Vojnotehnicki glasnik/Military Technical Courier
ISSN: 0042-8469
ISSN: 2217-4753
vojnotehnicki.glasnik@mod.gov.rs
University of Defence
Serbia

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Modelling and performance analysis of the BVP M-80A hybrid drive
Vojnotehnicki glasnik/Military Technical Courier, vol. 69, no. 1, 2021
University of Defence, Serbia
Available in: https://www.redalyc.org/articulo.oa?id=661771133004
DOI: https://doi.org/10.5937/vojtehg69-28232

Original scientific papers

Modelling and performance analysis of the BVP M-80A hybrid drive

Моделирование и анализ работы гибридного привода BVP M-80A

Симулациони модел и анализа перформанси хибридног погона БВП М-80А

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DOI: https://doi.org/10.5937/vojtehg69-28232
Redalyc: https://www.redalyc.org/articulo.oa?id=661771133004

Received: 02 September 2020
Revised document received: 28 October 2020
Accepted: 30 October 2020

Abstract:

Introduction: Hybrid technology has been successfully incorporated into the industry of passenger and commercial vehicles. Driven by the success and benefits that hybrid technology brings, many defense organizations around the world invest in the development of hybrid technology for combat vehicles and develop prototypes of tracked combat vehicles which have lower fuel consumption, better performance, better exhaust emission, and additional onboard electric power. However, various technical challenges must be resolved before it comes to the introduction of hybrid tracked combat vehicles in operational use. Several successful tests of prototypes have been conducted so far, but there are still restrictions on key technologies such as electric motors, electronics, and storage of electricity. In such conditions, where finance is limited, mistakes cannot be allowed nor spending a lot of resources on planning, building prototypes, and testing.
Method: Therefore, it is clever to run the simulation software with which it is possible to examine various parameters in simulated conditions which more or less mimic real operating conditions. This paper aims to show one of possible solutions concerning the selection of appropriate technologies of hybrid drive, to propose a system solution for a hybrid BVP M80A, and to display a simulation hybrid drive model and the results obtained from the model devised in Simulink.
Results: The results obtained by the simulation show that the proposed hybrid drive solution provides better performance while retaining key drivetrain elements of the vehicle.
Conclusion: Only turning parameters are considered during the simulation but it is clear that the hybrid drivertrain has advantages related to straight-line motion as well. Also, sound projections about the drivetrain performance and control can be made with the use of the proposed model.

Keywords: hybrid drive, combat vehicle, tracked vehicle performance, hybridization, MATLAB, Simulink.

Р е з ю м е :

Введение/цель: Гибридная технология успешно применяется в автопромышленности легковых и коммерческих автомобилей. Руководствуясь успехом и преимуществами гибридных технологий, многие оборонные организации по всему миру инвестируют в разработку гибридных технологий для боевых машин и разрабатывают прототипы гусеничных бронемашин, которые расходуют намного меньше топлива, отличаются лучшей производительностью, меньшим выбросом выхлопных газов и дополнительными возможностями. Однако перед вводом в боевую эксплуатацию гибридных гусеничных бронемашин необходимо решить различные технические проблемы. На данный момент был проведен целый ряд испытаний прототипов, но все еще существуют ограничения, относящиеся к ключевым технологиям, таким как:

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электродвигатели, проводниковая электроника и накопители электроэнергии. В условиях финансовых ограничений нельзя допускать ошибок и тратить много ресурсов на планирование, создание прототипов и испытания.
Методы: Следовтельно, разумным решением было запустить программное моделирование, с помощью которого можно исследовать различные параметры в имитационных условиях, которые более или менее имитируют реальные условия эксплуатации. В данной статье было представлено одно из возможных решений, касающихся выбора соответствующих технологий гибридного привода, предложено системное решение для гибридного привода бронемашины М-80А. В целях осуществления данного плана, была разработана имитационную модель гибридного привода с помощью программы MatlabSimulink.
Результаты: Результаты, полученные при моделировании, показывают, что предлагаемое решение гибридного привода обеспечивает лучшую производительность транспортного средства.
Выводы: При моделировании проверялись только параметры поворота, но эта модель может быть успешно примененав испытанияхпараметров при прямолинейном движении. Кроме того предложенную модельможно применять для испытания параметров и алгоритмов котроля трансмиссии.
Ключевые слова: гибридный привод, боевая машина, параметры гусеничных машин, гибридизация, MATLAB, Simulink.

Abstract:

Увод: Хибридна технологија је успешно инкорпорирана у индустрији путничких и комерцијалних возила. Вођени успехом и предностима које она доноси, многе одбрамбене организације широм света улажу у развој хибридне технологије за борбена возила. Развијају и прототипове гусеничних борбених возила која имају мању потрошњу горива, боље перформансе, бољу издувну емисију и више електричне енергије која се може искористити за различите елементе надградње. Међутим, постоје различити технички изазови који морају бити решени пре увођења хибридног гусеничног борбеног возила у оперативну употребу. Спроведено је неколико успешних испитивања прототипова, али још увек постоје ограничења у вези са кључним технологијама као што су електромотори, складиште електричне енергије и проводничка електроника. У таквим условима, у којима су финансије ограничене, не може бити много грешака ни трошења пуно ресурса на планирање, изградњу прототипова и њихово тестирање.
Метода: Рационално је развити софтверску симулацију помоћу које је могуће испитати различите параметре у симулираним условима који, мање или више, опонашају реалне услове експлоатације. У раду је приказано једно од могућих решења избора одговарајућих технологија хибридног погона и предложено системско решење хибридног погона за борбено возило пешадије М-80А. За усвојено решење развијен је симулациони модел хибридног погона у програмском окружењу MatlabSimulink.
Резултати: Резултати добијени симулацијом показују да предложено решење хибридног погона обезбеђује знатно боље перформансе погонске групе возила.
Закључак: Разматрани су само параметри заокрета, али модел се може успешно применити и при анализи перформанси праволинијског кретања. Поред тога, модел је могуће користити и за испитивање перформанси и алгоритама контроле трансмисије.
Keywords: хибридни погон, борбена возила, перформансе гусеничних возила, хибридизација, MATLAB, Simulink.

