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Research on Using Model Predictive Control for the Truck Dry Friction Clutch Control during the Starting-Up Process

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Abstract

Automating friction clutch engagement is essential to improving vehicle dynamic and driving comfort. The study proposes a Model Predictive Control (MPC) strategy for automated clutch engagement in truck launching, evaluated through simulation. The powertrain models are developed for designing and evaluating the controller. The MPC algorithm calculates the required friction torque to be transmitted through the clutch by minimizing the deviation between actual and desired parameters. Key performance metrics, including longidutianal jerk, specific friction work, and the dynamic load factor, are used to assess the effectiveness of the proposed control strategy. To further evaluate the controller's impact on ride comfort, longitudinal jerk is analyzed through a co-simulation approach using specialized software. Simulation results for a first-gear launch under moderate intensity conditions show that the specific friction work is 18.4 J/cm², the dynamic load factor is 1.8, and the longitudinal jerk is 16.84 m/s³. These results confirm that the proposed MPC-based clutch control strategy ensures smooth engagement, enhances driving comfort, and meets performance requirements.

Keywords: MPC, dry friction clutch control, starting-up control, jerk, truck.

1. Introduction

Dry friction clutches and mechanical transmissions have been widely used in trucks in Vietnam and worldwide [1]. Numerous studies have been conducted on friction clutches, with a particular focus on the urgent need for automatic control of the clutch system, attracting significant attention from researchers worldwide. Continuous clutch pedal operation, especially in heavy traffic conditions, can lead to driver fatigue and stress. Ha N.T. [2] found that driver fatigue and stress are the primary causes of traffic accidents, with drivers experiencing high-stress levels being 4.2 times more likely to be involved in accidents compared to those without stress symptoms. Additionally, for inexperienced drivers, improper clutch engagement may result in jerky vehicle movements, negatively impacting both driver and passenger comfort as well as the performance and lifespan of drivetrain components. Li J. *et al*. [3] analyzed the influence of clutch pedal travel and actuation force on driving comfort and convenience, revealing that incorrect adjustments can cause vibrations and reduce ride smoothness. Furthermore, improper clutch operation accelerates wear on components such as the friction disc and related parts, thereby reducing clutch lifespan [4]. Excessive wear of these components can lead to clutch slippage, reducing power transmission efficiency and increasing the risk of drivetrain failure. To enhance vehicle performance, extensive research has been conducted on automatic clutch control. Among these studies, vehicle start-up control has been of particular interest in dry friction clutch automation. Several studies have explored automatic launch processes in trucks, considering factors such as vehicle body vibrations, drivetrain dynamic load coefficients, and specific friction work [5]. Previous research has employed fuzzy logic algorithms [6] to control automatic clutch engagement, analyzing control stability and evaluating controller performance based on specific friction work criteria. Yulong Lei *et al*. [7] investigated the relationship between clutch engagement speed and vehicle body jerking during launch, as well as the correlation between engine speed and specific friction work, ultimately proposing a control strategy based on engine speed and clutch engagement speed. Le V.N. [8] examined launch control using feedback on the angular velocity difference between the driving and driven components of the dry friction clutch, assessing controller effectiveness based on specific friction work and drivetrain dynamic load coefficient. Previous studies have primarily estimated vehicle jerking during launch by analyzing the rotational acceleration of drivetrain components rather than directly considering vehicle body movement. This study focuses on the automatic clutch control of a Hino 300 truck during the launch phase using a Model Predictive Control (MPC)

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approach. The performance of the MPC controller is evaluated based on key parameters such as specific friction work, drivetrain dynamic load coefficient, and longitudinal vehicle body vibration, calculated through a full-vehicle dynamic model developed in specialized simulation software.

MPC has been increasingly applied in automotive automation systems, including autonomous vehicle control [9, 10]. This paper focuses on the application of MPC to control the start-up process of the truck, utilizing a simulation-based approach to model clutch engagement during the launch process. The predictive controller employs a model-based approach to forecast the future response of the system at discrete time steps over a defined prediction horizon. Based on these predictions, an optimization algorithm computes the future control signal sequence within the control horizon to minimize the deviation between the predicted response and the reference signal. Predictive control is a generalized time-domain control design approach applicable to both linear and nonlinear systems. However, since linear system controllers are easier to develop, the drivetrain system in this study has been simplified and linearized to reduce computational resource demands. The MPC controller's output serves as the input for a longitudinal full-vehicle dynamic model developed in specialized software, enabling the calculation of launch-induced vibrations while accounting for wheel slip effects.

