The drivetrain and the steering mechanism for the twin engines tracked vehicle

R Yu Dobretsov, G P Porshnev, E G Sakharova, N N Demidov, I A Komarov and D E Telyatnikov

St. Petersburg and Peter the Great St. Petersburg Polytechnic University, Polytechnicheskaya 29, 195251, St. Petersburg, Russia

E-mail: dr-idpo@yandex.ru

Abstract. The kinematic plan of the double flow transmission and the steering mechanism for the twin engines tracked vehicle is proposed. It could be used for the compact remotely operated vehicle or autonomous tracked chassis with twin electric engines. However, the plan is applied for skid steering wheeled vehicles or heavy transport vehicles. This plan makes it possible to use the parallel/series type hybrid propulsion system. In this case the powertrain is divided into two separate flows: heat engine with a gearbox and the reversible traction electric engine. The principle could be adapted for designing ships with diesel-electric hybrid propulsion system (it is possible to use internal combustion engine instead of electric engine), for the reduction unit modernization in the motor boat, and for special purpose ships design. Simplicity, portability, engine duplication, potential for energy recuperation during breaking/turning and providing turning around the center of gravity are the main advantages of this plan. The last two points are true for skid steering wheeled vehicles. The operating modes, the main kinematic and power aspects are provided. The required power of the electric engine traction in the hybrid transport tracked vehicle (weighing 45-50 tons) is estimated. The production of the hybrid propulsion system with proposed transmission based on Russian components is considered.

1. Introduction

Transport vehicles mobility is limited by the controllability and stability of the movement. To improve the steering control quality of tracked vehicles is a crucial task. It is necessary to provide a turning radius smooth variation and correspondence between the steering control position and the turning radius in the simplest way. It is desirable to ensure a turn in place and create conditions for power recuperation.

The aim of the work is to obtain kinematic transmission plan, which will provide high quality steering control of the tracked vehicle.

Let's consider the concept of twin engine car transmission (Figure 1). Both engines could be electric (the option that can be implemented for the compact tracked robot chassis). The proposed scheme that combines the internal combustion engine (ICE) and traction electric motor (TEM), is the basis for a parallel/series type hybrid propulsion system [1]. In this case, the transforming mechanism from the ICE is generally a gearbox and the TEM is connected with a dual mode reduction unit. In the simplest case, a single-mode reduction unit can be used to connect the TEM.

It is proposed to use the principle of parallel-serial hybrid to increase tracked vehicle survivability in the military operations or other extreme conditions. It is achieved by the engine function duplication. The main advantages of the system are the following: the ICE power reduction with
maintaining the total maximum power on the excessive rate; the reduction of the heat emission and noise level of the machine when using only the TEM [1]. The sequential hybrid principle [2] does not have these advantages.

![Diagram](image)

**Figure 1.** The transmission composition: 1 and 3 – flows from engines; 2 и 4 – reduction units; 5 – the summing reduction unit; 6 – on-board gearbox (hub drive); 7 – flow to the wheels; $T_{0R}$ and $T_{0L}$ – left and right tread brakes.

Suggested in the article scheme solutions [1] were aimed at modernization of the double-flow car transmission with central or on-board gearboxes. The solutions discussed in this article are useful to create a new transmission.

Let's consider the principle of the power flows summing from the two engines with the planetary gearbox. One of the power flows is intended to the steering system control of the tracked vehicle. Such systems are called "parallel". The principle of the closed-circuit steering system should increase the steering control [4,5]. The hydrostatic drive [3], friction steering mechanisms [5,6], electric engine [1] and etc. [7,8] are considered as a part of parallel type hybrid propulsion system. The quality of providing a turning radius smooth variation and correspondence between the steering control position and the turning radius value are of high account in considered conceptions. But the specific advantage of using the traction electric motor is the possibility of changing steering system parameters.