Introduction

In the case of wheeled vehicles, in general, the drive (propulsion) and steering functions are separated and the change of direction does not significantly affect the traction and braking performance of the vehicle. In contrast to this, in the case of tracked vehicles, the steering control is performed by achieving a difference in the speed of rewinding of the tracks, i.e. lateral drift. This way of steering significantly affects the traction performance of the vehicle because, as a rule, the resistance when making turns increases compared to the straight-line movement. Also, the control function built into the power transmission system affects the load distribution and power balance within the power transmission system itself (Muždeka et al, 2004). The turning performance and the effects of various elements of the transmission design on the performance have been the subject of numerous research papers. However, vehicle hybridization, which has recently experienced rapid development in the military industry, raises many unanswered questions (Khalil, 2009). Numerous states and defense organizations, guided by the advantages of hybrid vehicles such as improvement of mobility and fuel economy, higher specific power, silent mobility and silent watch capabilities, enhancement of onboard electric power generation, etc. invest significant resources in the development and testing of prototype hybrid vehicles (Bhatia, 2015), (Kramer & Parker, 2011), (Dalsjø, 2008). Some papers deal with the design of such vehicles (Johnson & Dueck, 2001), (Nederhoed & Walker, 2009) but, to the best of the authors'knowledge, there are no papers that deal with the performance of such a vehicle except one paper (Taira et al, 2018). The paper presents the conceptual hybrid drive solution for the infantry fighting vehicle BVP M-80A, analyzes the power flow, defines the model of turning resistance acting on the vehicle implemented in the Matlab Simulink software environment, and develops a model for simulating the vehicle drivetrain. The results of the comparative analysis of the symmetrical and asymmetrical turning mechanisms are presented because hybrid drive enables a selection of the kinematic parameter. The introduction is an introductory part of the article.

Influence of hybrid drive on the turning system

In addition to the advantages hybrid drive offers, which relate to the specific power of the vehicle, additional onboard electric power to meet power requirements of the vehicle subsystems and armament, silent mobility, etc., the question arises as to how vehicle hybridization affects the turning system and the turning performance. One of the assumptions is that, due to the existence of a braking force on the inner track, electricity will be generated during the turn because the electric motor will act as a driven machine - generator. Also, if it is made with two electric motors, hybrid drive allows not only an infinite number of calculated turning radii but also the possibility of choosing the kinematic turning parameter.

There are several possible conceptual solutions for the hybridization of the BVP M80A drive, and in this paper a solution was chosen that should not imply radical design changes to the conventional transmission. Therefore, it is a parallel hybrid architecture, with two electric motors installed in the auxiliary drive. The kinematic schemes of both conventional and hybrid transmissionsare given in Figure 1.

Unlike the conventional transmission, described in detail in (Vesić & Muždeka, 2007), which has a single-stage transmission in the auxiliary drive, the hybrid transmission has two independent power sources in the auxiliary drive in the form of two electric motors. The change in the rewinding speed of tracks is achieved by activating one or both electric motors whose angular velocities can be precisely controlled. For this reason, such a transmission offers several possibilities that the conventional transmission cannot fulfill. First of all, electric motors are additional drive units and thus increase the specific power of the whole vehicle, enabling the vehicle to overcome higher resistances or to achieve better acceleration, etc. Since two electric motors with a precise adjustment of their angular velocities are in the auxiliary drive as two independent drive units, it is possible to provide an infinite range of turning radii without power losses due to friction in the transmission elements. Also, since electric motors can easily change the angular velocity direction, such a turning system can be both symmetrical and asymmetrical, depending on the control method. If the electric motor of the outer track is held stiff, and the electric motor of the inner track is acted upon by the electric motor in the sense of reducing the speed of rewinding, it is an asymmetrical turning system. If the electric motors are acting upon the outer and inner track with the same angular velocity intensity, but in the opposite direction, it is a symmetrical turning system.

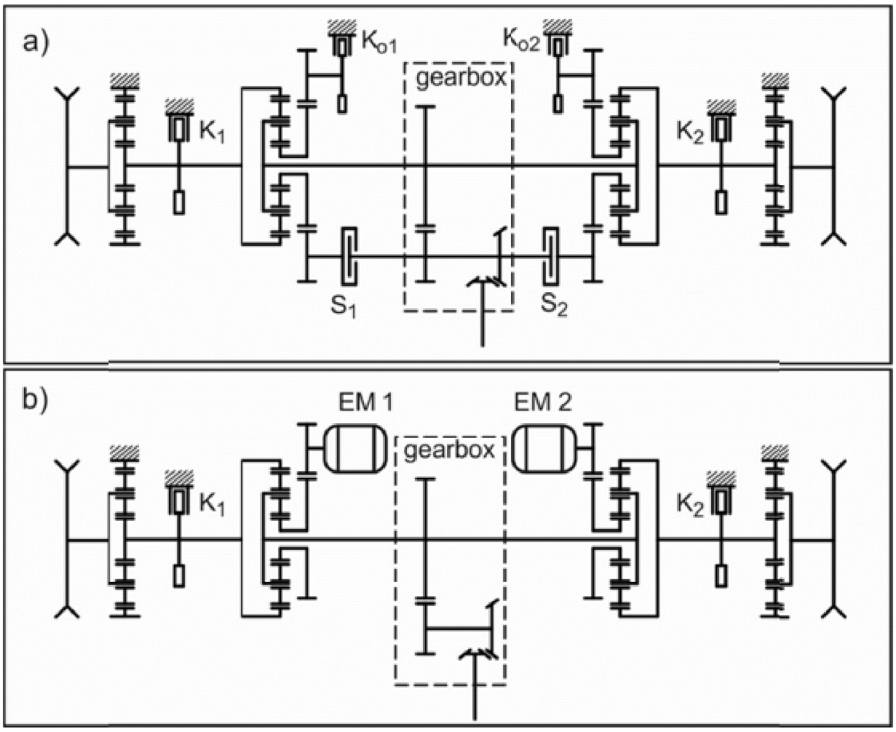

FIGURE 1
Kinematic scheme of the conventional (a) and the hybrid (b) drive system

This fact is very important from the aspect of performance. The possibility of simulating the vehicle turn by using both system types opens up a possibility of comparing the performance of turning and making a conclusion about which control algorithm for electric motors is the best.

