = 0.130, according to expression (7), horizontal lines are drawn to the intersection with the hyperbola 1, i.e. 2.

The points of intersection abscissae represent speeds of entering in the turn. In this way, the speeds: V =30.2 km/h (for the tank), i.e. V=31.1 km/h (for the IFV) are obtained.

In order to define turning radii in some gear ratios, the traction diagram is reqired. The tank traction diagram is drawn in Fig.6, and the IFV traction diagram is drawn in Fig.7.

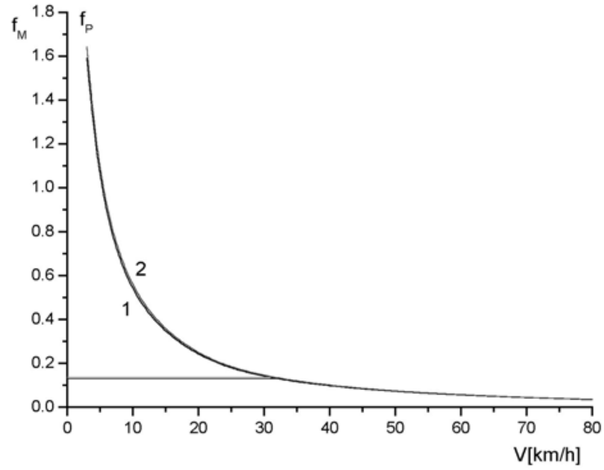

Figure 5. Diagram of the ideal load hyperbola during the straight-line motion: 1 – tank, 2 – IFV

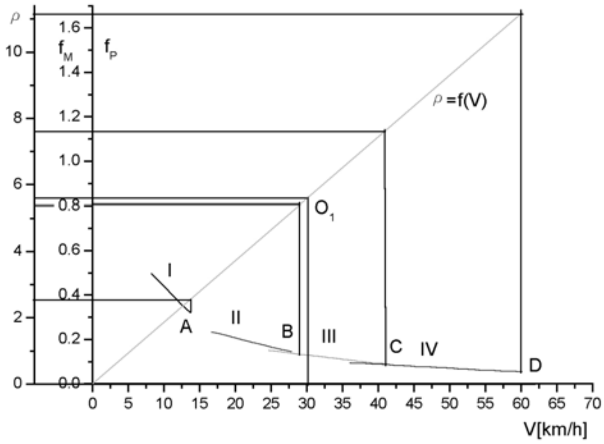

Figure 6. Tank traction diagram

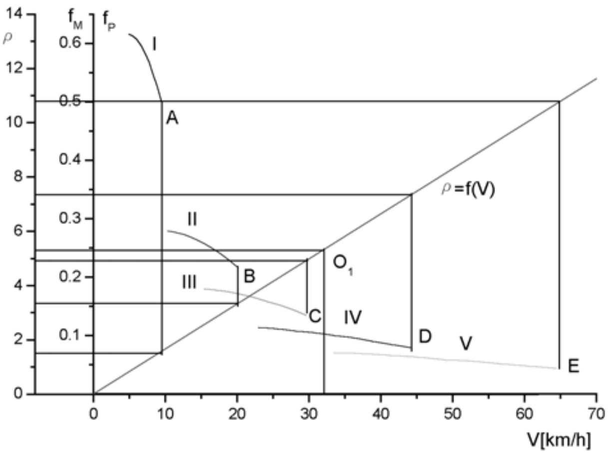

Figure 7. IFV traction diagram

The specific tractive force values, the force being provided by the engine and calculated by expression (18), are on the diagram ordinate. The specific tractive force values, the force being calculated according to the following expression, are on the vehicle speed abscissa in some gear ratios (v_i)

$$V_i = 0.377 \frac{r_{pt} n_M}{i_{Ti}} \left[\text{km/h}\right] \tag{22}$$

where

r_{pt} – sprocket wheel radius (r_{pt}=0.306 m for tank, and r_{pt}=0.256 m for IFV)

n_M – engine revolutions number

i_{Ti} – total gear ratio in some gear ratios from the engine to the sprocket wheels

On the same diagram, the change of ρ is drawn as another ordinate. Now, it is necessary to draw a straight line $\rho=f$(V), as a straight line through two points: [0(0;0), 0_1(29.83;5.85)] for the tank, i.e. [0(0;0),(32.02;5.41)] for the IFV.