2. Results and Discussion

A simplified kinematic scheme of the drivetrain and the steering mechanism is shown in Figure 2. Two algorithm variants of the movement control could be realized. The engines will perform different functions.

One engine (for example, 3) is used for the vehicle linear motion with a part power unit load. The control elements of planetary gearbox row, which is connected with the selected engine, must be turned on. The second engine could be disconnected from the drive wheels.

However, if the second engine is a reversible electric machine (REM), it can be connected through the control elements. In this case the second engine works in the alternator mode and provides a battery charging. In such case the traction alternator is unnecessary.

Difficulties may arise at the forced mode activation. The REM must be disconnected from the transmission, stopped and then started in the engine mode during connection to the transmission. The REM shaft requires a special break and extra time for that. This is unacceptable for a military vehicle, but it isn't crucial for a transport vehicle.

Three strategies could be used in a turning.

In the first case the engine 1 is connected to the high speed drive tread in the traction mode, achieves the operation mode and increases the torque to the forward drive tread. The engine 3 could be disconnected from the forward drive tread at this time, so the load to the engine will be sharply reduced.
In the second case the engine 1 is connected to the slowing tread in the alternator mode. A part of
the energy is recovered unlike a friction element stopping of the slowing tank tread. The load to the
engine 3 will be increased.

Figure 2. The skid steer transmission kinematic scheme: 1 and 2 – power way from the engines; 2 and
4 – reduction units; 6 – planetary gear reducer; 7 – power way to the drive wheels; $T_{0L}$ and $T_{0L}$ –
stopping brakes of left and right side; $T_{1R}$, $T_{1L}$, $T_{2R}$, $T_{2L}$ – planetary mechanisms control brakes.

In the third case both engines are working in parallel in the linear motion. Both engines are used in
the traction mode, when turning one of them is working in the forced mode. This option is justified if
the engines are electric.

The vehicle turning radius couldn’t be less than half track center distance in any case.

The introduction of the angular velocity symbol is necessary for the kinematics mechanism
description:

- $\omega_1$ and $\omega_2$ – the final drive carrier angular velocity of the high/low speed drive tread;
- $\omega_{01}$, $\omega_{02}$ – the sun gear angular velocity, connected with engine 1 and engine 2;
- $\omega_{T11}$ and $\omega_{T21}$ – the epicyclic gear angular velocity of the low speed drive tread, connected with
  engine 3 and engine 1;
- $\omega_{T12}$ and $\omega_{T22}$ – the epicyclic gear angular velocity of the high speed drive tread, connected with
  engine 3 and engine 1;
- $k_1$ and $k_2$ – kinematic parameters of the three-link planetary mechanism, connected with engine 3
  and engine 1 by the sun gears.

The drivetrain and the steering mechanism kinematics are described by the equations:

$$
\omega_{01} = k_1 \omega_{T11} + (1 - k_1) \omega_1; \quad \omega_{02} = k_2 \omega_{T21} + (1 - k_2) \omega_1; \\
\omega_{01} = k_1 \omega_{T11} + (1 - k_1) \omega_2; \quad \omega_{02} = k_2 \omega_{T22} + (1 - k_2) \omega_2.
$$

The angular velocity of any sun gear is assumed as the constant $\omega_{01} = 1$; in the general case,
another sun gear angular velocity is an independent parameter; it could be expressed through the $\omega_{01}$
fraction.

In addition, the links motion laws, connected to the control elements, should be prescribed.

It follows from the equations shown above, that when the engines work together the equality
should be obeyed (in the linear motion):

$$
\omega_{01}/(1-k_1) = \omega_{02}/(1-k_2).
$$

Two theoretical turning radiuses could be realized by using this kinematic scheme.

The turning around the stopping tank tread is realized by full stop of the low speed tread brakes:

$\omega_{01} = 0; \quad \rho_{11} = 0.5$

The value of this radius is the lowest possible.

The second theoretical turning radiuses are calculated by torque summing from the both engines of
high speed tread 2: $\omega_{T11} = 0$ and $\omega_{T22} = 0$.