Power flow analysis

The electric motors can fine-tune angular velocity, in both directions, independent of the internal combustion engine (ICE) which means that electric motors can have a wide range of angular velocities, from a maximum angular velocity with a negative sign to a maximum angular velocity with a positive sign without any dependence on the ICE. For that reason, when determining power flows, it is necessary to pay attention to the direction of the sun gear angular velocity and to the possibility that the sun gear angular velocity is that much higher than the carrier angular velocity that the ring gear angular velocity changes direction. This case is precisely the flexibility offered by the independent auxiliary drive that gives the option to select (adjust) the kinematic turning parameter.

Since vehicle turns are most often performed with a turning radius smaller than the radius at which the inner track force equals zero ("free" radius Rs), power flow was analyzed only for that case.The power flow when turning with a radius smaller than Rs is shown in Figure 2.

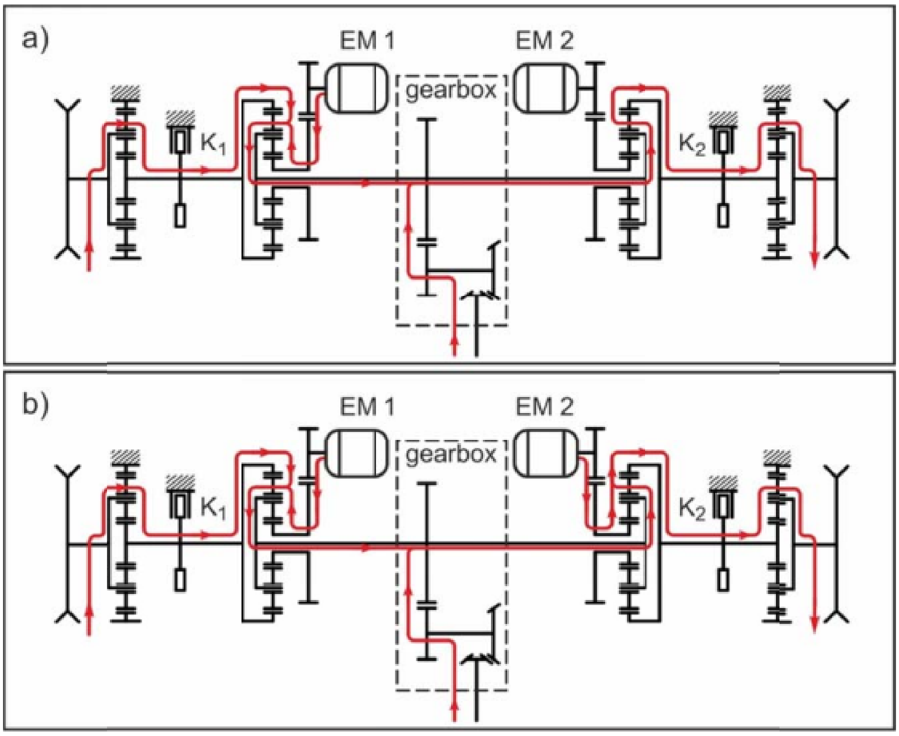

FIGURE 2
Power flow during asymmetrical (a) and symmetrical turning (b) at a turning radius smaller than Rs

In the case of an asymmetrical turn with a radius smaller than Rs, the power flow is identical to the conventional transmission for the radius at which there is no power loss due to friction in the drivetrain (the "calculated" radius Rp). The calculated radius is achieved by activating the clutch S1 (Figure 1a), with the difference that the auxiliary drive supplies the electric motor power and not the ICE power as in the conventional transmission. The recuperation power from the inner track, which enters the transmission via the ring gear, is added to the power of the auxiliary drive and transmitted to the outer track via the driven gearbox shaft. In the case of a turn with a radius greater than Rs, the power of the ICE is divided into two when it reaches the carrier; one part goes to the sun gear while the other part goes to the ring gear, where the electric motor works as a generator. The situation on the planetary gear set of the outer track is the same in both cases as in the case of the straight-line motion when the vehicle is powered only by the ICE.

The hybrid transmission, unlike the conventional one, offers several flexibilities that can be used in a turn: in addition to the classic asymmetric turn with inner track deceleration, it is possible to achieve an asymmetric turn with outer track acceleration, as well as a symmetrical turn. The possibility of an asymmetrical turn with the acceleration of the outer track is not justified due to high power demand, but the possibility of a symmetrical turn certainly enables the improvement of the vehicle's turning flexibility and performance.

With a symmetrical turn, the case with the inner track is the same as with the asymmetrical turn, where the rewinding speed of the inner track decreases, but the case with the outer track is not. For the outer track to accelerate, it is necessary for the sun gear of the planetary gear set to be driven with a velocity of the negative sign. In this case, the drive of the outer track is done by summing the power of the outer track electric motor and the ICE power in the main drive.

What attracts attention in the analysis of power flow is the fact that there is no regenerative braking during a turn with a radius smaller than Rs. With this kinematic configuration of the transmission, regenerative braking is achieved only at turns with a radius greater than Rs, but since the vehicle is in such a mode for a very short time, regenerative braking would be negligible.

Hybrid drivetrain model

In order to analyze the performance of different variants of hybridized transmission operation, a Simulink model of the M80A infantry fighting vehicle drive compartment with the hybridized transmission was developed. The basic structure of the drive group model is shown in Figure 3. Before the formation of the drive group simulation model, the basic components of the electric drive were selected, which is described in more detail in (Milićević, 2019).

