International Journal of Heavy Vehicle Systems
DOI: 10.1504/IJHVS.2012.046833

The working regime analysis of a track-type tractor

Ticuşor Ciobotaru*

Military Technical Academy,
81-83 G. Cosbuc Bvd.,
Bucharest 050141, Romania
E-mail: cticusor2004@yahoo.com
*Corresponding author

Daniel Frunzeti, Ioan Rus and Lorentz Jäntschi

Technical University of Cluj-Napoca,
103-105 Muncii Bvd.,
Cluj-Napoca 400641, Romania
E-mail: danfrunzeti@yahoo.com
E-mail: Ioan.Rus@yahoo.com
E-mail: lori@academicdirect.org

Abstract: The estimation of the vehicle acceleration performance plays an important role in the assessment of the vehicle mobility. The estimation becomes difficult in the case of tracked vehicles owing to the complexity of their power pack (consisting mainly of the engine and the hydromechanical transmission) and also because of the tracked running gear. Because of that, this case was taken into investigation and an algorithm was proposed. The algorithm is focused on the estimation of the acceleration performances. The accuracy of the proposed algorithm is compared with the experimental data resulted from the testing of a 50 tonnes tracked vehicle running up to 60 km/h, concluding that the model-based data are in good agreement with the experimental results.

Keywords: tracked vehicle; acceleration; mobility; torque converter.

Reference to this paper should be made as follows: Ciobotaru, T., Frunzeti, D., Rus, I. and Jäntschi, L. (2012) 'The working regime analysis of a track-type tractor', *Int. J. Heavy Vehicle Systems*, Vol. 19, No. 2, pp.172–187.

Biographical notes: Ticuşor Ciobotaru received his BSc (1979), MSc (1991) and PhD (1995) in Mechanical Engineering from the Military Technical Academy of Bucharest. In 1998 he received his degree in Security Studies from George C. Marshall European Center, and in 2005, his MD in Military and Security Studies from the War College of National Defense University, Washington DC. He is affiliated to several professional bodies, such as the Society of Automotive Engineers (USA) and the Romanian Automotive Engineers Society. His expertise covers the area of tracked vehicle engineering and military vehicle testing.

Daniel Frunzeti received his BSc (2003), MSc (2005) and PhD (2011) in Mechanical Engineering from the Technical University of Cluj-Napoca. He obtained his MD degree in Political Sciences from the 'Lucian Blaga'

University of Sibiu. He is affiliated to the Romanian General Engineers Association, and the Romanian Automotive Engineers Society. His research area of interest includes military vehicle testing, braking systems for vehicle, international relations, security systems.

Ioan Rus received his BSc (1982), MSc (1985) and PhD (1994) in Mechanical Engineering from the Technical University of Cluj-Napoca. He is currently a Professor in Automotives Engineering at the same university. His research focuses on safety systems and sustainable development. In particular, he is interested in braking systems for vehicles, renewable energy sources, biogas and diesel engines for urban public transportation. He is affiliated to several professional bodies, such as the Society of Automotive Engineers (USA), the Romanian General Engineers Association and the Romanian Automotive Engineers Society.

Lorentz Jäntschi received his BSc (1995) in Informatics, his second BSc (1997) in Chemistry and Physics from the 'Babeş-Bolyai' University, his MSc (2002) in Management in Agriculture, his PhD (2010) in Horticulture from the University of Agricultural Sciences and Veterinary Medicine and his second PhD (2000) in Organic Chemistry from the 'Babeş-Bolyai' University of Cluj-Napoca. He has been a Faculty Member since 1999 and Associate Professor at the Technical University of Cluj-Napoca from 2008. His research area of interest includes applied informatics, statistics and probabilities, and mathematical modelling. He has been working on several sponsored projects from national funding agencies.

1 Introduction

A broad range of models exists to predict the longitudinal dynamics of the tracked vehicles. Because the longitudinal dynamics of the tracked vehicles includes the acceleration performances, the ride and the track–soil interaction, the scientific works focused on these topics accordingly. All these aspects have a direct impact on the mobility of the vehicles (Korlath, 2007).

The range of models begins with the point mass models embedded in the automotive performance simulations that are more focused on powertrain performance (McCullough et al., 2006; Schmid et al., 1998; Xiaoping and Shu, 2002; Rakha et al., 2001), to commercial rigid body dynamics codes that model the individual track blocks and the detailed contact interactions of the running gear elements (Sandu and Freeman, 2005; Slattengren, 2000; Ma and Perkins, 2002). Between these extremes are well-known modelling tools such as the NATO Reference Mobility Model (NRMM) vehicle dynamics (VEHDYN) module and several commercial and custom purpose-driven modelling tools (Priddy, 1995).

There are many papers describing the intensive works on modelling the longitudinal dynamics taking into consideration the limitation of the tractive effort due to the soil (Wong and Wei, 2005; Kim et al., 2005).

Some models focus on the control of the traction in correlation with the soil's characteristics and the powertrain output (Zhejun et al., 1994; Kang et al., 2009).

A fast and robust algorithm for assessment of the tracked vehicles acceleration performances is useful both for the rough estimation of the acceleration performances and for the estimation of the actual speed the tracked vehicle can achieve in the considered condition, as preliminary data needed for the detailed study of the running gear dynamics.

Because of the complexity of the models, several specialised programs were designed, such as MOSES (Ferretti and Girelli, 1999), NTVPM (Wong and Wei, 2005) or commercial software were used to develop new applications (Rubinstein and Hitron, 2004; Kang et al., 2009).

