MATHEMATICAL MODELING OF VEHICLE DYNAMIC RESPONSE UNDER TRANSIENT ENGINE OPERATING CONDITIONS

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Abstract - Accurate prediction of vehicle dynamic response under transient engine operating conditions is important for the analysis of vehicle stability, handling, and safety under realistic driving conditions. This study develops a three-dimensional mathematical model for investigating the coupled dynamic behavior of a vehicle and its powertrain during transient engine operation. The vehicle is represented as a spatial mechanical system with a variable number of degrees of freedom depending on the wheel-slip conditions. The model is governed by a system of twelve nonlinear differential equations derived using Lagrange's equations and the fundamental laws of mechanics. It accounts for tire–road interaction forces, wheel normal reactions, suspension elasticity and damping, steering-wheel stabilizing moments, wheel rotational dynamics, and the time-dependent variation of engine torque. Two approaches are introduced for representing transient engine behaviour: the first is based on successive transitions through partial steady-state engine characteristics, whereas the second incorporates the transient evolution of cyclic fuel delivery. The specific contribution of the proposed framework lies in incorporating these two alternative representations of transient engine operation within the same spatial vehicle model, enabling their direct comparison in terms of the resulting vehicle dynamic response. The mathematical model is implemented in MATLAB/Simulink, and numerical experiments are performed to compare the resulting vehicle trajectories and velocity responses. The simulations show that both approaches predict stable curvilinear motion; over the investigated time interval, the vehicle velocity increases from approximately 5 m.s^{-1} to 13 m.s^{-1} for Approach 1 and to 15 m.s^{-1} for Approach 2, which also produces a greater lateral displacement. The proposed framework provides a computational basis for analyzing coupled vehicle–engine dynamics and for evaluating the influence of powertrain transients on vehicle motion.

Keywords: Vehicle dynamics; Transient engine operation; Nonlinear dynamic model; Multibody mechanical system; Powertrain–vehicle interaction; Tire–road interaction; MATLAB/Simulink; Numerical simulation.

1. Introduction

Mathematical modelling is widely used for the investigation of vehicle dynamic behaviour because it enables the interaction among the vehicle body, wheels, suspension, driveline and road surface to be described within a unified framework. The fundamental approaches cover longitudinal, lateral and vertical motion, load transfer, tyre–road interaction and the spatial representation of the vehicle as a multibody mechanical system [1–7]. Depending on the purpose of the investigation, the models range from detailed spatial formulations with a large number of degrees of freedom to simplified representations intended for analytical calculations and control applications.

The increasing complexity of dynamic systems has also stimulated the development of analytical and reduced-order modelling approaches. These methods preserve the dominant physical relationships while reducing the number of state variables and the computational effort required for transient analysis [8,9]. Their application is particularly useful when the objective is not only to reproduce the system response numerically, but also to identify the influence of individual parameters

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and obtain a physically interpretable solution. This modelling philosophy is relevant to vehicle dynamics, where excessive model complexity may complicate parameter identification and obscure the principal interactions among the engine, driveline, wheels and vehicle body. Experimental investigations of coupled mechanical dynamics, including bending and torsional vibrations, also provide complementary methodological background for the analysis of transient responses in mechanical systems [10].

The dynamic response of a vehicle during acceleration is strongly influenced by the behaviour of the engine and the power transmission system. Existing engine models describe torque generation, rotational-speed variation, fuel-supply processes and internal dynamic delays with different levels of detail [11–13]. When the engine is coupled with the transmission and the vehicle, the model must additionally account for rotating inertias, gear ratios, clutch operation and the resisting moments acting on the driveline [14–16]. Detailed thermodynamic formulations provide an extensive description of engine operation, whereas reduced or mean-value models are generally more suitable for complete vehicle simulations. Therefore, the selected representation should reproduce the essential variation of engine torque during a transient process without introducing parameters that cannot be reliably identified.

The numerical implementation of coupled vehicle–engine models is commonly performed in specialised simulation environments, which allow the governing differential equations and subsystem interactions to be integrated into a common computational structure [17]. The combination of analytical relationships and numerical verification represents an established engineering approach for the analysis of complex mechanical systems [18]. Nevertheless, the use of a simulation platform does not in itself ensure the validity of a model. The equations, initial conditions, subsystem connections and physical parameters must remain consistent with the actual configuration and operating conditions of the investigated system.