2. Drivetrain Modeling

The schematization of the powertrain system was carried out based on the principles of regular dynamic schemes [5, 8]. To facilitate analysis, the actual mechanical structure of the truck powertrain was replaced with an equivalent dynamic model, as illustrated in Fig. 1. This model consists of elastic-dissipative elements, which characterize the system's ability to transmit torque and absorb vibrations, along with lumped inertial masses, representing the mass distribution of key drivetrain components.

To ensure a realistic yet computationally efficient representation, the powertrain system was modeled using the multi-body system approach, which simplifies the transmission of power from the engine to the wheels while preserving the fundamental dynamics of clutch engagement and vehicle launch.

This approach allows for a more accurate evaluation of torque transmission, energy dissipation, and dynamic interactions within the drivetrain while maintaining a structure suitable for analysis and simulation. The resulting simplified model, depicted in Fig. 1, serves as the basis for designing and testing the MPC-based clutch control strategy under various operating conditions [11-14].

In this figure we use the following notation:

Γ_e : Engine torque, Nm.

Γ_c : Clutch friction torque, Nm.

Γ_w : Wheel torque, Nm

Γ_d : Diffirential torque,Nm

J_e: Engine inertia torque, kg.m^2

J_g: Transmission inertia torque, kg.m^2

J_t: Axle-shaft inertia torque, kg.m^2

J_w: Wheel inertia torque, kg.m^2

i: Transmission gear ratio

k_t: Axle-shaft stiffness, Nm/rad.

β_t : Damping coefficient of axle-shaft, Nm.s/rad

F_x: Longitudinal force at the wheel, N

The Lagrange-d'Alembert principle, a well-established concept in classical mechanics, was employed in the mathematical formulation of the developed powertrain dynamic model. This principle facilitates the application of analytical mechanics to derive the equations of motion governing individual elements within the drivetrain system. In the formulated model, the angular velocities of the inertial masses and the torques in the elastic-dissipative elements were chosen as generalized coordinates, providing a comprehensive representation of the system’s dynamic behavior. This approach allows for an accurate characterization of torque transmission, elastic deformations, and dissipative effects within the drivetrain. As a result, the simulation model is expressed using Type II Lagrange's differential equations, which serve as the foundation for analyzing the system’s dynamic response.

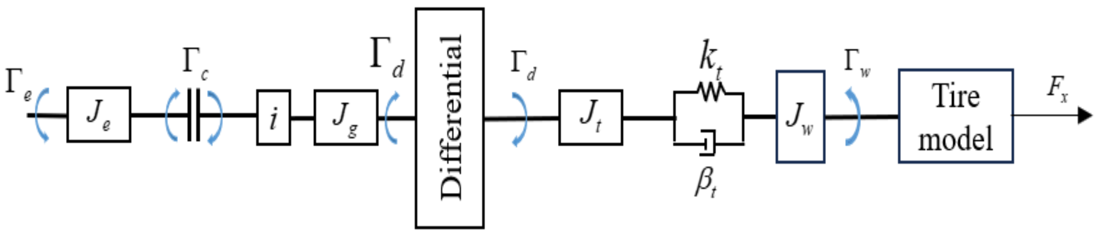

Fig. 1. Powertrain system diagram for simulation

Additionally, based on Newton's Second Law, the motion of individual elements in the powertrain system is described by the following system of equations:

$$\begin{aligned}
&J_e.\dot{\omega}_e = \Gamma_e - \Gamma_c \\
&J_g.\dot{\omega}_g = i.\Gamma_c - \Gamma_d \\
&J_t.\dot{\omega}_t = \Gamma_d - k_t.\theta_t - \beta_t.(\omega_t - \omega_w) \\
&J_w.\dot{\omega}_w = k_t.\theta_t + \beta_t.(\omega_t - \omega_w) - R_w.F_x \\
&\dot{\theta}_t = \omega_t - \omega_w
\end{aligned} \tag{1}$$

Where:

ω_e : angular velocity of the engine, rad/s;

ω_g : angular velocity of the shaft after gearbox, rad/s;

ω_t : angular velocity of axle shaft, rad/s;

ω_w : angular velocity of the wheel, rad/s;

$\dot{\theta}_t$: velocity deviation between axle shaft and wheel, rad/s;

The system of differential (1) allows the determination of specific friction work and dynamic load factor in the transmission system when the friction torque is transmitted through the clutch, according to [5, 8, 14]. The torque is transmitted to the drive wheels, generating longitudinal force that enables vehicle motion. In the following section of the study, the longitudinal dynamic model of the vehicle is presented to evaluate jerking vibrations during the starting-up process.