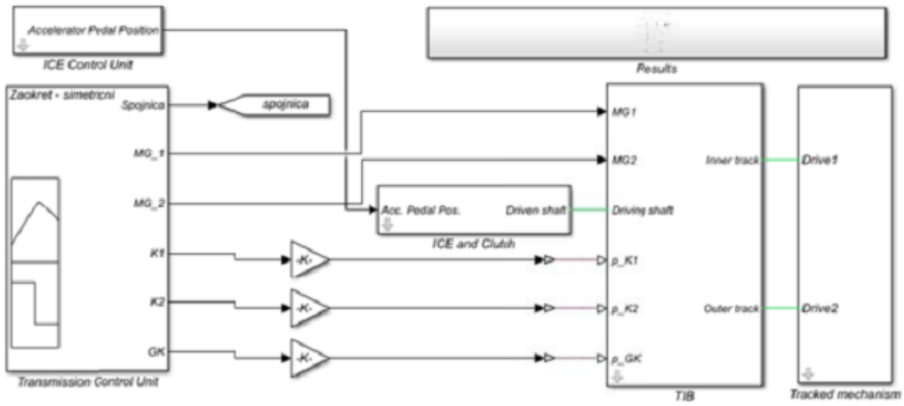

FIGURE 3
BVP M-80A hybrid drive topology in Simulink

Drivetrain model control is achieved with the "ICE Control Unit" and "Transmission Control Unit" blocks. The key element of the "ICE Control Unit" block is the standard Simulink *Longitudinal Driver block* based on literature (MacAdam, 1980). The main function of this block is to define the required speed of the vehicle, which is achieved by adjusting the accelerator pedal position so that the preset speed is reached as soon as possible. The transmission control unit is modeled in the form of control signals which adjust the transmission parameters according to a predefined scenario. As the subject of the paper is not the automation of electric motor control, it was adopted to control them by speed signals that are set according to the predefined control scenario. In addition to the signals controlling the electric motors, the block also includes the control signals for the clutch and the main brakes. The clutch control signal allows the simulation of the vehicle when it starts to accelerate from the point of zero velocity, while the brake control signals in this particular simulation are not used, since brakes are not activated in the considered turning mode.

The "ICE and Clutch" block consists of a standard Simulink *Generic Engine* block and a hydraulically activated friction clutch model (Krsmanović, 2008), (Grkić et al, 2009). The ICE parameters are set in accordance with the power curve of the BVP M80A engine (10V003), with the initial angular speed of 900 $min^{-1}$and the regulation of the maximum speed at 2500 min^{-1}. The friction clutch model contains a standard Simulink block *Fundamental Friction Clutch* with parameters selected so that it is possible to simulate the acceleration of the vehicle from the point of zero velocity, without in-detail dealing with the process of engaging the clutch.

The *TIB* block (Figure 4) includes the mechanical transmission elements and the auxiliary electric drive. The gearbox is modeled as a gear pair because the simulation involves movement in one gear, and the parameters of the gears (gear ratios) correspond to the parameters of the real transmission. The only difference between the mechanical model shown in Figure 4 and the real model is the KM brake which simulates the braking of the ICE drive shaft in the case of starting the vehicle from the place only by employing an electric motor. Also, the main brakes K1 and K2 were modeled within the transmission model.

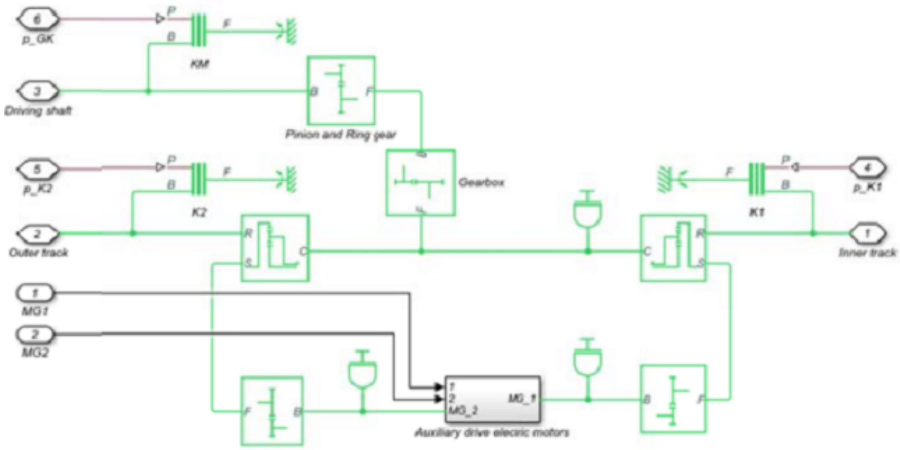

FIGURE 4
TIB (Transmission in Block) model in Simulink

The "Auxiliary drive electric motors" block includes a power supply unit (battery) and electric motors. For the battery model whose scheme is shown in Figure 5, an existing generic lithium-ion battery model was adopted (Balch et al, 2001), (Gao & Ehsani, 2012), represented by the equations (Tremblay & Dessaint, 2009):

- discharge model (i* > 0)

$$f_1(i_t, i^*, i) = Eo - K \cdot \frac{Q}{Q - i_t} \cdot i^* - K \cdot \frac{Q}{Q - i_t} \cdot i_t + A \cdot e^{(-B \cdot i_t)} \tag{1}$$

- charge model (i* < 0)

$$f_2(i_t, i^*, i) = Eo - K \cdot \frac{Q}{i_t + 0.1 \cdot Q} \cdot i^* - K \cdot \frac{Q}{Q - i_t} \cdot i_t + A \cdot e^{(-B \cdot i_t)} \tag{2}$$

And the battery charge status (State of Charge - SOC%) is calculated from the following equation:

$$SOC = 100\left(1 - \frac{1}{Q}\int_0^t i(t)dt\right)[\%] \tag{3}$$

where: E_0 - constant voltage [V]; K - polarization constant [V/Ah], or polarization resistance [Ω]; i^* - low-frequency current dynamics [A]; i - battery current [A]; i_t - extracted capacity [Ah]; Q - maximum battery capacity [Ah]; A - exponential voltage [V]; and B - exponential capacity [Ah^{-1}].