The equations:

$$
\omega_{01} = (1 - k_1) \omega_1; \quad \omega_{02} = (1 - k_2) \omega_2.
$$
Consequently:
differential ratio between the tank treads, is respectively [10]:
\[ u_{21} = \omega_2 / \omega_1 = (\omega_{02} / \omega_{01}) \cdot [(1-k_1)/(1-k_2)] > 1. \]

In a special case, \( k_1=\frac{k_2}{2} \):
\[ u_{21} = \omega_2 / \omega_1 = (\omega_{02} / \omega_{01}) > 1. \]

The theoretical radius is expressed through the differential ratio between the tank treads [10]:
\[ \rho_{\phi_2} = 0,5 \cdot (u_{21} + 1)/(u_{21} - 1) = 0,5 \cdot [\omega_{02} (1-k_1) + \omega_{01} (1-k_2)]/[\omega_{02} (1-k_1) - \omega_{01} (1-k_2)]. \]

The value of \( \rho_{\phi_2} \) is determined by \( \omega_{02}/\omega_{01} \) and the kinematic parameters \( k_1/k_2 \).

The equations:
\[ \omega_{01} = (1-k_1)\omega_1; \quad \omega_{02} = (1-k_2)\omega_2. \]

Consequently:
differential ratio between the tank treads, is respectively [10]:
\[ u_{21} = \omega_2 / \omega_1 = (\omega_{02} / \omega_{01}) \cdot [(1-k_1)/(1-k_2)] > 1. \]

If \( k_1=\frac{k_2}{2} \): \( u_{21} = \omega_2 / \omega_1 = (\omega_{02} / \omega_{01}) > 1. \)

The free turning radius is determined by the resistance to the vehicle movement when the engine, connected to the low speed tread, is stopped: off on the lagging side \( \omega_{01} = 0 \), \( \rho_{\phi_2} = \rho_{\omega_1}. \) The value of the free turning radius and uncontrolled turning radius is the same, that is typical of mechanisms with \( q_m = 0.5 \) [10]. It takes place when stopping brakes of the low speed treads are used during the turn. However, the low speed treads retain the speed of linear motion with increasing the ratio \( \omega_{02}/\omega_{01} \), because the value of the kinematic parameter \( q_m = (0.5) \) of the drivetrain and the steering mechanism is atypical. However, it refers to the on-board type mechanisms according to the classification: a single tread speed changing doesn’t cause a consistent speed change of another tank tread.

The linear motion when driving a vehicle on a real ground will be unstable, if \( \omega_{02} = \omega_{01} (\rho_{\phi_2} = \infty) \). The cause is a track rolling resistance changing.

For \( \omega_{02}/\omega_{01} \gg 1 \) \( \rho_{\phi_2} \rightarrow 0.5. \) However, the function \( \rho_{\phi_2} \) is conditionally limited to 1, because this phenomenon is weakly expressed after reaching \( \omega_{02}/\omega_{01} = 3. \)

Thus, the considered function is continuous in the interval \( \rho_{\phi_2} \in [1, \infty) \). The theoretical turning radius smooth changing of such range for a tracked vehicle offered practical advantages.

The urgent task is to ensure the lock down mode of the ICE and REM for the hybrid propulsion system. The REM can be used as the starter or the alternator. The clutch C12 installation (Fig. 3) provides such blocking. However, this solution is not satisfactory, because the inertial moment of rotating parts will be too large.

The drive wheels torque determining for each connection option is not complicated. The on-board gearbox carrier moment is the summation of the planetary row carrier moments.