2 The modelling of the engine output

The engine represents the source of power used by the vehicle to overcome the resistance opposed by internal friction of the running gear, soil deformation, slope and inertia. The output characteristics of the engine are fully defined by the variation of the output power and the output torque vs. angular speed and engine load. For a given engine, the output characteristics are provided as diagrams or as tables. To use the engine's output characteristics for the determination of the vehicle's acceleration performances, it is convenient to obtain an analytical form of these characteristics. The following polynomial form was found as acceptable, after several structures of relationships have been tested in terms of the accuracy of modelling the experimental data of the engine:

$$p_e(n_e,\Psi) = \frac{1}{P_m} \cdot \left[A + B \cdot \left(\frac{n_e}{n_m}\right) + C \cdot \left(\frac{n_e}{n_m}\right)^2 + D \cdot \Psi + E \cdot \Psi^2 + F \cdot n_e \cdot \Psi \right] [-], \qquad (1)$$

where the following notations were used: A, B, C, D, E, F – constant coefficients, n_e – the actual rpm of the engine, Ψ – the load of the engine (for full load of the engine $\Psi = 1.0$), P_m – the maximum output power of the engine, which occurs for the speed n_m and $\Psi = 1.0$, P_e – the actual rated power of the engine for the actual speed n_e and the actual load Ψ.

The maximum value, $p = 1$, corresponds to $n_e = n_m$ and $\Psi = 1.0$. This form was preferred because it allows the modification of the output power within a margin of ±15% assuming that the allure of the curve remains unchanged. This easiness is useful in studying the influence of the engine output maximum power on the acceleration performances.

The actual power delivered by the engine at the speed n_e and the load Ψ is given by:

$$P_e = P_m \cdot p_e, \qquad (2)$$

where P_e represents the actual output power of the engine.

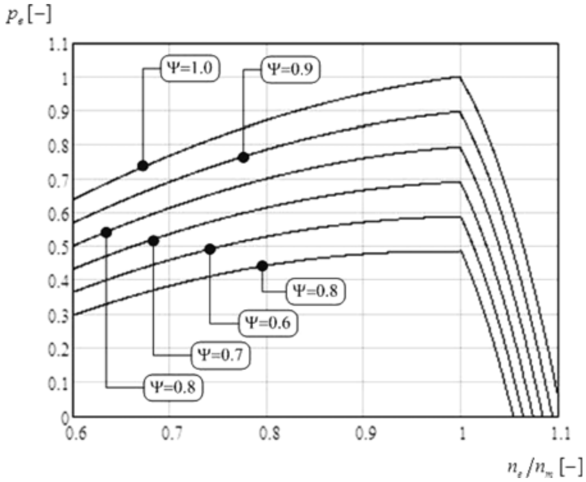
For the simulation included into this paper, a Diesel engine with 610 kW of maximum output power at 2300 rpm was selected. To calculate the coefficients included in formula (1), the least square fit method was applied to experimental data; finally, the following form of equation (1) has resulted:

$$Pe(n,\psi) = \frac{1}{610}\Big(-522.875 + 0.57241 \cdot n_e - 1532 \cdot 10^{-7} \cdot n_e^2$$
$$+85.43\psi + 5.8667\psi^2 + 0.23305 \cdot n_e \cdot \psi\Big).$$

For the engine taken into consideration, the injection pump includes a mechanical speed regulator, which cut-off the fuel for speeds greater than the maximum speed of the engine, n_m. For the engine speeds greater than the maximum speed, n_m, a linear drop of the output power is adopted. Consequently, the equation of the rated power for speed greater than n_m has resulted.

The graphical representation of the engine output power's characteristics is presented in Figure 1; the allure of the curves is parabolic followed by a linear drop for the engine speeds greater than the maximum speed.

Figure 1 The characteristics of the engine output power



The output torque of the engine (noted T_e) for the speed n_e and the load Ψ are calculated using the following formula:

$$T_e = \frac{30}{\pi \cdot n_e} \cdot P_e(n_e, \Psi). \tag{3}$$

To find the value of the engine maximum output torque, the following equation having the unknown n_e is solved:

$$\frac{\partial}{\partial n_e} T_e(n_e, 1.0) = 0. \tag{4}$$

Consequently, the value of the speed (noted n_1), which corresponds to the maximum value of the torque, T_1, results. Thus, it becomes possible to calculate the value of the maximum torque T_1:

$$T_1 = \frac{30}{\pi \cdot n_1} \cdot P_e(n_1, 1.0). \tag{5}$$

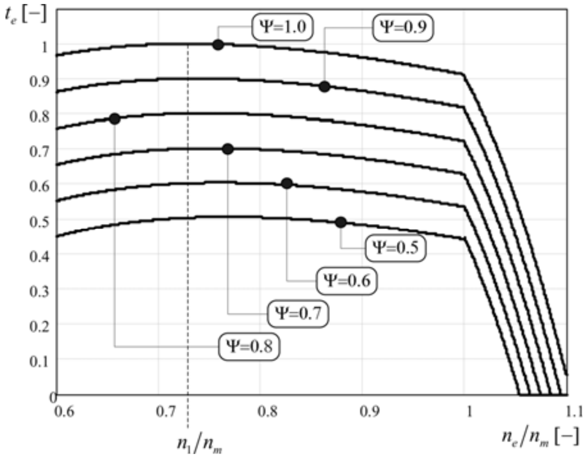
Finally, the rated actual torque output characteristic is done by the following expression:

$$t_e = \frac{30}{\pi \cdot n_e \cdot T_1} \cdot P_e(n_e, \Psi) = \frac{n_1}{n_e} \cdot \frac{P_e(n_e, \Psi)}{P_e(n_1, 1.0)}, \tag{6}$$

where t_e represents the rated actual torque.