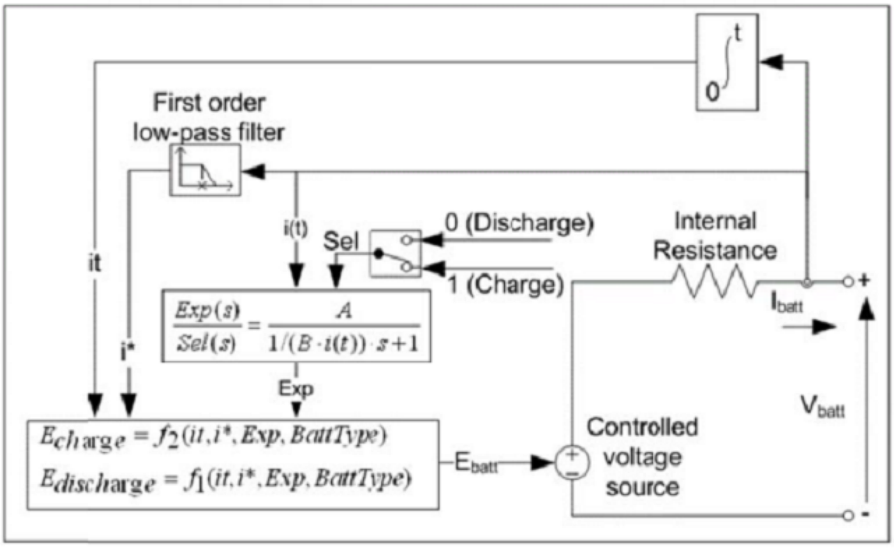

FIGURE 5
Equivalent battery circuit presented by equations (1), (2), and (3)

The electric motor model is a *DTC Induction motor drive* model from the Simulink library modified so that its input is direct current. In Figure 6, a block diagram of a drive with the DTC control is displayed (Rosić, 2016).

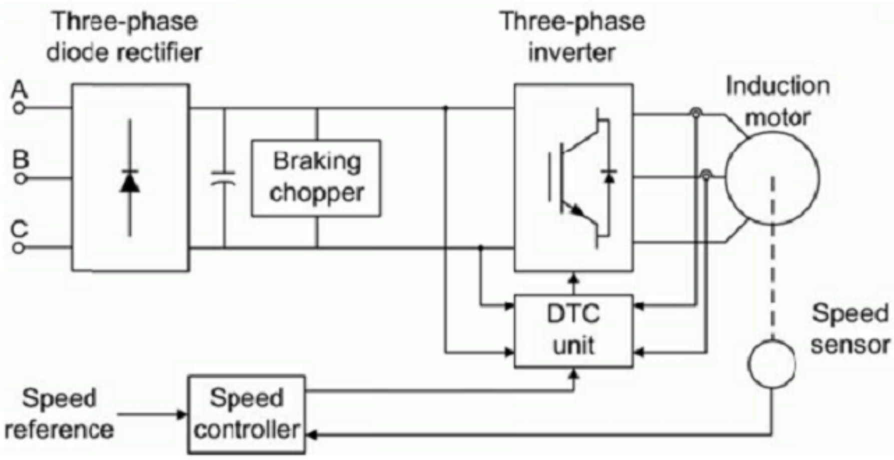

FIGURE 6
Block diagram of the conventional DTC drive

The "Tracks" block (Figure 7) is a subsystem of the Simulink model which represents resistance loads of the drive compartment, i.e. the resistance to the movement of the vehicle when performing a turn. Resistances to movement have a complex nature and depend on the type of soil, slope, distribution of specific pressures on the soil, turning radius, etc. (Muždeka et al, 2004). For this paper, a simplified turning resistance model is considered in the analysis of turning performance with the following assumptions:

- the turn is achieved on a hard horizontal surface, at low speed, so that the influence of the centrifugal force is not considered,
- the center of gravity is in the middle of the contact surface of the track so that the specific pressure of the track on the soil is rectangular, and
- track slip under the influence of the brake and traction forces is not considered.

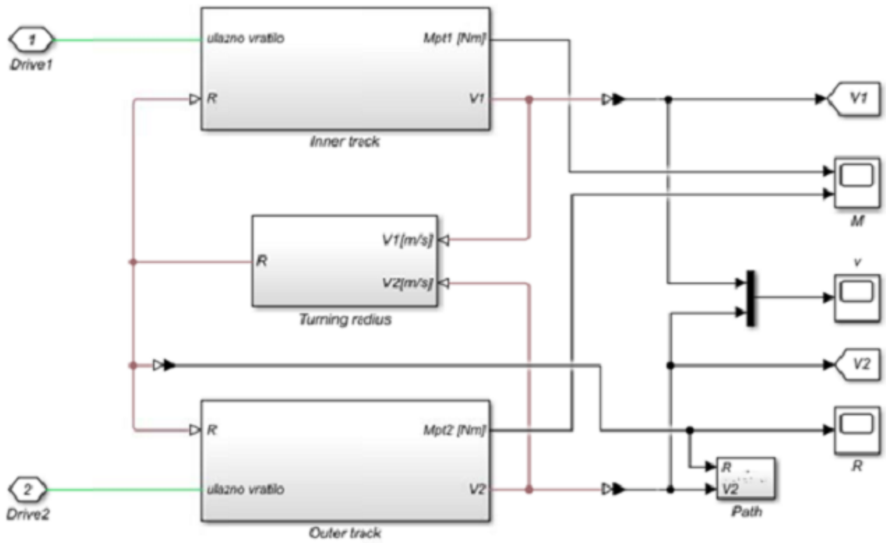

FIGURE 7
Simulink model of tracks

The turning radius of a tracked vehicle is obtained from the following equation:

$$R = \frac{B}{2} \cdot \frac{v_2 + v_1}{v_2 - v_1} \tag{4}$$

where: *V1, V2* – the rewind speed of the inner and outer tracks and B –the track width.

Figure 8 shows the forces acting on the vehicle during turning under the conditions described above. Based on the presented mechanical model, the equations for determining the required forces on the tracks are obtained as follows:

$$F_2 = R_{k2} + \frac{M_c}{B}$$
$$F_1 = R_{k1} - \frac{M_c}{B} \tag{5}$$

where: R_{k1}, R_{k2} – the straight-line motion resistance and M_c – the turning resistance.