The differential 5 is installed to the power way, connected to one of the engines, by analogy with considered in the article [1] schemes. The scheme works in the same way as shown in fig. 2 when the clutch C5 is turned on. The differential gear ratio is \(-1\) when the T5 brake is applied. In this case, the planetary mechanisms sun gears of this power way rotate with angular velocities, equal in value but opposite in sign. One tread slows down and becomes a low speed tread, the other one, conversely, becomes a high speed tread. It corresponds with the differential steering mechanism concept for a scheme \( q_m = 0. \)
Figure 3. Transmission kinematic scheme for the modes of the engines blocking and the turning around the center of mass: 1 and 3 – power way from the engines; 2 and 4 – reduction units; 5 – differential; 6 – planetary gear reducer; 7 – power way to the drive wheels; $T_{0R}$ and $T_{0L}$ – the stopping brakes of left and right side; $T_{1R}$, $T_{1L}$ and $T_{2R}$, $T_{2L}$ – the planetary mechanisms control brakes; $C_5$ and $T_5$ – differential control elements; $C_{12}$ – disc lock-up clutch.

It is not necessary to accomplish extra switching for the turning in the power way providing linear motion with a partial power unit load.

It is possible to implement modes similar to the forward drive, because both power ways are reversed in the back drive. The power way reverse is carried out by the reduction 4 reverse gear when using the ICE in the main power way. The reverse REM should be used in the parallel power way.

The torques: $M_1$ and $M_2$ – low/high speed tread carriers torques; $M_{01}$ and $M_{02}$ – the sun gears torques connected to the engines 3 and 1; $M_{T11}$ and $M_{T21}$ – epicycle torques connected with engine 3; $M_{T12}$ and $M_{T22}$ – epicycle torques connected with engine 1.

The torques of the brakes connected with the epicycles are equal to the epicycles torques but they are opposite in sign.

It is understood that reduction gearbox 4 and 2 are installed in the power ways. For example, the ICE is connected to the central gearbox and the REM is connected to the dual-mode reduction gearbox. The equations in any operation mode are the following:

$$M_1 = [D_{11}(1-k)M_{01} + D_{12}(1-k)M_{02}];$$
$$M_2 = [D_{21}(1-k)M_{01} + D_{22}(1-k)M_{02}];$$
$$M_{T11} = -k_1M_{01};$$
$$M_{T12} = -k_1M_{02};$$
$$M_{T21} = -k_2M_{01};$$
$$M_{T22} = -k_2M_{02}.$$

According to the equations, if the control element is on, then the coefficient $D_i$ is equal to 1, otherwise 0. In particular, $M_1 = M_2$ if motion is linear. $D_{11}D_{21}=0$ or $D_{12}D_{22}=0$ if the power unit is working with partial load. $D_{11}D_{22}D_{12}D_{21}=1$ if the power unit is working in the forced mode. Values of $D_{ij}$ are set in accordance with the Table 1

A quasi-continuous variable transmission principle described in the article [11] could be realized when using the considered kinematic scheme as a part of hybrid propulsion system. In this system the torque fluctuations are compensated by the REM in parallel power way without switching to the related gear (for automatic gearbox).

The on-board reduction gearbox could be cut out from the scheme. The on-board reduction gearbox kinematic parameter should be chosen for the planetary rows of the drivetrain and the steering mechanism. The transmission scheme becomes simpler and the vehicle central gearbox is used in the ICE power way. The transmission scheme shown in figure 3 allows waiving the clutch $C_{12}$ and control elements $T_{11}$ and $T_{2R}$. The kinematic scheme should be revised for declining the stopping brakes torque.

Estimation of the engine power in the parallel way can be carried out if it is necessary to complete the turning around the center of gravity. As far as such operation mode is not typical of the high-speed traction vehicles in use, it is possible to apply the short-time overloading to get this kind of mode and to set less power for REM. The related turning radius in the turning around the center of gravity is $\rho=R/B = 0$ ($R$ – turning radius, $B$ – track width): $N_{e2} = M \omega l / \eta_{e1}$. 


Where \( M = \mu GL / 4 \) – turning resistance moment \((G – \) vehicle weight, \( L – \) the supporting surface length); \( \omega = (V_2 - V_1)/B \) – the turning angular velocity \((V_2, V_1 – \) the line speed of the high/low speed treads); \( \eta_{GT} - \) gear train efficiency.