3. MPC Controller Design

To simplify the design of the MPC controller, it is necessary to linearize the control plant model, which, in this case, is the powertrain system. Since accurately describing the dynamics of the powertrain system using the derived differential equations is highly complex, this paper introduces several assumptions to simplify and linearize the system:

– Neglect the torsional deformation of components in the powertrain system and assume that the speed after the clutch is equal to the speed of the driven part;

– The tires are only subjected to rolling resistance force;

– The moments of inertia are referred to the clutch through the transmission ratio;

– All friction within the system and the resistance caused by stirring the lubricating oil in the transmission system are converted through the damping coefficient.

After simplification, the system of equations becomes with**.**

$$\begin{cases} J_e.\dot{\omega}_e = \Gamma_e - \Gamma_c \\ (J_g^{'} + J_v^{'})\dot{\omega}_g = \Gamma_c - \Gamma_{\text{resis}} \end{cases} \Rightarrow \begin{cases} J_e.\dot{\omega}_e = a.\omega_e - \Gamma_c \\ (J_g^{'} + J_v^{'})\dot{\omega}_g = \Gamma_c - \Gamma_{\text{resis}} \end{cases} \tag{2}$$

Where $\Gamma_e = f(\mathrm{w}_e(n)) = a_n.\mathrm{w}_e$, Γ_{resis} is the equivalent resistive torque from the wheels and the drivetrain, reflected to the output side of the clutch.

In (2), the inertia moments of the powertrain components are referred to the friction clutch, respectively.

The above system of equations is rewritten in the following form:

$$\begin{pmatrix} \dot{\omega}_e \\ \dot{\omega}_g \end{pmatrix} = \begin{pmatrix} \dfrac{a}{J_e} & 0 \\ 0 & 0 \end{pmatrix} \begin{pmatrix} \omega_e \\ \omega_g \end{pmatrix} + \begin{pmatrix} -\dfrac{1}{Je} \\ \dfrac{1}{J_g^{'} + J_v^{'}} \end{pmatrix} \Gamma_c + \begin{pmatrix} 0 \\ -\dfrac{T_{can}}{J_g^{'} + J_v^{'}} \end{pmatrix}$$

and is represented in the state-space form:

$$\dot{x} = \boldsymbol{A}.x + \boldsymbol{B}.\Gamma_c + d \tag{3}$$

Where:

$$A = \begin{bmatrix} \dfrac{a}{J_e} & 0 \\ 0 & 0 \end{bmatrix}, B = \begin{bmatrix} \dfrac{-1}{J_e} \\ \dfrac{1}{J_g{'} + J_v{'}} \end{bmatrix},$$

$$x = \begin{bmatrix} \omega_e \\ \omega_g \end{bmatrix}, d = \begin{bmatrix} 0 \\ -\dfrac{\Gamma_{can}}{J_g{'} + J_v{'}} \end{bmatrix}$$

The research subject used in this paper is the Hino 300 truck [15], with the specifications of its components presented in Table 1. The rotation inertia of the transmission parts is converted to the friction clutch through the rotation ratio.

Table 1. Input parameters for simulation

Parameter	Symbol	Value	Unit
Engine rotational inertia	J_e	2.5	$kg.m^2$
Transmission rotational inertia	J_g	0.047	$kg.m^2$
Inertia of the left half-shaft	J_{tl}	0.012	$kg.m^2$
Inertia of the right half-shaft	J_{tr}	0.012	$kg.m^2$
Driver wheel inertia	J_w	3.62441	$kg.m^2$
Full-load Vehicle mass	M	5500	kg
First gear ratio of the gearbox	i_1	5.342	
Transmission final drive ratio	i_{tlc}	4.625	
Wheel dynamic radius	R_w	349.5	mm
Rotational inertia of the driveshaft	$J_{cđ}$	0.0082	$kg.m^2$
Rotational inertia of the pinion gear	J_{cd}	0.00029	$kg.m^2$
Rotational inertia of the ring gear	J_{brvc}	0.03537	$kg.m^2$

At each sampling point, the signals are computed by optimizing the cost function J to minimize its value as following:

$$J = \sum_{k=1}^{N}[L_x(\Delta\omega_e(t+k/t), \Delta\omega_g(t+k/t)) + L_u(\Gamma_c(t+k/t))] \tag{4}$$

where: $\Delta\omega_e(t+k/t), \Delta\omega_g(t+k/t), \Gamma_c(t+k/t)$ represent the deviations of the engine speed, driven part speed, and the friction torque transmitted through the clutch at time $t+k$, while L_x, L_u are the cost functions used to adjust the priority levels of the components.