The engine output torque characteristics for the considered vehicle are presented in Figure 2.

Figure 2 The characteristics of the engine output torque



The fan of the cooling system of the engine represents the main internal consumer of energy; for the maximum speed of the engine, the power absorbed by the cooling system fan represents about 8–12% of the engine maximum output power; for the considered tracked vehicle, the characteristics of the absorbed power of the fan are calculated using the following formula, based on experimental data, which give a maximum 10.8% absorbed power by the cooling system fan:

$$p_{\text{fan}} = 0.108 \cdot \left(\frac{n_e}{n_m} \right)^3 [–]. \tag{7}$$

Finally, the amount of power transmitted to the torque converter is calculated by subtracting from the engine output power the amount of power used by the cooling system fan:

$$p_u = p(n_e, \Psi) - p_{\text{fan}}(n_e) = \frac{1}{P_m} \cdot \left[A + B \cdot \left(\frac{n_e}{n_m} \right) + C \cdot \left(\frac{n_e}{n_m} \right)^2 - 0.108 \cdot \left(\frac{n_e}{n_m} \right)^3 + D \cdot \Psi + E \cdot \Psi^2 + F \cdot n_e \cdot \Psi \right]. \tag{8}$$

Consequently, the value of the torque (T_u), which may be used for traction, is calculated with the following formula:

$$T_u = \frac{30}{\pi \cdot n_e} \cdot P_u(n_e, \Psi) = \frac{30}{\pi \cdot n_e} \cdot P_m \cdot p_u(n_e, \Psi). \tag{9}$$

3 The modelling of the torque converter output

Almost all modern tracked vehicles include a torque converter into the hydromechanical transmission. The torque converter provides a continuous variation of the torque ratio within each stage of the mechanical gearbox, and allows faster and automatic shifting. The main variable of the torque converter is the ratio between the impeller speed (noted n_{impller}) and the turbine speed (noted n_{turbine}):

$$i'_h = \frac{n_{\text{impeller}}}{n_{\text{turbine}}}, \quad i'_h \in [0, 1.0]. \tag{10}$$

This ratio, also named the torque converter slip coefficient, becomes null for total slip of the torque converter, and takes a value as high as 1.0 for the locked up torque converter.

The torque ratio, noted K_h, varies from a maximum value corresponding to the total slip of the turbine $(i'_h = 0)$ to a minimum value equal to 1.0 when the speed of the turbine becomes equal to the speed of the impeller:

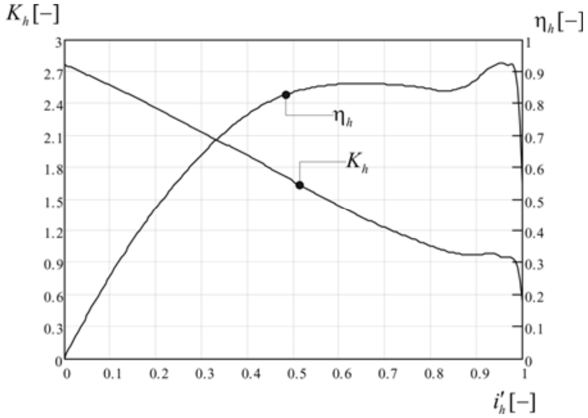
$$K_h = \frac{T_{\text{impeller}}}{T_{\text{turbine}}}, \quad K_h \in [1.0, K_{h\max}]. \tag{11}$$

The torque converter efficiency (noted η_h) is calculated as follows:

$$\eta_h = \frac{P_{\text{output}}}{P_{\text{input}}} = \frac{T_{\text{impeller}}}{T_{\text{turbine}}} \cdot \frac{n_{\text{impeller}}}{n_{\text{turbine}}} = K_h \cdot i'_h. \tag{12}$$

The characteristics of the torque converter considered for the simulation are presented in Figure 3.

Figure 3 The characteristics of the torque converter



The torque absorbed by the impeller depends on the equivalent hydraulic diameter of the torque converter (noted D_h), the mass density of the fluid (noted ρ_h), the coefficient λ_p, which takes into consideration the influence of the turbine torque on the impeller torque and the speed of the impeller (Lechner and Naunheimer, 1999):

$$T_{\text{impeller}} = \left(\lambda_p \cdot \rho_h \cdot D_h^5\right) \cdot n_{\text{impeller}}^2. \tag{13}$$

For the torque converter considered, the diameter D_h and the density of the fluid ρ_h are constants, and the variation of the expression within brackets depends only on the coefficient λ_p, and is graphically presented in Figure 4.

Figure 4 The absorbed torque characteristics of the torque converter

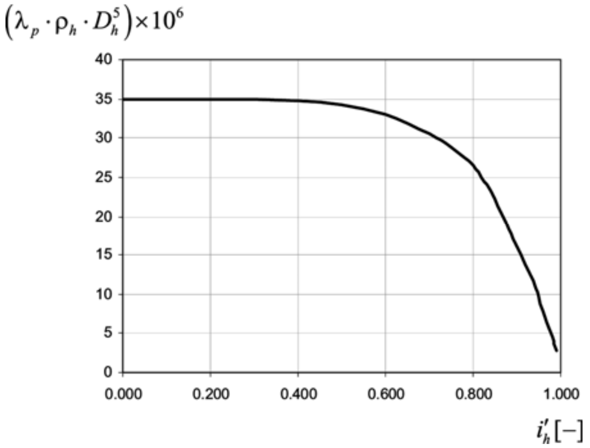
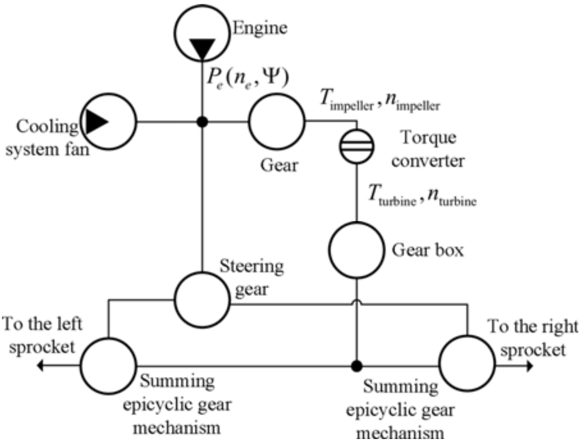