The turning resistance coefficient is explained by Nikitin’s empirical equation \([3,10]\): \( \mu = \mu_{max} / (0.925 + 0.15\rho) \).

However, the coefficient of the maximum turning resistance \( \mu_{max} \) in the case of turning around the stopping track is defined experimentally \([3,10,12]\). In such conditions, \( \mu_{max} = \mu(0.5) \). If \( \rho < 0.5 \), it is necessary to extrapolate function \( \mu(\rho) \) or to correct the equation for \( \mu \) defining (for example, \([10]\)).

The key tank chassis (as a T-80) power should be next to 400 kWt for evenly turning around the gravity on a dry turfy argil sand ground (horizontal surface) with \( \omega = 1.0 \text{ rad/s} \).

This power will be enough to complete a turning with \( \rho_c > 2 \) according to the REM required power deterministic evaluation with a fractional skidding (dirt road running with \( \mu_{max} = 0.8 \)). With less turning radiuses the vehicle angular velocity is limited by cornering resistance. The threat of the skid is of minor importance in this case.

Accordingly, the low-powered ICE (according to a serial vehicle) could be used in the main power way, \( N_1 = (1.5...2.0)N_t \).

**Table 1.** The transmission operation modes

| № | Mode          | Active control elements | Active engines | Notes                        |
|---|---------------|-------------------------|----------------|------------------------------|
| 1 | Linear motion | \( C_5^* \) \( T_{1L} \) \( T_{1R} \) \( T_{1L} \) \( T_{0R} \) | 3              | Part power unit load         |
| 2 | Turning       | \( C_5^* \) \( T_{2L} \) \( T_{2R} \) \( T_{1L} \) \( T_{0L} \) | 1              | Part power unit load         |
| 3 | Forced mode   | \( C_5^* \) \( T_{1R} \) \( T_{2R} \) \( T_{1R} \) \( T_{2R} \) | 3 and 1        | Left high-speed tread        |
| 4 | Turning       | \( C_5^* \) \( T_{2L} \) \( T_{2R} \) \( T_{0L} \) \( T_{0L} \) | 1              | Left low-speed tread         |
| 5 | Left high-speed tread \( C_5^* \) \( T_{1R} \) \( T_{0L} \) \( T_{1R} \) \( T_{0L} \) | 3 and 1        | Left high-speed tread        |
| 6 | Right high-speed tread \( C_5^* \) \( T_{2R} \) \( T_{0R} \) \( T_{2R} \) \( T_{0R} \) | 1              | Right high-speed tread       |
| 7 | Engine locking| \( C_5^* \) \( C_{12}^{**} \) \( T_{1L} \) \( T_{1L} \) \( T_{1L} \) | 3 and 1        | Right high-speed tread       |
| 8 | Constrained rotation of the engine shaft **| \( C_5^* \) \( C_{12}^{**} \) \( T_{1R} \) \( T_{1R} \) \( T_{1R} \) | 3 and 1        | Constrained rotation of the engine shaft **|
| 9 | \( C_5^* \) \( T_{2L} \) \( T_{2R} \) \( T_{0R} \) \( T_{0R} \) | 1              | Right high-speed tread       |
| 10| \( C_5^* \) \( C_{12}^{**} \) \( T_{1L} \) \( T_{1L} \) \( T_{1L} \) | 3 and 1        | Right high-speed tread       |
| 11| \( C_5^* \) \( C_{12}^{**} \) \( T_{1L} \) \( T_{0R} \) \( T_{0R} \) | 3 and 1        | Right high-speed tread       |
| 12| \( C_5^* \) \( C_{12}^{**} \) \( T_{0L} \) \( T_{0L} \) \( T_{0L} \) | 1              | Right high-speed tread       |
| 13| \( C_5^* \) \( C_{12}^{**} \) \( T_{1R} \) \( T_{0R} \) \( T_{0R} \) | 3 and 1        | Right high-speed tread       |
| 14| \( C_5^* \) \( C_{12}^{**} \) \( T_{1L} \) \( T_{1L} \) \( T_{1L} \) | 1              | Right high-speed tread       |