Transient vehicle operation has been examined in relation to acceleration, braking, driver response, collision processes and changes in engine operating conditions [19–24]. These studies confirm that vehicle motion during a transient regime cannot be represented adequately by assuming constant acceleration or an instantaneous change in engine torque. The response is determined by the coupled influence of torque development, driveline inertia, wheel rotation, tyre forces and external resistance. The investigated formulations, however, are frequently intended for particular manoeuvres or operating configurations and often analyse the vehicle, engine or driveline as partially separated subsystems.

Despite the substantial development of vehicle, engine and driveline models, an integrated mathematical formulation describing the influence of transient engine operation on the overall vehicle dynamic response remains limited.

Most existing approaches focus on individual subsystems rather than their coupled dynamic interaction. Therefore, the present study develops a spatial vehicle model incorporating two alternative approaches for modelling transient engine torque and evaluates their influence on vehicle dynamics.

The specific contribution of the present study is the integration of a spatial vehicle-dynamics formulation with two alternative representations of transient engine operation within a common simulation framework. This formulation enables a direct comparison of the influence of different transient engine representations on the predicted vehicle trajectory and velocity under identical operating conditions.

2. Dynamic Model of the Vehicle

The vehicle is modelled as a spatial mechanical system with twelve generalized coordinates describing the translational and rotational motion of the vehicle body, the wheel rotation, the steering motion of the front wheels, and the engine dynamics.

The twelve generalized coordinates comprise the three translational coordinates x_c, y_c, z_c of the vehicle mass centre; the three Euler angles ψ, θ, φ describing the spatial orientation of the vehicle body; the four wheel rotation angles γ_i (i=1,…,4); the steering coordinate describing rotation of the steerable wheels about their steering axes; and the rotational coordinate associated with the engine dynamics.

The kinematic constraints and the contact interactions between the tyres and the road surface vary depending on the driving conditions and the occurrence of longitudinal or lateral wheel slip. The adopted computational model of the vehicle is shown in Figure 1, including the principal coordinate systems and geometrical parameters used in the mathematical formulation.

The mathematical model consists of a system of twelve differential equations derived from the Lagrange equations of the second kind and the fundamental laws of mechanics [20,21]. The translational motion of the vehicle centre of mass with respect to the inertial reference frame is described by:

$$m \cdot \ddot{x}_c = \sum_{i=1}^{4}[F_{ix}] + m \cdot g \cdot \sin\alpha - w_x \cdot \sqrt{\dot{x}_c^2 + \dot{y}_c^2} \cdot \dot{x}_c \qquad (1)$$

$$m \cdot \ddot{y}_c = \sum_{i=1}^{4}[F_{iy}] + m \cdot g \cdot \sin\beta - w_y \cdot \sqrt{\dot{x}_c^2 + \dot{y}_c^2} \cdot \dot{y}_c \qquad (2)$$

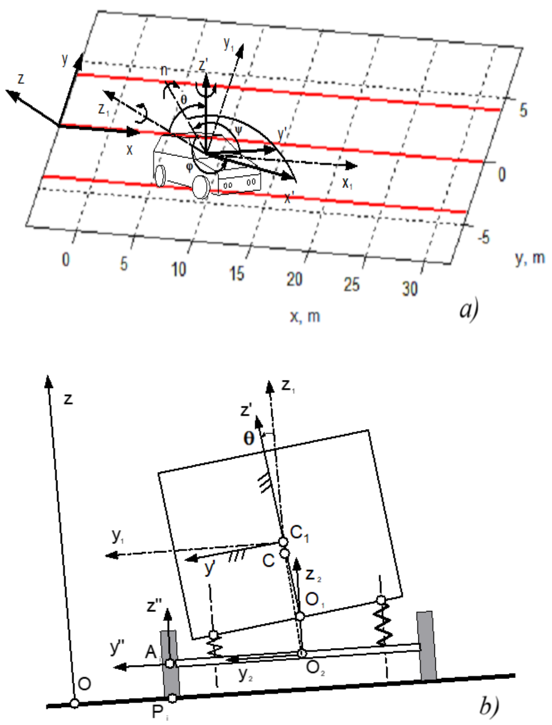

Figure 1: Computational model of the vehicle moving on a road surface: (a) vehicle model and adopted coordinate systems; (b) geometrical parameters and coordinate systems used in the mathematical formulation [2].