Figure 5 details the power flow diagram of the typical power pack used for the tracked vehicles.

Figure 5 Power flow diagram of the power pack



The gear interlaid between the engine and the torque converter has the ratio i_{r0} and the efficiency η_{r0}; this gear allows the matching of the engine output torque and the torque absorbed by the impeller. Providing a gear ratio below 1.0, it is possible to decrease the diameter of the torque converter for the same amount of the transmitted power.

The torque provided by the engine through the gear becomes:

$$T_{\text{impeller}} = T_u \cdot i_{r0} \cdot \eta_{r0}. \tag{14}$$

The speed of the impeller becomes:

$$n_{\text{impeller}} = \frac{n_e}{i_{r0}}, \tag{15}$$

where, for the considered transmission, $i_{r0} = 0.848$.

From equations (13)–(15), and taking into consideration formula (9), the following equation is obtained:

$$(\lambda_p \cdot \rho_h \cdot D_h^5) \cdot \left(\frac{n_e}{i_{r0}}\right)^2 = \frac{30}{\pi \cdot n_e} \cdot P_m \cdot p_u(n_e, \Psi) \cdot i_{r0} \cdot \eta_{r0}. \tag{16}$$

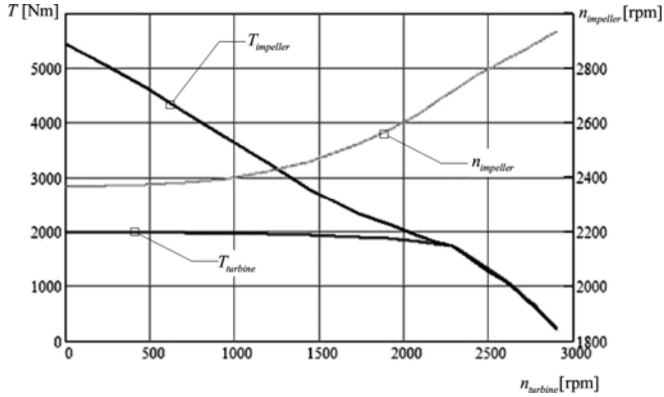
The solutions of equation (16) represent the matching points of the engine and the torque converter. Solving equation (16) for $\Psi = 1$, and for the values of i_h' varying from 0 up to 1.0, the values of the engine speed result. For each considered value $\left(i_h'\right)_j$, using relations (8), (9), (14) and (15), the values of the input torque and speed for the torque converter (noted $(T_{\text{impeller}})_j$ and $(n_{\text{impeller}})_j$, respectively) result. For these points, the output parameters of the torque converter become:

$$(n_{\text{turbine}})_j = (n_{\text{impeller}})_j \cdot (i_h')_j \tag{17}$$

$$(T_{\text{turbine}})_j = (T_{\text{impeller}})_j \cdot (K_h)_j. \tag{18}$$

Finally, the torque converter output characteristics are obtained; the graphical representation of these characteristics is shown in Figure 6.

Figure 6 The output characteristics of the torque converter (see online version for colours)



The output characteristics of the torque converter permit the calculation of the available traction effort of the sprockets. For the straight line running of the vehicle, the steering gear mechanism provides the blocking of the sun gears of the summing epicyclic gears. Consequently, the output power of the torque converter is equally divided between the two sprockets. The data regarding the transmission ratios and efficiencies considered for simulation are summarised in Table 1.

Table 1 The parameters of the transmission components

Transmission component		Ratio		Efficiency	
		Symbol	*Value*	*Symbol*	*Value*
Gear box	1st stage	i_{GB_1}	4.640	η_{GB_1}	0.973
	2nd stage	i_{GB_2}	2.600	η_{GB_2}	0.976
	3rd stage	i_{GB_3}	1.667	η_{GB_3}	0.982
	4th stage	i_{GB_4}	1.000	η_{GB_4}	1.000
Summing epicyclic gear mechanisms		i_{SM}	1.345	η_{SM}	0.991
Final gears		i_{FG}	3.88	η_{FG}	0.974

The tractive effort developed by the vehicle is calculated using the following relationship:

$$F_T = \frac{T_{\text{turbine}} \cdot i_{GB_j} \cdot i_{SM} \cdot i_{FG}}{r_S} \cdot \eta_{GB_j} \cdot \eta_{SM} \cdot \eta_{FD}, \tag{19}$$

where r_s represents the rolling radius of the sprockets. The speed of the vehicle is done by the following relationship:

$$V = 3.6 \cdot \frac{\pi \cdot n_{\text{turbine}} \cdot r_S}{30 \cdot i_{GB_j} \cdot i_{SM} \cdot i_{FG}} [\text{Km/h}]. \tag{20}$$