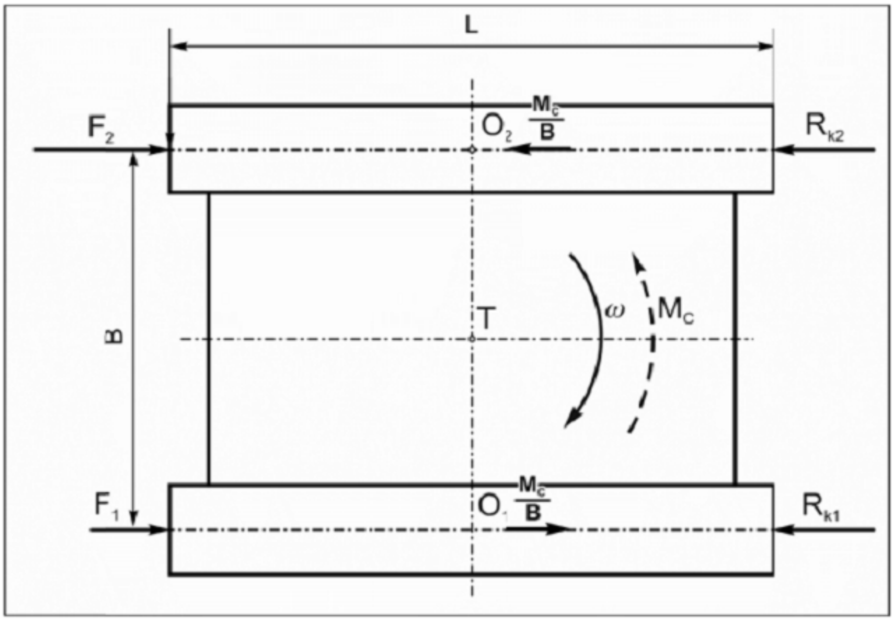

FIGURE 8
Forces and moments acting on the tracked vehicle during a steady state turn

In the case of the uniform turn of the tracked vehicle at low speeds on a horizontal surface, the straight-line motion resistances, which are a consequence of the surface deformation, are calculated from the expressions:

$$R_{k1} = fN_1$$
$$R_{k2} = fN_2 \tag{6}$$

where: f – the rolling resistance coefficient (adopted value $f = 0{,}07$) and N_1, N_2 - the surface reaction forces.
For the considered case of turning, the normal surface reactions are equal to half the weight of the vehicle:

$$R_{k1} = R_{k2} = f\frac{G}{2} \tag{7}$$

The turning resistance is obtained as follows:

$$M_c = \frac{\mu GL}{4} \tag{8}$$

where: μ – the turning resistance coefficient and G, L – the vehicle weight and the contact surface length, respectively.
The turning resistance coefficient (Nikitin & Sergeev, 1962) is calculated using the following expression:

$$\mu = \frac{\mu_{max}}{a + (1-a)\left(\frac{R}{B} + \frac{1}{2}\right)} \tag{9}$$

where: R – the turning radius; μ_{max} – the coefficient value when $R=\frac{B}{2}$ (adopted value μ_{max} = 0,85), and a – the experimental coefficient (a = 0,8 - 0,85, adopted value a = 0,85).

After adopting the values of the coefficients, the final expression for the coefficient of resistance to rotation is obtained:

$$\mu = \frac{0.85}{0.85 + 0.15\left(\frac{R}{B} + \frac{1}{2}\right)} \tag{10}$$

The final inner and outer track forces expressions:

$$\begin{aligned} F_2 &= R_{k2} + \frac{M_c}{B} = \frac{fG}{2} + \frac{\mu GL}{4B} \\ F_1 &= R_{k1} - \frac{M_c}{B} = \frac{fG}{2} - \frac{\mu GL}{4B} \end{aligned} \tag{11}$$

The track forces are radius dependent as shown in Figure 9. It can be seen that the outer track force has a positive sign, i.e. it is a tractive force at all turning radii, while the inner track force is a tractive force at radii larger than Rs, while it is a braking force at smaller ones.

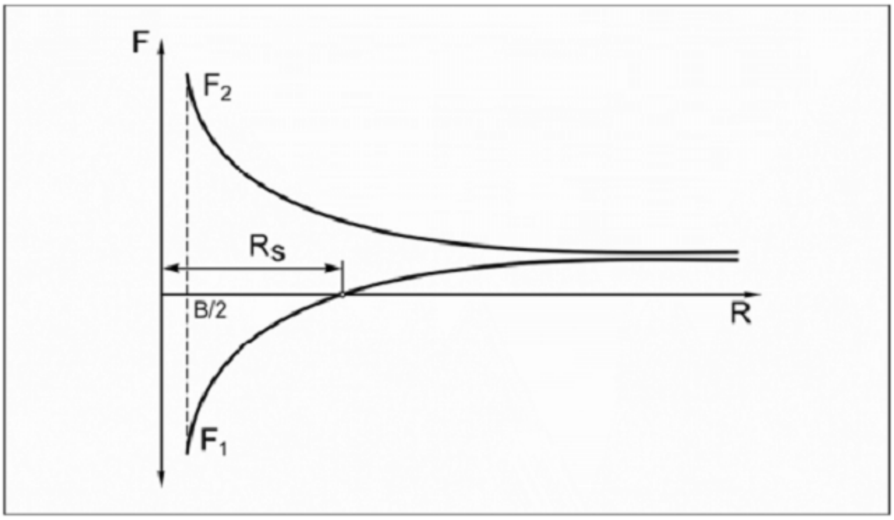

FIGURE 9
Dependence relation of the track forces on the turning radius

In addition to external resistances, internal resistances have an important influence on the tracked vehicles' power balance. For a vehicle performance analysis, losses in transmission and electric motors can be neglected, but losses in the tracked mechanism cannot, so they are taken into account through the efficiency coefficient of the tracked mechanism. The efficiency coefficient of the tracked mechanism in the general case depends on the magnitude of the force on the track, the speed of movement, the type of hinge, and the like. A simple calculation can use a linear dependence from the equation (Muždeka et al, 2004)

$$\eta_{gm} = 0.95 - 0.018 \cdot V \tag{12}$$

where: η_{gm} – the efficiency coefficient of the tracked mechanism and V – the vehicle speed.

To be able to simulate the impact of losses in the tracked mechanism, it is necessary to express the efficiency coefficient in relation to the torque. In the case where the force on the track is tractive, the losses in the tracked mechanism increase the required power, so that the torque at the sprocket wheel that represents the resistance is increased and calculated from the expression:

$$M_{pt} = \frac{F_i \cdot r_{pt}}{\eta_{gm}} \tag{13}$$

In the case where there is a braking force on the track, the resistances of the tracked mechanism cause the torque at the sprocket wheel to be reduced and calculated from the expression:

$$M_{pt} = F_i \cdot r_{pt} \cdot \eta_{gm} \tag{14}$$

On the outer track, the torque on the sprocket wheel is always calculated from expression (13), and on the inner track from expressions (13) and (14), depending on the sign of the force on the track. The block diagram of the load model on the inner track is shown in Figure 10.