$$mm\cdot\ddot{z}_c=\sum_{i=1}^{4}N_i-\frac{m\cdot g}{\sqrt{1+\mathrm{tg}^2\alpha+\mathrm{tg}^2\beta}} \tag{3}$$

where:

- x_c, y_c, and z_c - the coordinates of the vehicle centre of mass;
- $F_{x,i}$ [N]and $F_{y,i}$ [N] - the longitudinal and lateral tyre forces acting at the wheel–road contact patches, respectively;
- N_i [N] - the normal tyre forces;
- α_x and α_y - denote the longitudinal and lateral road inclination angles;
- w_x, w_y - the aerodynamic drag coefficient;
- m [kg] - the vehicle mass,
- g $[m/s^2]$ – the gravitational acceleration.

The rotational motion of the vehicle body about its centre of mass is expressed in matrix form as:

The relationship between the time derivatives of the Euler angles and the angular velocity components of the body-fixed coordinate system is given by:

$$[A]\cdot[\dot{\alpha}]=[\omega] \tag{4}$$

$$[A]=\begin{bmatrix}\sin\theta\cdot\sin\phi & \cos\phi & 0\\ \sin\theta\cdot\cos\phi & -\sin\phi & 0\\ \cos\theta & 0 & 1\end{bmatrix};$$

$$[\dot{\alpha}]=\begin{bmatrix}\dot{\psi}\\ \dot{\theta}\\ \dot{\phi}\end{bmatrix};\quad [\omega]=\begin{bmatrix}\omega_{x'}\\ \omega_{y'}\\ \omega_{z'}\end{bmatrix}$$

$$[\dot{\alpha}]=\begin{bmatrix}\dot{\psi}\\ \dot{\theta}\\ \dot{\varphi}\end{bmatrix};[\omega]=\begin{bmatrix}\omega_{x'}\\ \omega_{y'}\\ \omega_{z'}\end{bmatrix}. \tag{5}$$

where:

- ψ, θ, ϕ [°] - the Euler angles;
- $[\dot{\alpha}]$ [rad.s^{-1}]- derivatives of the Euler angles with respect to time;
- ω [rad.s^{-1}] - angular velocity vector of the coordinate system rigidly attached to the vehicle body.

The rotational dynamics of the wheels and the associated driveline components are described by a system of four differential equations derived using the Lagrange method:

$$[I_\gamma]\cdot[\ddot{\gamma}]=[M_\gamma];m\cdot\ddot{x}_c$$
$$M_{\gamma i}=F_{i\tau}\cdot r_i+sign\,(\dot{\gamma}_i)\cdot[M_{di}-f_i\cdot N_i-M_{si}] \tag{6}$$

where:

- I_γ [kg·m^2] - square matrix whose elements depend on the equivalent moments of inertia of the wheels and the rotating components of the engine and transmission;
- γ_i (i=1,...,4) [°] - the wheel rotation angles about their axes;
- $\ddot{\gamma}$ [rads^{-2}]- the vector of angular accelerations;
- $F_{i\tau}$- tangential tyre force acting on the *i*-th wheel [N];
- r_i [m] - the dynamic rolling radius of the corresponding wheel;
- $f_{r,I}$ - the rolling resistance coefficient;

$$\vec{N_i}=c_{ok,i}\cdot\delta_{ok,i}-\beta_{ok,i}\cdot\dot{z}_{ok,i} \tag{7}$$

where:

- $c_{ok,i}$ [N·m^{-1}] - the equivalent vertical stiffness of the suspension and tyre;
- $\delta_{ok,i}$ [m] - the deformation of the equivalent elastic element;
- $\beta_{ok,i}$ [N·s·m^{-1}] - the equivalent damping coefficient;
- $M_{d,i}$ [N·m] - the driving torque referred to the corresponding driven wheel,
- $M_{b,i}$ [N·m] - the braking torque.