Considering the total weight of the vehicle $W = 500{,}000$ [N], the coefficient of the tractive effort (noted c_T) is defined as the ratio between the tractive effort and the vehicle weight:

$$c_T = \frac{F_T}{W} = \frac{T_{\text{turbine}} \cdot i_{GB_j} \cdot i_{SM} \cdot i_{FG}}{W \cdot r_S} \cdot \eta_{GB_j} \cdot \eta_{SM} \cdot \eta_{FD}. \tag{21}$$

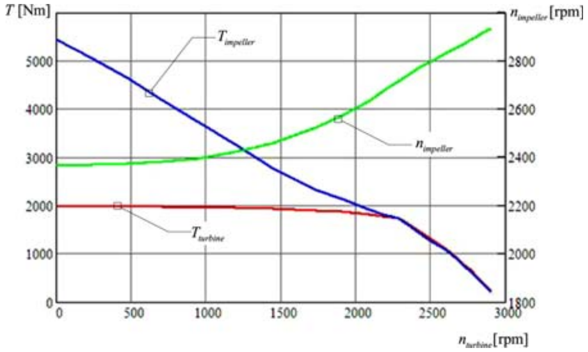
The next step consists in the calculation of the variation of the tractive effort coefficient vs. the vehicle speed; the outcome and the calculations are presented graphically in Figure 7.

4 The estimation of the rolling resistances

The rolling resistances coefficient includes the coefficient of internal resistance of the track system (noted f_m), the coefficient of resistance due to the soil compaction (noted f_{soil}), as well as the resistance due to the slope (defined by the angle α); consequently, the coefficient of the total resistance (noted c_R) is done by the following sum:

$$c_R = f_{\text{in}} + f_{\text{soil}} \cdot \cos\alpha + \sin\alpha. \tag{22}$$

Figure 7 Tractive effort coefficient vs. vehicle speed characteristics (see online version for colours)



For the firm, hard soil and horizontal terrain, relation (22) yields:

$$c_R \approx f_{\text{in}}. \tag{23}$$

There are a large variety of empirical relations for the calculation of the internal resistance coefficient of the track system. The generalised form of these relations is a polynomial function with the speed of the vehicle as variable:

$$c_R = C_0 + C_1 \cdot V + C_2 \cdot V^2, \tag{24}$$

where C_0, C_1 and C_2 represents constants.

For the purpose of the present simulation, the double pad, rubber-bushed pins type of track is considered, giving the following values of the coefficients: $C_0 = 0.03$, $C_1 = 143.10^{-6}$ and $C_2 = 1.2.10^{-6}$.

It is obvious that the motion of the vehicle becomes possible once the condition $c_T \geq c_R$ is fulfilled. The excess of tractive effort is used for the acceleration of the vehicle, overcoming its inertia.

5 The estimation of the acceleration performances

The balance of the forces acting on the tracked vehicle yields the following equation (Ogorkiewicz, 1991):

$$F_T = F_R + F_{\text{inertia}}, \tag{25}$$

where: F_T – tractive effort, F_R – resistance force, F_{inertia} – inertia force due to the acceleration of the vehicle having the total weight W, and the acceleration of the running gear components (tracks, road wheels, idle wheels, sprockets) and of the power pack components (gears, shafts, torque converter, engine shaft, etc.).

Ogorkiewicz (1991) proposes the following formula for expressing the inertia force:

$$F_{\text{inertia}} = (1+\delta) \cdot \frac{W}{g} \cdot \frac{\mathrm{d}v}{\mathrm{d}t}, \tag{26}$$

the coefficient δ taking into consideration the influence of the inertia of the moving components of the vehicle (Ogorkiewicz, 1991).

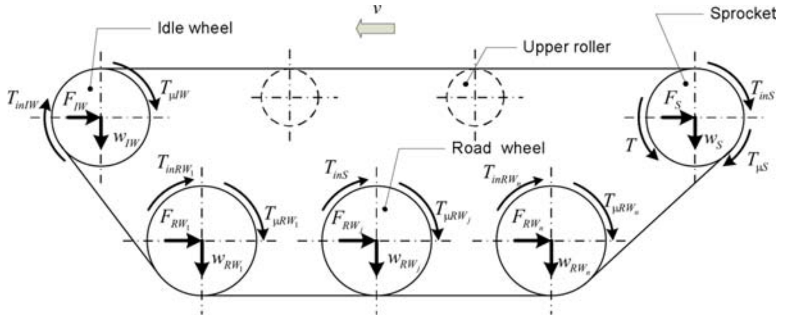
Taking into consideration relation (26), equation (25) may be re-written as:

$$c_T = c_R + \frac{1+\delta}{g} \cdot \frac{\mathrm{d}v}{\mathrm{d}t}. \tag{27}$$

Equation (27) allows the calculation of the acceleration performances of the tracked vehicle.

The inertia mass factor δ takes into consideration the inertia of the track system and of the rotating items within the power pack (engine, torque converter and transmission). To calculate the contribution of the track system's components to the inertia mass factor, the schematisation presented in Figure 8 is considered.

Figure 8 Schematisation of the track system



The schematisation of the track system consists in the representation of the track, the idle wheel and of the *n* road wheels. On each wheel acts the mass gravitational force, noted *w*, which is included into the vehicle's total mass, and is balanced by the normal force of the terrain. The friction torques, noted T_μ, as well as the internal friction of the track are taken into consideration through the rolling resistance coefficient. In the same time, inertia acts on each element; the sum of the inertial moments reduced at the axle of the sprocket is done by:

$$F_{\text{track system}} = \left(\frac{J_{IW}}{r_{IW}^2} + \frac{J_S}{r_S^2} + \cdot \sum_{j=1}^{n} \frac{J_{RW_j}}{r_{RW_j}^2} + \frac{w_{\text{track}}}{g} \right) \cdot \frac{\mathrm{d}v}{\mathrm{d}t}, \tag{28}$$

where $g = 9.81$ m/s^2, r – rolling radii of the running gear wheels, J – moments of inertia of the running gear wheels, IW – index for the idle wheels, S – index for the sprockets, RW – index for the road wheels, w_{track} – the weight of the track.