During the control process, the cost function must require low computational effort to avoid impacting the real-time performance of the algorithm. Therefore, it is rewritten in the following:

$$J(U) = k_1\sum_{k-1}^{n}\left[\omega_e - T(\omega_e)\right]^2 + k_2\sum_{k-1}^{n}\left[\omega_g - T(\omega_g)\right]^2 + \sum_{k-1}^{n}\Gamma_c^2 \tag{5}$$

From the above equations, the paper develops an engagement control model, as presented in Fig. 2:

The input signals for the MPC controller consist of two components: the reference value and the instantaneous values of the driving (ω_e) and driven component speeds (ω_g). The matrix $\boldsymbol{A}$ is defined based on the engine torque at the initial moment of clutch engagement. The controller predicts the future response of the system at discrete time steps within a defined prediction horizon. These predicted responses are utilized in an optimization algorithm to minimize the cost function $J(U)$, ensuring that the deviation between the predicted value and the given reference signal is minimized. The output of the MPC controller is the required friction torque to be transmitted through the clutch. This torque value is then used as an input to the powertrain system model, enabling the calculation of the next-time-step velocities of both the driving and driven components during the engagement process.

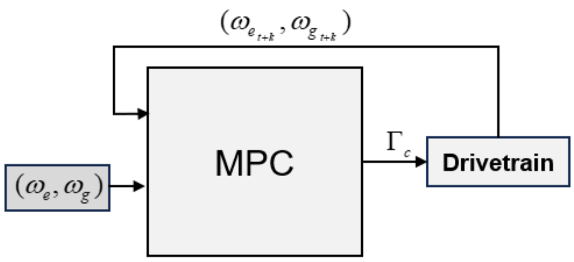

Fig. 2. Working principles of MPC controller

To effectively implement the MPC controller, it is essential to select appropriate control parameters. In this study, a combination of the Tuner tool and the trial-and-error method was used to determine the most suitable parameter set, with the primary objective of achieving an optimal clutch engagement time.

The designed MPC parameters are as follows:

- Sampling time: 0.1 s;
- Prediction horizon: 7 steps;
- Control horizon: 3 steps;
- Weight coefficients: 1:0:1.

The selected control parameters maintain a balance between computational efficiency and control performance, ensuring that the clutch engagement process is executed smoothly and within the desired time constraints.

The friction torque computed by the MPC controller is used to evaluate three key performance indicators: specific friction work of the clutch, dynamic load factor, and longitudinal jerk. These metrics are essential for assessing both operational efficiency and driver comfort.

To further analyze longitudinal jerk, the longitudinal vehicle dynamics model, which accounts for tire slip during vehicle launch, is employed. This model enables a comprehensive evaluation of vehicle vibrations during clutch engagement. The simulation is conducted using Simcenter software, with the corresponding block diagram representation illustrated in Fig. 3.

In SimCenter, the vehicle's longitudinal dynamics is described as following equation [16]:

$$M.\dot{V} = \sum F_x \tag{6}$$

Where:

V : vehicle velocity

$\sum F_x$: longitudinal force

The model is developed under the assumption that the forces F_z acting on each wheel are equal and constant. With the input being the friction torque transmitted through the clutch, the model calculates the longitudinal slip ratio (k), from which the longitudinal jerk is determined. The simulation is conducted using SimCenter software, employing the Simplified Pacejka tire model [17] to compute the longitudinal force F_x based on the reference tire characteristic curve.

$$k = 100.\frac{R_{roll}.\omega_{wh} - V_x}{|V_x|} \tag{7}$$

$$F_x = D.\sin\left[C.\arctan(B.k)\right]$$

$$\text{with } B = \frac{BCD}{C.D} \tag{8}$$

In these expressions, D is peak factor, C is shape factor, BCD is stiffness factor, R is the rolling radius of the wheel, V is the longitudinal velocity.

4. Results and Discussion

To evaluate the effectiveness of the controller during the launch process, a simulation scenario is designed with the following initial conditions:

- The truck launches on an asphalt road with a rolling resistance coefficient as specified in [5, 8].

- The desired reference speed values are defined as shown in Fig. 4 [8].

- The engine torque-speed characteristic curve is illustrated in Fig. 5 [17].