In the present formulation, the combined vertical compliance of the suspension and tyre is represented by an equivalent linear spring–damper element. Accordingly, the elastic and damping contributions are proportional to the deformation $\delta_{ok,i}$ and its time derivative $\dot{\delta}_{ok,i}$, respectively, with $c_{ok,i}$ and $\beta_{ok,i}$ treated as constant equivalent parameters for the investigated operating conditions.

The suspension travel is represented by the deformation $\delta_{ok,i}$ of the equivalent elastic element within the considered simulation range.

For the i-th wheel, the longitudinal slip ratio is defined as:

$$\kappa_i = \frac{r_i \dot{\gamma}_i - v_{x,i}}{max(|v_{x,i}|, v_\varepsilon)} \tag{8}$$

where:

- $v_{x,i}$ - the longitudinal velocity component of the wheel centre,
- $r_i \dot{\gamma}_i$ - the circumferential velocity of the wheel,
- v_ε - a small positive value introduced to avoid numerical singularity at very low vehicle speed.

The lateral slip angle is defined as:

$$\alpha_i = tan^{-1}\left(\frac{v_{y,i}}{max(|v_{x,i}|, v_\varepsilon)}\right) \tag{9}$$

where:

- $v_{y,i}$ - the lateral velocity component of the corresponding wheel.

The longitudinal and lateral tyre–road force components are limited by the available friction according to:

$$F_{x,i}^2 + F_{y,i}^2 \le (\mu_i N_i)^2 \tag{10}$$

where:

- μ_i - the tyre–road friction coefficient,
- N_i - the normal reaction at the i-th wheel.

Under sliding conditions, the resultant tangential force acts opposite to the relative slip velocity at the tyre–road contact.

The present study adopts equivalent elastic parameters for the suspension system. These parameters are physically associated with the mechanical response of the suspension spring elements, whose stiffness characteristics directly influence the vertical dynamics of the vehicle. Spring components are commonly manufactured from high-strength spring steels, and previous investigations on cold-drawn 1566 steel wire subjected to recrystallization annealing have demonstrated the significant influence of microstructural evolution on strength, ductility and elastic behaviour, providing a reliable material basis for the design and analysis of dynamically loaded suspension systems [26,27].

The steering motion of the steerable wheels about the kingpin axis is described by the following equation of motion [23,24]:

$$I_\theta \cdot \ddot{\theta}_k = \sum_{j=1}^{2} M_{kj} \tag{11}$$

where:

- $\theta_k(t)\ [rad]$ - the steering angle of the front wheels;
- I_θ [kg.m^2]- is the equivalent mass moment of inertia about the kingpin axis;
- M_s [N·m]- the self-aligning moment generated by the tyre–road contact forces.

The engine dynamics are introduced through the moment equilibrium equation of the crankshaft:

$$I_d \cdot \frac{d\omega_d}{dt} = M_{di} - M_c \tag{12}$$

where:

- M_c [N·m] - effective engine torque;
- M_{di} [N·m] - transient variation as a function of the cyclic fuel delivery.

The proposed mathematical formulation integrates the translational and rotational motion of the vehicle body, the wheel dynamics, the steering system, and the engine behaviour into a unified spatial model. This enables the analysis of vehicle trajectory, velocity, acceleration, wheel loads, and tyre slip under various driving conditions.

3. Simulation Model

Based on the mathematical formulation presented in the previous section, a simulation model was developed in the MATLAB/Simulink environment. The model performs the numerical integration of the governing differential equations and enables time-domain analysis of the principal dynamic variables of the vehicle.

To investigate the influence of engine operation on vehicle dynamics under transient driving conditions, two alternative approaches were developed for modelling the variation of the effective engine torque.

In Approach 1, the transient process is represented by a sequential transition of the engine through a series of equally spaced partial steady-state characteristics, from the initial operating point to a new steady-state condition or until the external speed characteristic is reached. After completion of the transient process, the engine continues to operate along the corresponding steady-state characteristic, as illustrated in Fig. 2.