Taking into consideration the possible existence of the upper rollers, relation (28) may be re-written in a generalised form:

$$F_{\text{track system}} = \left(\frac{w_{\text{track}}}{g} + \sum_{j=1}^{N} \frac{J_j}{r_j^2} \right) \cdot \frac{\mathrm{d}v}{\mathrm{d}t}. \tag{29}$$

The moment of inertia due to the power pack is calculated with the following relation:

$$F_{\text{power pack}} = \left(J_{\text{turbine}} \cdot \frac{\mathrm{d}\,\omega_{\text{turbine}}}{\mathrm{d}\,t} + J_{\text{impeller}} \frac{\mathrm{d}\,\omega_{\text{impeller}}}{\mathrm{d}\,t} \cdot K_h \right) \cdot \frac{i_{GB_i} \cdot i_{SM} \cdot i_{FD} \cdot \eta_{GB_i} \cdot \eta_{SM} \cdot \eta_{FD}}{r_S^2} \cdot \frac{\mathrm{d}\,v}{\mathrm{d}\,t}, \tag{30}$$

where J_{impeller} represents the total inertia moment of the items connected to the impeller shaft (engine, intermediate gear, etc.), and J_{turbine} include the inertia moments of the transmission.

Equation (30) is re-written as:

$$F_{\text{power pack}} = \left(J_{\text{turbine}} + J_{\text{impeller}} \frac{\dfrac{\mathrm{d}\,\omega_{\text{impeller}}}{\mathrm{d}\,t}}{\dfrac{\mathrm{d}\,\omega_{\text{turbine}}}{\mathrm{d}\,t}} \cdot K_h \right) \cdot \frac{\mathrm{d}\,\omega_{\text{turbine}}}{\mathrm{d}\,t} \cdot \frac{i_{GB_i} \cdot i_{SM} \cdot i_{FD} \cdot \eta_{GB_i} \cdot \eta_{SM} \cdot \eta_{FD}}{r_S^2} \cdot \frac{\mathrm{d}\,v}{\mathrm{d}\,t},$$

and taking into consideration that:

$$\frac{\mathrm{d}\,\omega_{\text{turbine}}}{\mathrm{d}\,t} = i_{GB_i} \cdot i_{SM} \cdot i_{FD} \cdot \frac{\mathrm{d}\,v}{\mathrm{d}\,t},$$

and:

$$\frac{\mathrm{d}\,\omega_{\text{turbine}}}{\mathrm{d}\,t} = \frac{\pi}{30} \cdot \frac{\mathrm{d}\,n_{\text{turbine}}}{\mathrm{d}\,t}; \quad \frac{\mathrm{d}\,\omega_{\text{impeller}}}{\mathrm{d}\,t} = \frac{\pi}{30} \cdot \frac{\mathrm{d}\,n_{\text{impeller}}}{\mathrm{d}\,t},$$

finally, relation (30) becomes:

$$F_{\text{power pack}} = \left(J_{\text{turbine}} + J_{\text{impeller}} \frac{\mathrm{d}\,n_{\text{impeller}}}{\mathrm{d}\,n_{\text{turbine}}} \cdot K_h \right) \cdot \frac{(i_{GB_i} \cdot i_{SM} \cdot i_{FD})^2 \cdot \eta_{GB_i} \cdot \eta_{SM} \cdot \eta_{FD}}{r_S^2} \cdot \frac{\mathrm{d}\,v}{\mathrm{d}\,t}. \tag{31}$$

From relations (29) and (31), the final relation for the calculation of inertia mass factor is obtained:

$$\delta = \frac{g}{W}(F_{\text{track system}} + F_{\text{power pack}}), \tag{32}$$

$$\delta = \frac{w}{W}\left(\frac{\text{track}}{g} + \sum_{j=1}^{N} \frac{J_j}{r_j^2} \right) g + \frac{g}{W}\left(J_{\text{turbine}} + J_{\text{impeller}} \frac{\mathrm{d}\,n_{\text{impeller}}}{\mathrm{d}\,n_{\text{turbine}}} \cdot K_h \right) \frac{(i_{GB_i} \cdot i_{SM} \cdot i_{FD})^2 \cdot \eta_{GB_i} \cdot \eta_{SM} \cdot \eta_{FD}}{r_S^2}. \tag{33}$$

The following notations are introduced:

$$i_{\text{power pack}} = i_{GB_i} \cdot i_{SM} \cdot i_{FD};$$

$$\eta_{\text{power pack}} = \eta_{GB_i} \cdot \eta_{SM} \cdot \eta_{FD}.$$

Using the above-listed notation, relation (33) becomes:

$$\delta = \frac{g}{W}\left(\frac{w_{\text{track}}}{g} + \sum_{j=1}^{N} \frac{J_j}{r_j^2}\right)$$

$$+\frac{g}{W}\left(J_{\text{turbine}} + J_{\text{impeller}} \frac{\mathrm{d}\,n_{\text{impeller}}}{\mathrm{d}\,n_{\text{turbine}}} \cdot K_h\right) \frac{i_{\text{powerpack}}^2 \cdot \eta_{\text{powerpack}}}{r_S^2}.$$

The first term of the above-mentioned expression does not depend on the overall ratio of the transmission and represents the contribution of the running gear components (track, sprockets, wheels, etc.) to the inertia mass factor. The second term takes into consideration the contribution of the engine and transmission.