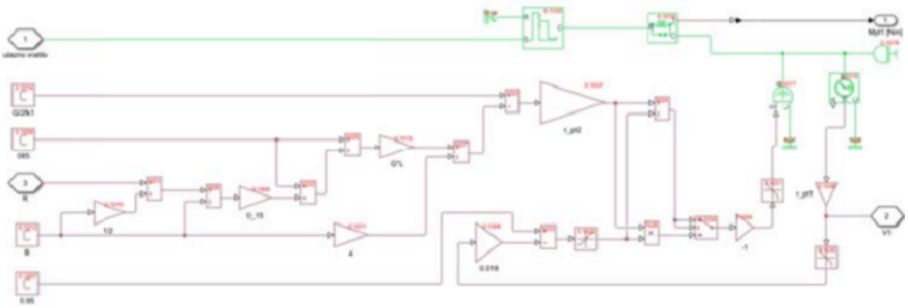

FIGURE 10
Inner track load model

Simulation results

Based on the developed model, the simulation of the vehicle movement with the hybridized transmission during turning was performed. In order to see the hybridization influence on the turning performance, a simulation was performed in three motion modes: turning with the deceleration of the internal track - asymmetric turn (case A), turning with the deceleration of the inner and acceleration of the outer track, with a turning radius as in the previous case - symmetrical turn with a lower initial speed (case B), and turning with the deceleration of the inner and acceleration of the outer track but without reducing the speed of the vehicle - symmetrical turn (case C). The first case is completely equivalent to the behavior of the conventional transmission and it is a reference case for assessing the hybridization impact on the performance of turning. It is important to note that even though the turn is performed equivalent to the conventional system, there are significant advantages over the conventional system. The main advantage is the fact that the rotation is continuous, i.e. it is achieved without power losses due to friction in the transmission elements. The B and C cases of turning are not possible with the conventional transmission. In contrast, the hybrid transmission allows such cases of turning by choosing the way of controlling electric motors in the auxiliary drive, which means that the possibility of their use is realistic. The B case represents the rotation of the vehicle with the same radius as in the A case, and since the rotation is essentially symmetrical, it is achieved with a lower initial speed. The C case is a classical symmetrical turn and is achieved at the same initial speed as in the A case. The diagram, Figure 11, shows the change of the turning radius for all three considered cases, where the previous statements are confirmed.

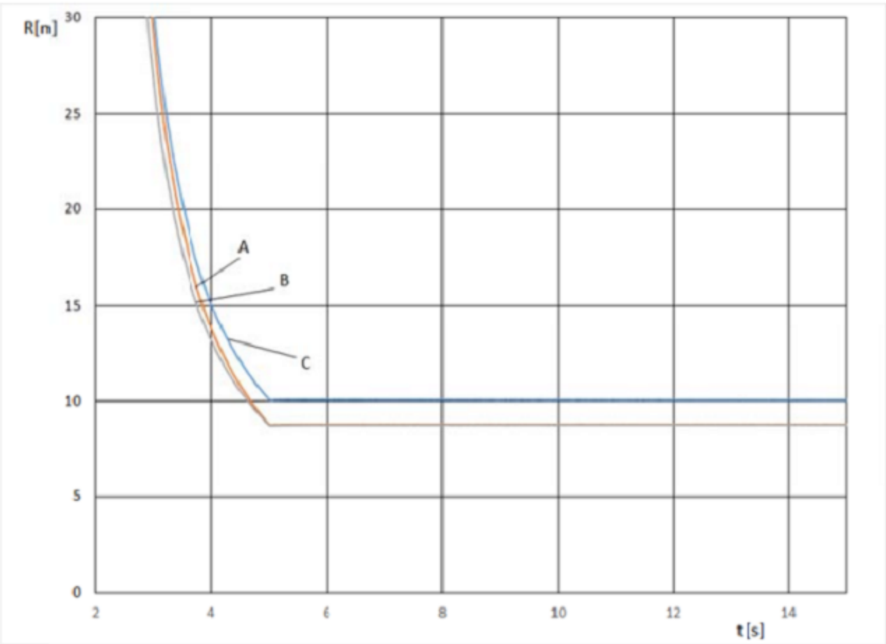

FIGURE 11
Turning radius change in time for the A, B, and C cases

Figure 12 shows the speeds of the vehicle during the simulation of the turning process. The simulation includes starting the vehicle from the point of zero velocity, moving at a constant speed, entering a turn, and moving in a turn with a constant radius. It can be seen that from the aspect of changing the speed of movement, the B and C cases are favorable because there is no change in the speed of the turn.