From the tractive curve final points A, B ,C, D and E, corresponding to the maximum speeds in some gear ratios, verticals should be drawn to the section with the horizontal line ρ=f(V). In that way, the points, the ordinates of which represent the minimum relative turning radii values in some turning radii, are obtained. Therefore, the following values are obtained for:

- tank: ρ_I =2.6; ρ_{II} =5.34; ρ_{III} =7.6; ρ_{IV} =11.6
- IFV: ρ_I =1.4; ρ_{II} =3.3; ρ_{III} =4.9; ρ_{IV} =7.25; ρ_V =10.6

In relation to the analytically obtained values, determined by expression (20) for

- tank: ρ_I =2.62; ρ_{II} =5.32; ρ_{III} =7.86; ρ_{IV} =11.48
- IFV: ρ_I =1.59; ρ_{II} =3.375; ρ_{III} =5; ρ_{IV} =7.46; ρ_V =10.88

the differences are negligible.

The relative turning radii real values, provided for these vehicles by the built-in steering control systems, are:

- tank: ρ_I =1; ρ_{II} =1.97; ρ_{III} =3.08; ρ_{IV} =3.17
- IFV: ρ_I =1.65; ρ_{II} =3.51; ρ_{III} =5.21; ρ_{IV}=7.77; ρ_V =11.33

By comparing these values with the previous ones, it is seen that in the case of the tank, i.e. the transmission with side gearboxes, there is a big difference between the recommended and real turning radius value, while in the case of a block transmission in the IFV that difference is negligible.

The main reason for this state is that the gear ratios in a gearbox are determined on condition that the required kinematics and dynamics during the straight-line motion are provided. Since the turning radii of side gearboxes are a gear ratio function in gearboxes, and the gearboxes are composed of planetary gear trains, which requires the fulfillment of certain conditions, it is not possible to obtain the turning radius calculated values but considerably lower ones.

Because of this kind of turning radii differences, the turn will undoubtedly be executed with a partial skidding in lower gear ratios, at the maximum speeds, and with a complete skidding in higher gear ratios. If tanks did not have to move along an exactly determined path, which is the most frequent case, this problem would not be so expressed. However, if the tank drives along the exactly determined path of limited width, a narrow winding road through gorges and on a craggy terrain, the vehicle dynamics will be imperilled considerably because the tank will not move at the regimes of complete engine dynamic characteristics utilization, but it will have to move at partial regimes.

In the case of block transmission, the gear ratios in the gearbox are determined independently from the auxiliary unit, so it is possible to correct the turning ratio by selecting suitable auxiliary unit gear ratios.

However, one should keep in mind that the tracked vehicle turning with the mechanical transmission, at higher engaged gear ratios (V>40 km/h), is mainly executed with the partial skidding.

Conclusion

On the basis of the above presented, the following conclusions can be derived:

- proper approach to the turning radius selection is very important for obtaining the required vehicle maneuverability and dynamics
- turning radii may not be determined according to need, independently from the adopted power train conception
- for defining turning radii, in the vehicle and power train design phase, two methods can be used: the analytical method and the graphic-analytical method
- shown analytical method has been developed on the basis of the existing graphic-analytical method and its basic advantage is the shortening of the work process since it is not necessary to draw a large number of diagrams

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Definisanje poluprečnika zaokreta borbenih guseničnih vozila u funkciji dinamičkih karakteristika

Jedan od osnovnih problema koji se javlja kod projektovanja sistema za prenos snage borbenih guseničnih vozila je kako uskladiti poluprečnike zaokreta sa dinamikom pravolinijskog kretanja. U radu je razrađen postupak određivanja minimalnih vrednosti poluprečnika zaokreta u zavisnosti od dinamike pravolinijskog kretanja vozila korišćenjem dve metode jedne analitičke, kao novorazvijene, i druge grafoanalitičke, kao postojeće.

Ključne reči: **borbena vozila, tenk, oklopni transporter, borbeno vozilo pešadije, poluprečnik zaokreta, analitička metoda, grafoanalitička metoda.**

Analyse du rayon de virage chez les véhicules chenillés de combat en fonction des caractéristiques dynamiques

La coordination des rayons de virage avec la dynamique du mouvement rectiligne est un problème important pendant la création des systèmes de transmission chez les véhicules chenilles de combat. L'article donne un procédé de détermination des valeurs minimales du rayon de virage en fonction de la dynamique du mouvement rectiligne du véhicule à l'aide de deux méthodes: une méthode graphique et analytique, déjà existante, et l'autre nouvelle et analytique.

Mots-clés: **véhicules de combat, char, véhicule transporteur de troupe, véhicule de combat d'infanterie, rayon de virage, méthode analytique, méthode graphique et analytique.**