For the tracked vehicles with mechanical transmission, the experimental data indicates for the inertia mass factor the following relation (Zabavnikov, 1975):

$$\delta = (0.2 \div 0.4) + (0.002 \div 0.003) \cdot i_{\text{transmission}}^2. \tag{34}$$

Taking into consideration relations (33) and (34), as well as the average values of the moment of inertia for the items connected to the impeller of the torque converter, and identifying the terms of the expressions, the final relation for mass inertia factor is concluded:

$$\delta = 0.2 + \left(0.0002 + 0.005 \cdot K_h \cdot \frac{\mathrm{d}\,n_{\text{impeller}}}{\mathrm{d}\,n_{\text{turbine}}}\right) \cdot i_{\text{power pack}}^2. \tag{35}$$

For the locked up torque converter, relation (35) yields:

$$\delta = 0.2 + 0.0052 \cdot i_{\text{power pack}}^2 \tag{36}$$

The derivative, which appears in formula (35), was calculated numerically, using the cubic spline interpolation of the variation of the impeller speed vs. turbine speed, graphically presented in Figure 6.

From relation (27), the resulted expression for the calculation of the vehicle acceleration is:

$$\frac{\mathrm{d}\,v}{\mathrm{d}\,t} = \frac{g}{1+\delta}(c_T - c_R). \tag{37}$$

Relation (37) allows the estimation of the time required to accelerate the vehicle from the speed v_1 to the speed v_2:

$$t = \int_{v_1}^{v_2} \frac{1+\delta}{g \cdot (c_T - c_R)} \mathrm{d}\,v. \tag{38}$$

The distance that the vehicle travels during the acceleration from the speed v_1 to the speed v_2 can be calculated by re-writing equation (37) as:

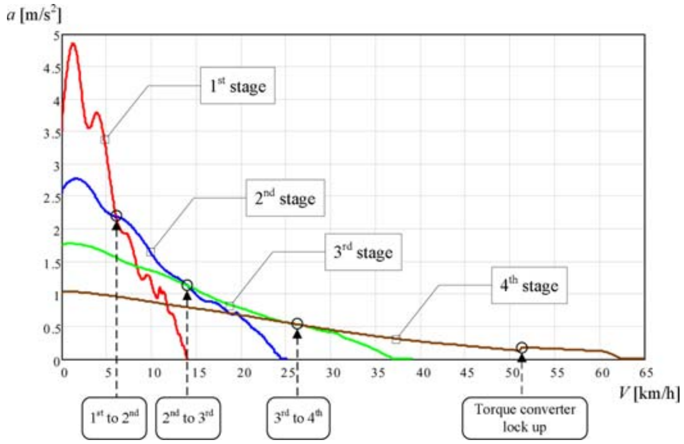
$$\frac{\mathrm{d}v}{\mathrm{d}t} = \frac{\mathrm{d}S}{\mathrm{d}t} \cdot \frac{\mathrm{d}v}{\mathrm{d}S} = v \cdot \frac{\mathrm{d}v}{\mathrm{d}S} = \frac{g}{1+\delta}(c_T - c_R). \quad (39)$$

The following relation is concluded:

$$S = \int_{v_1}^{v_2} \frac{v \cdot (1+\delta)}{g \cdot (c_T - c_R)} \mathrm{d}v. \quad (40)$$

Because the achievement of the speed v_2 may imply the gearbox shifting, it is necessary to define the shifting strategy. If the maximum acceleration performance is desired, the speed at which the shifting of the gearbox is scheduled must be adopted so that the acceleration has the maximum value. The principle of the method is graphically presented in Figure 9.

Figure 9 The acceleration characteristics (see online version for colours)



It is mentioned that, according to relation (21), the tractive coefficient depends on the turbine speed (which is proportional to the vehicle speed). The rolling resistance coefficient depends on the vehicle's speed too. Meanwhile, the factor of the mass inertia has a complex dependence of the regime of the torque converter and of the transmission total ratio.

Consequently, the integration of equation (37) is performed numerically. The results represent the acceleration performances of the vehicle expressed in terms of speed vs. time. The acceleration characteristics are presented in Figure 10.

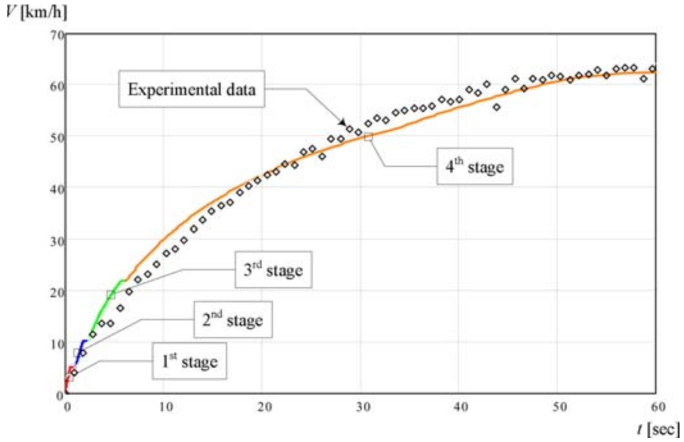
To assess the efficiency of the above-presented algorithm, experimental tests were carried out. The tests consisted in the repetitive acceleration of the vehicles running on hard terrain, up to the maximum speed; the vehicle's speed was measured by optical encoders fitted on the shafts of the sprockets. The pulse signals generated by the optical encoders are decoded by frequency/voltage converters, the resulting signal being recorded by a data acquisition system.