The engine characteristic is represented by two regions, namely a linear region and a nonlinear region.

The linear region is described by:

$$M_{di}^{l} = k \cdot n_d + a_i \tag{13}$$

where:

- a_i [N.m]- the perpendicular distance from the origin of the coordinate system to the corresponding straight line;
- k [N·m·min]- the slope of the linear characteristic;

- n_d [min⁻¹]- the crankshaft rotational speed.

The nonlinear region corresponds to operation along the external speed characteristic and is approximated by a quadratic function of the form:

$$M_{di}^{II} = a_1 + a_2 \cdot n_d + a_3 \cdot n_d^2 \quad (14)$$

where a_1 [N·m], a_2 [N·m·min], and a_3 [N·m·min^2] are approximation coefficients determined from experimental data.

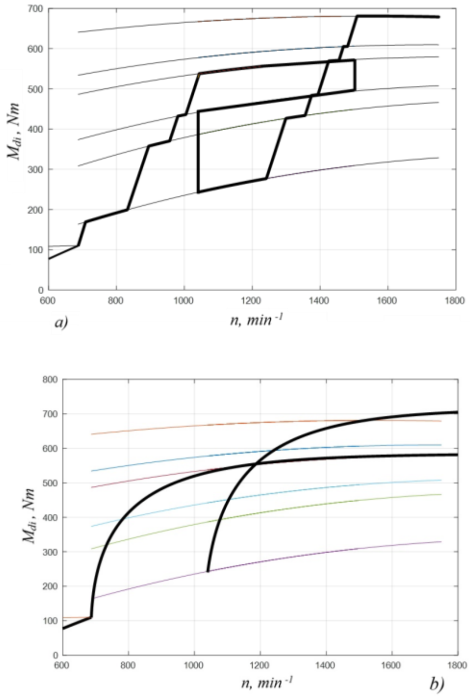

Figure 2: Effective engine torque characteristics during vehicle acceleration: (a) transient transitions through partial steady-state engine characteristics; (b) corresponding engine torque characteristics for different transmission gears.

In Approach 1, experimentally obtained relationships for the effective engine torque and the cyclic fuel delivery are employed [20-22]. The effective engine torque is calculated from:

$$M_i = a_1 + a_2 \cdot q + a_3 \cdot \omega_d + a_4 \cdot q^2 + a_5 \cdot \omega_d^2 + a_6 \cdot q \cdot \omega_d \quad (15)$$

The cyclic fuel delivery is determined by:

$$q_c = b_1 + b_2 \cdot \omega_p + b_3 \cdot h + b_4 \cdot \omega_p^2 + b_5 \cdot h^2 + b_6 \cdot \omega_p \cdot h \quad (16)$$

where:

- ω_d [rad·s⁻¹]– the engine crankshaft angular speed;
- $a_1, a_2, a_3, a_4, a_5, a_6, b_1, b_2, b_3, b_4, b_5, b_6$ - approximation coefficients;
- ω_p [rad·s⁻¹] - the angular speed of the fuel injection pump shaft;

$$\omega_p, \ (\omega_p = i_p \cdot \omega_d) \quad (17)$$

- i_p is the transmission ratio between the engine crankshaft and the pump shaft;
- $h = \varphi_p . i$ - the position of the fuel-control rack;
- φ_p - the rotation angle of the fuel-control pedal;
- i is the transmission ratio between the control mechanism and the fuel rack.

Equations (15) and (16) generate the complete family of partial engine characteristics corresponding to different positions of the fuel-control mechanism, together with the external speed characteristic.
This provides a more accurate representation of engine operation under different operating conditions.

In Approach 2, the transient response is described using the following relationship [23]:

$$q_c = q_0 + q_1 \cdot (1 - e^{-a \cdot t}) \quad (18)$$

which represents the time-dependent variation of the cyclic fuel delivery.