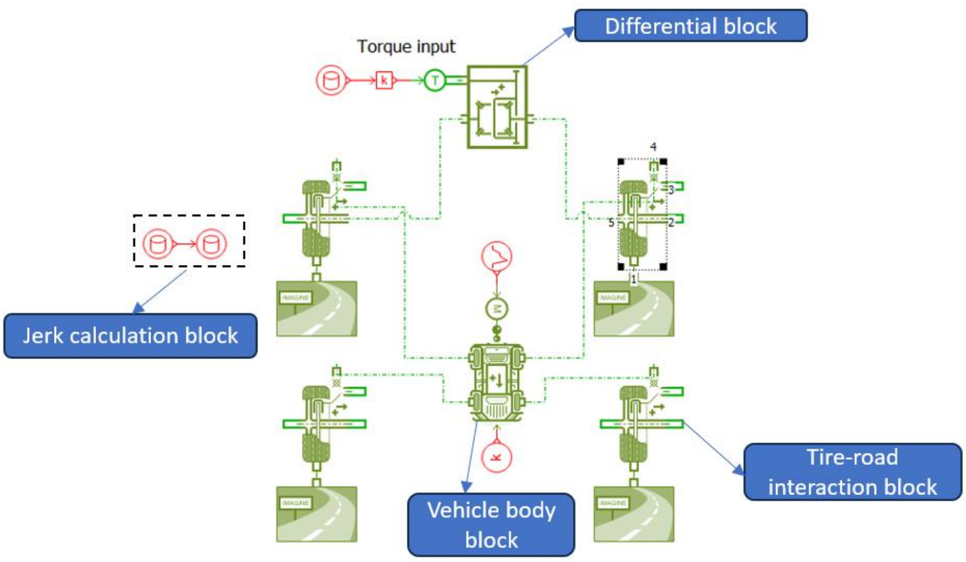

Fig. 3. Longitudinal dynamic model of the truck

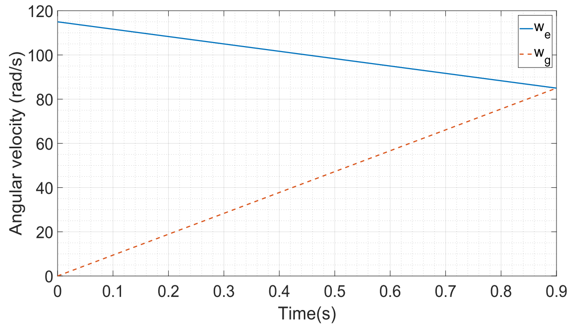

Fig. 4. Reference of angular velocities

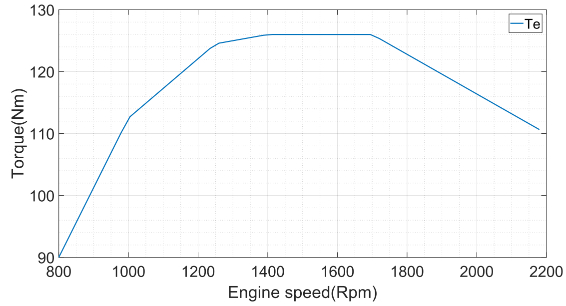

Fig. 5. Assumed engine characteristics

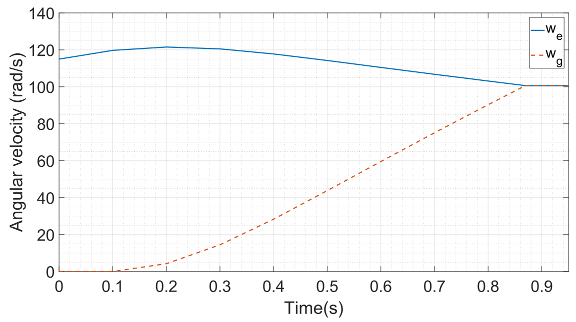

Fig. 6. Angular velocities simulation result

The simulation results of the clutch engagement process are presented in Fig. 6, illustrating the system's response under the designed control strategy. The total engagement time, measured from the initial clutch closing to full engagement, is 0.87 seconds, corresponding to an average clutch engagement rate of 1,1 s^{-1}. The differences between the desired and simulated time of clutch engagement are small, around 3.3%. These results indicate that the clutch engages smoothly and within a reasonable time frame for practical vehicle operation. From the graph shape and the engagement duration, it can be observed that the MPC controller maintains a stable response throughout the engagement process, effectively tracking the reference speed values of the driving and driven components. The controlled torque application ensures a gradual and well-regulated clutch engagement, minimizing sudden torque fluctuations that could lead to excessive wear or driver discomfort.