Notes:
* the control elements are used when the differential unit 5 is a part of the transmission;
** – the control element is used for the REM working in the alternator operation mode or for the ICE emergency start with the REM as an element of the hybrid propulsion system; the operation mode is possible when the friction \( C_{12} \) is a part of the transmission;
*** – the operation mode is possible if the differential unit 5 is used in the transmission; the turning direction is defined by the REM rotation direction.

The power way gear ratios selection determines the engines combined action.
Nowadays, the domestic industry technologies allow creating REM and batteries with sufficient operating life and good reliability for the transport tracked vehicles. It is proved by the article [1].

The variants of the considered transmission could be adapted to increase the steering control quality of: the compact tracked chassis, different transport/ military vehicles, transport tracked vehicles and support service vehicles. It is possible to create the summing reduction unit for the ship with two engines types according to the on-board reduction kinematic scheme.

3. Conclusion

1. The proposed drivetrain and the steering mechanism allows using two engines coherently in the tracked vehicles (as a part of the hybrid propulsion systems as well). In the case of the shut-down engine, the mechanism has the overlap of the steering control.
2. The transmission allows providing high-quality steering control.
3. The technology level of the REM and batteries production allows realizing the hybrid propulsion system with proposed transmission in the high-speed traction vehicles weighing up to 30-35 tons. It is possible to use this technology in the key tank in the nearest future.

References

[1] Demidov N N 2016 The choice of the transmission formation schematic option of the military vehicle with hybrid power plant Design and using of electric transmission for armament models and military machines (Saint-Petersburg: Publishing house of JSC VNIITransmash) 87-100
[2] Gusev M N, Zaitsev V A and Kurtz D V 2010 The concept and the main provisions of efficient choice and justification of the hybrid propulsion system parameters for unified new-generation base chassis Actual problems of protection and safety. Armored vehicles and armament Moscow: Publishing house of FSBI RARAN 28-32
[3] Nosov N A 1972 The traction vehicle design and calculation Leningrad: Mashinostroyeniye 559
[4] Galyshev Yu V 2014 Closed-loop control systems for tracked vehicle steering Scientific-and-technical bulletin of SPbSPU: Science and education 3 201-208
[5] Galyshev Yu V 2014 SPbSPU scientists’ research and development in the area of defence equipment Scientific-and-technical bulletin of SPbSPU: Science and education 1 26-32
[6] Dobretsov R Yu 2014The steering friction mechanism of the double flows transmissions of the tracked vehicles Inventors in the innovation process of Russia: the conference proceedings Saint-Petersburg, Publishing house of SPbPolyTechU 121-124
[7] Filippov A N 2016 Transport tracked vehicles: the steering mechanism with non-linear characteristic Modern mechanical engineering: Science and education, Saint-Petersburg: Publishing house of SPbPolyTechU 898-912
[8] Izotov V Z, Pyatkov V A, Starovoytov V S and Suslov A A The steering mechanism of tracked vehicle, Patent USSR 521174
[9] Lozin A V 2015 The double-flows transmission of the transport tracked vehicles with skid steering, Patent RU 2599855
[10] Shelomov V B 2013 The theory of multi-purpose tracked and wheeled vehicles movement. Traction calculation of curvilinear motion: a manual for universities in the specialty "Automobile and tractor" Saint-Petersburg: Publishing house of SPbPolyTechU 90
[11] Bukashkin A Yu 2017 Split Transmission of Tractor with Automatic Gearbox Procedia Engineering 206 1728-1734
[12] Zabavnikov N A 1975 The basics of the transport traction vehicles theory Leningrad: Mashinostroyeniye 448