The principal distinction between the two approaches lies in the representation of the transient engine torque. In Approach 1, the effective engine torque is determined through the family of partial steady-state characteristics generated from the experimentally obtained torque and cyclic fuel-delivery relationships. In Approach 2, the cyclic fuel delivery is treated explicitly as a time-dependent quantity according to Eq. (18), thereby introducing the transient evolution of the fuel-supply process directly into the torque calculation.

The proposed simulation model accounts for the inertial behaviour of the fuel supply system and represents the transient variation of the effective engine torque. Consequently, it enables the analysis of vehicle dynamic response under transient operating conditions.

4. Numerical Experiments and Results

To evaluate the performance of the developed model, numerical experiments were conducted under different engine loading and control conditions. The investigated operating scenarios included vehicle acceleration, transient variation of the engine torque, and the possible occurrence of wheel slip.

The obtained results describe the time variation of the vehicle velocity and acceleration, the angular speeds of the wheels, and the reactions at the tyre–road contact patches. The graphical results presented in Figs. 3 and 4 enable the influence of transient engine operation on the vehicle dynamic response, stability, and handling behaviour to be evaluated.

The simulations demonstrate that variations in the engine torque have a significant influence on the vehicle dynamic characteristics and may cause noticeable changes in the tyre–road adhesion conditions.

The variation of the effective engine torque and the corresponding cyclic fuel delivery obtained using Approach 2 are illustrated in Fig. 3(a) and Fig. 3(b), respectively.

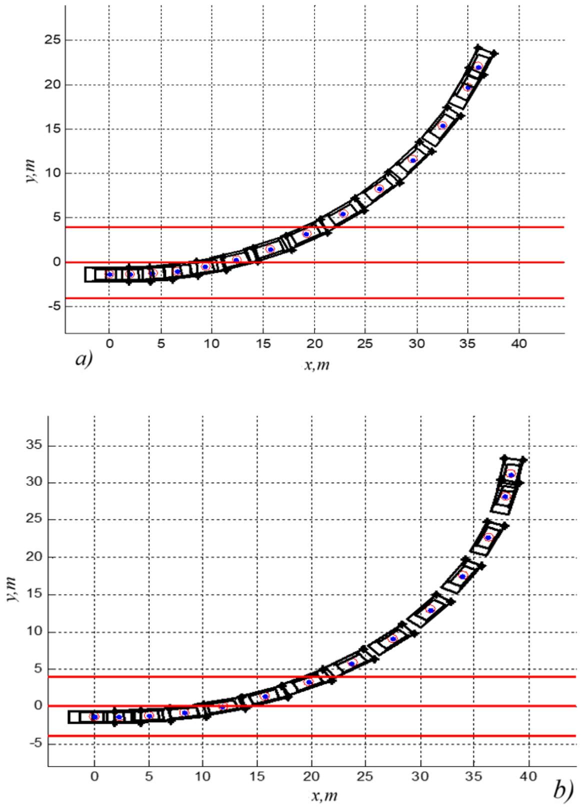

Figure 3: Transient engine response predicted using Approach 2: (a) variation of the effective engine torque; (b) variation of the cyclic fuel delivery.

In both cases, the vehicle follows a stable curvilinear trajectory without loss of directional stability or controllability. Nevertheless, certain differences in the vehicle position along the trajectory can be observed. These differences result from the different time-dependent variations of the engine torque predicted by the two models.

For Approach 2, the vehicle reaches a greater lateral displacement over the same time interval. This behaviour results from the more intensive increase in vehicle speed and the corresponding increase in the lateral inertial force acting during curvilinear motion.

The predicted vehicle trajectory and vehicle velocity obtained using the two transient engine modelling approaches are compared in Fig. 4(a) and Fig. 4(b), respectively.

The initial velocity is approximately 5 m.s^{-1}. At the end of the analysed time interval, it reaches approximately 13 m.s^{-1} for Approach 1 and 15 m.s^{-1} for Approach 2.

The obtained results indicate that Approach 2 predicts a more intensive increase in the effective engine torque and, consequently, a higher vehicle acceleration.