Fig. 7 illustrates the specific friction work of the clutch, which is determined based on the torque output from the controller and the relative speed deviation throughout the engagement period.

Under the assumed rolling resistance conditions outlined earlier, and for first-gear engagement, the maximum specific friction work remains below the allowable threshold specified in [5, 8]. This finding confirms that the proposed clutch engagement control strategy effectively regulates torque transmission, ensuring that the wear rate of the clutch friction disc stays within the acceptable range [8]. As a result, the strategy contributes to enhancing clutch durability and improving the overall reliability of the powertrain system.

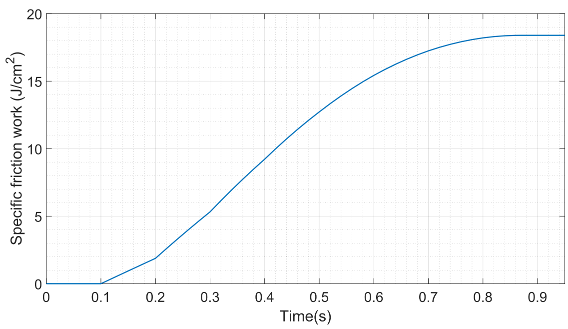

Fig. 7. Specific friction work

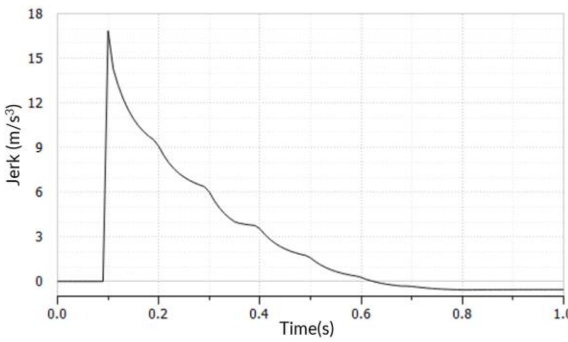

Fig. 8. Vehicle longitudinal jerk

The longitudinal vehicle dynamics model was developed based on the equations (6). Solving this system of equations allows for the computation of longitudinal jerk during truck motion, which serves as a key metric for evaluating driver comfort under the operation of the proposed MPC-based controller.

The simulation results, presented in Fig. 8, indicate that the peak longitudinal jerk reaches 16.84 m/s^3. The recommended jerk limit varies across different countries. In this study, the reference limit is based on the Chinese standard, which specifies a threshold of 17.64 m/s^3 [14]. Given that the obtained jerk value remains below this limit, the results suggest that the clutch engagement process does not induce driver discomfort according to the proposed standard. However, further research could explore the system's behavior under varied clutch engagement rates, providing a more comprehensive evaluation of the controller's robustness across different starting conditions.

The dynamic performance parameters of the powertrain system during the vehicle start-up process were computed using the developed dynamic model. The results indicate that the specific friction work is 18.4 J/cm², while the dynamic load factor is 1.8. These values remain within the recommended limits, as established in previous studies [5, 8, 14], thereby confirming the effectiveness of the proposed MPC-based control strategy in ensuring stable clutch engagement and optimized power transmission during the start-up phase.

5. Conclusion

This study developed a powertrain system model and a longitudinal dynamic model for a truck, serving as the foundation for designing and evaluating an MPC-based controller for automated clutch engagement during vehicle start-up process. The simulation results for a first-gear launch with a rolling resistance coefficient $f = 0.02$ and a clutch engagement rate of 1.1 s^{-1} demonstrate that the proposed control strategy effectively regulates the engagement process while maintaining key dynamic performance metrics within recommended limits. Specifically, the specific friction work is 18.4 J/cm², the dynamic load factor is 1.8, and the longitudinal jerk is 16.84 m/s³, all of which comply with established standards.

The results confirm that the MPC-based control strategy ensures smooth and stable clutch engagement, mitigating excessive jerk and improving driver comfort. Furthermore, by optimizing the engagement process, the controller helps minimize clutch wear, thereby enhancing the longevity and reliability of the powertrain system. These findings highlight the practical feasibility of implementing MPC-based clutch control in commercial truck applications.

Future research can focus on further optimization of the control parameters, incorporating nonlinear dynamics and real-time adaptation mechanisms to improve robustness under varying road and load conditions. Additionally, experimental validation on a physical system would provide further insights into the real-world applicability of the proposed approach.

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