The experimental data are presented in Figure 10 for comparison with the calculated data.

The analysis of the experimental data demonstrated that the proposed algorithm provides an acceptable accuracy for the estimation of the acceleration performances.

The main sources of errors of the simulation are the mass inertia factor and the rolling resistance. Nevertheless, using the proposed relations for the estimation of the mass inertia factor and for the rolling resistance coefficient, an acceptable accuracy for simulation was obtained.

Figure 10 Acceleration performances of the tracked vehicle (see online version for colours)



6 Conclusions

The proposed algorithm was easily implemented into a MathCad application, but any equivalent mathematics software may be used. The input data includes the main design and working parameters of the vehicle, strictly needed for the determination of the working regime of the track-type vehicle. The relations are simple, and stress the influence of the main design parameters of the vehicle on working regime.

The simulation of the vehicle running on concrete way demonstrated an acceptable accuracy of the algorithm compared with the experimental data.

If the load of the engine is considered as variable, the algorithm is appropriate for real-time simulation of the tracked vehicle running.

For a specific type of vehicle, the accuracy of the simulation could increase if the rolling resistance coefficient and the inertia mass factor are experimentally measured.

References

Ferretti, G. and Girelli, R. (1999) 'Modelling and simulation of an agricultural tracked vehicle', *Journal of Terramechanics*, Vol. 36, No. 3, July, pp.139–158.

Kang, O., Park, Y. and Park, Y. (2009) 'Look-ahead preview control application to the high-mobility tracked vehicle model with trailing arms', *Journal of Mechanical Science and Technology*, Vol. 23, pp.914–917, ISSN: 1738-494X.

Kim, M.H., Yi, S.J. and Wei, Y. (Eds.) (2005) 'Dynamic analysis of the high speed tracked vehicles', *International Conference on Mechatronics and Information Technology 2005, Proceedings of SPIE, the International Society for Optical Engineering*, Vol. 6042, pp.60424D.1–60424D.5, 0-8194-6074-5.

Korlath, G. (2007) 'Mobility analysis of off-road vehicles: Benefits for development, procurement and operation', *Journal of Terramechanics*, Vol. 44, No. 5, November, pp.383–393.

Lechner, G. and Naunheimer, H. (1999) *Automotive Transmissions. Fundamentals, Selection, Design and Application*, Springer-Verlag, Berlin, ISBN 3-540-65903-X.

Ma, Z.D. and Perkins, N.C.A. (2002) 'Track-wheel-terrain interaction model for dynamic simulation of tracked vehicles', *Vehicle System Dynamics*, Vol. 37, No. 6, pp.401–421, ISSN: 0042-3114.

McCullough, M., Khubchandani, A., Shyu, S. and Simoni, D. (2006) 'Verification and fidelity of high mobility tracked vehicle dynamic models', *Paper 2006-01-0928 presented at SAE 2006 World Congress & Exhibition*, 3–7 April, Detroit, MI, USA.

Ogorkiewicz, R.M. (1991) *Technology of Tanks*, Jane's Information Group Limited, United Kingdom, ISBN 0-7106-0595-1.

Priddy, J.D. (1995) *Stochastic Vehicle Mobility Forecasts using the NATO Reference Mobility Model, Report 3 – Database Development for Statistical Analysis of the NRMM II Cross-Country Traction Empirical Relationships*, Technical Report 95-8, U.S. Army Corps of Engineers Waterways Experiment Station.

Rakha, H., Lucic, I. and Demarchi, S.H. (2001) 'Vehicle dynamics model for predicting maximum truck acceleration levels', *Journal of Transportation Engineering*, Vol. 127, No. 5, pp.418–425, ISSN: 0733-947X.

Rubinstein, D. and Hitron, R. (2004) 'A detailed multi-body model for dynamic simulation of off-road tracked vehicles', *Journal of Terramechanics*, Vol. 41, Nos. 2–3, April–July, pp.163–173, 14th International Conference of the ISTVS.

Sandu, C. and Freeman, J.S. (2005) 'Military tracked vehicle model. Part II: case study', *International Journal of Vehicle Systems Modelling and Testing*, Vol. 1, Nos. 1–3, pp.216–231.

Schmid, I.C., Ehlert, W. and Pott, S. (1998) 'Mobility of tracked vehicles and simulation of the dynamics of motion on the paisi test plant', *Paper 885114 presented at SAE 1998 World Congress & Exhibition*, 5–7 October, Detroit, MI, USA.

Slattengren, J. (2000) 'Utilization of adams to predict tracked vehicle performance', *Paper 2000-01-0303 presented at SAE 2000 World Congress*, 6–9 March, Detroit, MI, USA.

Wong, J.Y. and Wei, H. (2005) 'Evaluation of the effects of design features on tracked vehicle mobility using an advanced computer simulation model', *International Journal of Heavy Vehicle Systems*, Vol. 12, No. 4, pp.344–365, ISSN: 1744-232X.

Xiaoping, B. and Shu, H. (2002) 'Transient modelling for heavy tracked vehicle performance and fuel consumption under accelerating conditions', *Paper 2002-01-0630 presented at SAE 2002 World Congress & Exhibition*, 4–7 March, Detroit, MI, USA.

Zabavnikov, N.A. (1975) Osnovi teorij transportnih gusenicinih mashin, Moscow, Nauka.

Zhejun, F., Yoran, K. and David, W. (1994) 'Traction control and modelling of tracked vehicles', *Paper 942375 presented at SAE Brazil '94*, 7–9 October, Sao Paulo, Brazil.