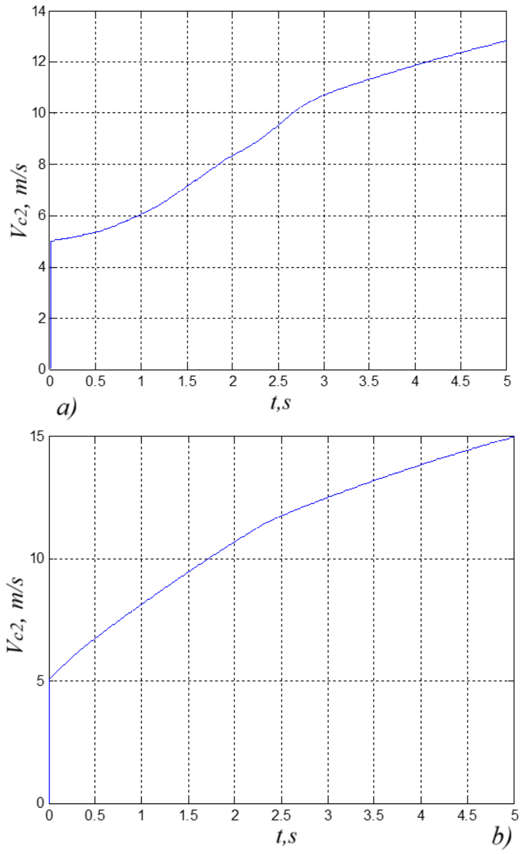

Figure 4: Comparison of the simulation results obtained using the two transient engine modelling approaches: (a) predicted vehicle trajectory; (b) predicted vehicle velocity.

Therefore, higher vehicle velocities are obtained throughout the investigated time interval. At the same time, the velocity variation remains smooth, confirming that the adopted relationship provides a physically consistent representation of transient engine torque development.

The numerical experiments demonstrate that the developed spatial mathematical model provides a consistent description of vehicle dynamics under transient engine operating conditions. The predicted vehicle trajectories and velocity characteristics correspond to the expected physical behaviour of the system. The model can therefore be applied to analyse and evaluate vehicle dynamic performance, directional stability, and handling under different operating conditions.

The comparative analysis shows that both transient engine modelling approaches predict the same general character of the vehicle motion. However, quantitative differences are observed in the vehicle velocity and lateral displacement, particularly toward the end of the investigated time interval. These differences demonstrate the sensitivity of the vehicle dynamic response to the adopted representation of transient engine torque.

4. Discussion

The numerical simulations demonstrate that the developed mathematical model provides numerically consistent vehicle responses within the assumptions adopted for transient engine operating conditions. Both transient engine models produce similar qualitative trends in the principal dynamic variables, while quantitative differences are observed in the predicted vehicle trajectory and velocity. These differences indicate that the selected transient engine model influences the predicted vehicle dynamic response, particularly during acceleration.

The obtained numerical results indicate that the representation of transient engine torque can influence the predicted longitudinal and lateral vehicle response. The proposed modelling approach provides a computational basis for analysing transient vehicle dynamic behaviour and for the further development and comparative evaluation of vehicle dynamic simulation models.

5. Conclusions

This study presents a spatial mathematical model for analysing vehicle dynamic response under transient engine operating conditions. Two alternative approaches for modelling the transient variation of the effective engine torque were incorporated into the simulation model and evaluated through numerical experiments.

The simulation results indicate that transient engine operation influences the predicted vehicle dynamic behaviour, including the vehicle trajectory and velocity characteristics. Although both modelling approaches predict stable vehicle motion, quantitative differences are observed during the transient acceleration process, indicating the sensitivity of the predicted vehicle response to the adopted engine torque model.

The developed model provides a unified computational framework for investigating the coupled interaction between transient engine operation and vehicle motion. Its methodological contribution lies in the possibility of implementing and directly comparing alternative transient engine representations within the same spatial vehicle model under identical operating conditions. Owing to its modular structure, the proposed methodology can be extended to include more advanced tyre models, transmission dynamics, traction control systems, and other driver assistance or vehicle control algorithms, making it suitable for further numerical studies of vehicle dynamic response under transient operating conditions. Further work will focus on extending the range of investigated operating scenarios, performing systematic parametric sensitivity analyses, and experimentally validating the proposed modelling framework using measured vehicle data under controlled transient operating conditions.

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