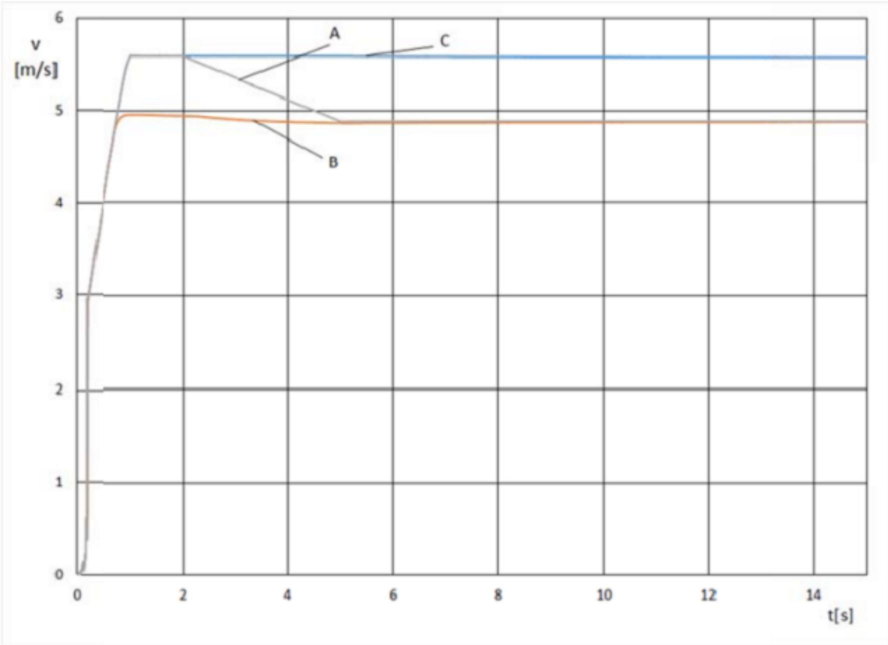

FIGURE 12
Change of the vehicle speed during the turn for the A, B, and C cases

In order to perceive the performance of the turn, it is necessary to analyze the power needed for individual cases of the turn. As the power of the basic drive unit - diesel engine and the power of the electric motor in the auxiliary drive - is engaged in the turn, it is necessary to analyze the value of these powers, as well as their ratio. Figure 13 shows the total power engaged to perform the turn. In the C case, the greatest power is engaged, bearing in mind that the turn takes place at a higher initial speed, but it is important to note that, in this case, the turning radius is larger, which is even more unfavorable. In the cases of A and B, practically the same power is engaged, but two facts must be taken into account: 1) the initial speeds are different and 2) the same turning radius is achieved. As the movement speed directly affects the number of revolutions of the ICE, it is important to analyze the relationship between the power of the drive motor and the power that is supplied to the transmission via the auxiliary drive, i.e. via the electric motor.

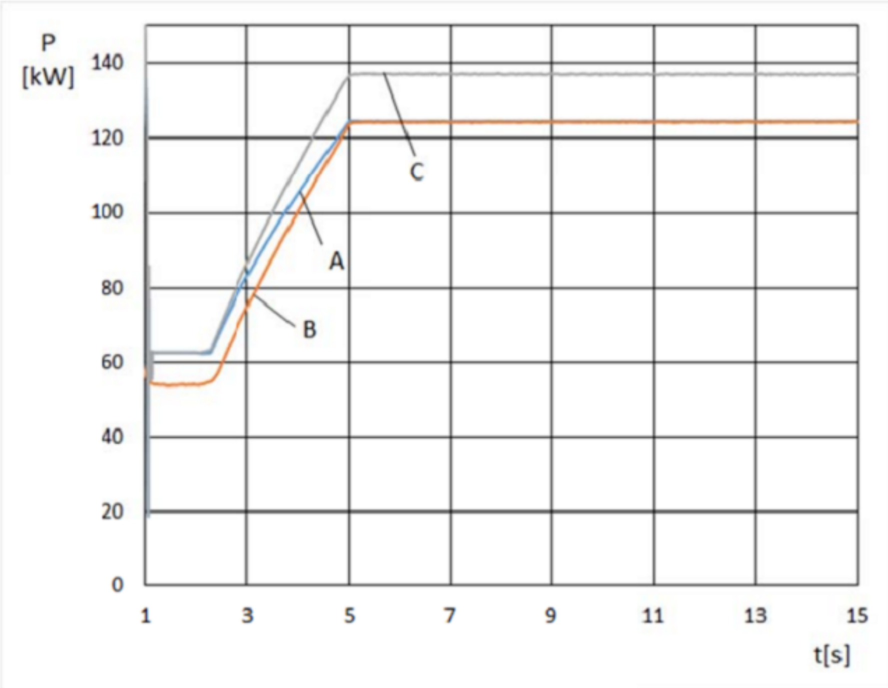

FIGURE 13
Total power needed for turning

Figure 14 shows the ratio of the power engaged by the diesel engine (P_{DM}) and the power of the electric motor (P_{EM}) for the A case and the B case, which are performed with the same turning radius. It can be seen that the power balance is more favorable because less diesel engine power is used when turning, which is a consequence of the fact that in this case two electric motors are working.

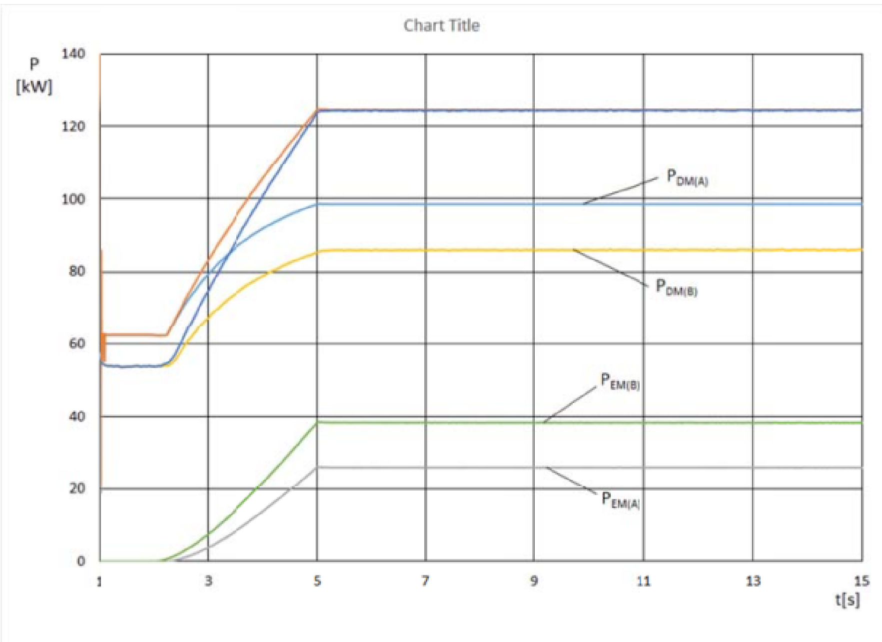

FIGURE 14
Total power needed for turning

Conclusion

Based on the results presented in the paper, it can be concluded that the proposed hybridization model of the BVP M80A powertrain would provide a significant improvement in vehicle performance while retaining the key components of the powertrain. The developed Simulink model provides a simulation of the vehicle movement both in straight-line motion and in turn, with the fact that loads of the drive group are real only with a uniform movement. The simulation results show significant advantages of hybrid drive when performing turns, although it is clear that hybridization also has advantages related to straight-line motion, use of combined drive when starting the vehicle, silent mobility, regenerative braking, etc. Another important application of the Simulink model is its application in the development of an appropriate transmission control system, where it is necessary to integrate mechanical, electrical, and hydraulic components in order to obtain an optimal control system of the proposed hybridized drive group solution.

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