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Investigation on dynamic electromechanical coupling effects in diesel generator sets

Simulation Technologies, Digital Twins and Complex System Simulation

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ABSTRACT

Due to the exceptional reliability and cost-effectiveness, the diesel generator (DG) set is extensively utilized in marine power systems and nuclear power plants. The performance of the DG set is significantly influenced by the interaction between the mechanical and electrical systems. As DG sets progress toward higher power density and greater electromechanical integration, this interaction becomes more pronounced. However, the current DG sets are designed conventionally by analyzing the mechanical system and the electrical system individually, which may lead to unexpected vibrations and oscillations, thus resulting in premature aging and catastrophic failure of the mechanical components, as well as instability of the power system. Therefore, to investigate the interaction mechanism and avoid failures caused by the electromechanical oscillation, this paper proposes a novel modelling method for DG sets which considers the interaction of the mechanical and electrical system.

First, the electromechanical coupling model between the diesel engine speed control system and the shafting torsional vibration is developed. Second, the model of the synchronous generator and its excitation system is established, and the interrelationship between the generator electrical system and the torsional vibration of the shaft system is deduced based on the Lagrange theory. As a result, the overall electromechanical coupling model of the DG set is obtained. To verify the effectiveness of the proposed modelling method, the validation is carried out using the experiment data under various operating conditions. Finally, the influence of the engine speed governor and the shafting parameters on the steady-state and transient performance of the DG set is investigated based on this model. The results illustrate that the controller parameters and the shafting parameters have a coupling impact on the electromechanical dynamic characteristics. And mismatching of the mechanical and electrical parameters will cause oscillation and intensified torsional vibration of the shaft system.

1 CINTRODUCTION

Diesel generator sets (DG sets) are crucial in modern power systems, providing reliable energy solutions across various applications, such as marine power systems, nuclear power plants, and data centers [1-3]. DG sets consist of critical mechanical and electrical subsystems, including diesel engines, generators, and control systems. The efficient coordination between these subsystems is essential for stable operation of DG sets. Any mismatch between these subsystems may lead to performance degradation, impacting the safety and efficiency of the entire power system. Therefore, it is crucial to reveal the interaction mechanisms and study their influence on the dynamic behavior of the DG set.

The interaction between the mechanical and electrical subsystems, known as the electromechanical coupling (EMC) effect, is widespread in electromechanical systems. As research advances, the importance of understanding EMC dynamics has grown, especially in high-reliability applications like wind power [4, 5], railway systems [6, 7], and electric vehicles [8, 9]. EMC effects significantly impact mechanical vibrations [6, 8], electrical dynamics [7, 9], and overall system performance of electromechanical systems [4]. For instance, Chen et al. [5] provided a dynamic model for the gear system in wind turbines, identifying dangerous resonance speed. Hu et al. [8] proposed an active damping control strategy for the electric drive system of electric vehicles to mitigate the torsional vibration, thereby extending the service life of the electric drive system.

DG sets, as typical electromechanical energy conversion devices, are similarly affected by these EMC effects. However, traditional designs often treat DG sets as independent mechanical and electrical systems, analyzing them separately. Regarding mechanical vibrations, torsional vibration is the most common form in DG sets, and there is extensive research on torsional vibration model simplification, nonlinear dynamics, transient torsional vibration response, and practical approaches to reduce vibration [10-13]. However, these studies typically treat the shaft system as an open-loop system, overlooking the regulatory effects of the control system, the influence of speed fluctuations on the engine torque, and the real-time impact of the electromagnetic torque. Consequently, while such simplifications reduce the computational complexity, they will induce considerable errors under transient conditions characterized by significant changes in speed and torque. At the same time, electrical engineers generally focus on electrical dynamics [14]. Moreover, with the increasing complexity of power grids, these control systems have progressively incorporated intelligent control technologies to enhance control system performance and improve responsiveness and adaptability under multiple conditions [15-17]. However, in these studies, the engine torque is typically assumed to be constant, and the shaft system is simplified as a rigid rotor. These assumptions ignore the periodic fluctuations in the engine torque and several qualitative dynamic properties of the mechanical systems, such as inertial distribution, torsional flexibility, and damping effects. As a result, the influence of torsional vibration on the electrical dynamics cannot be investigated with sufficient accuracy.

To summarize, although existing methods are useful for analyzing individual subsystems, they limit the understanding of the overall dynamic performance of DG sets. As DG sets advance towards higher integration and power density [18-20], the interactions between mechanical and electrical components have become more critical. The limitations of traditional analysis methods in analyzing these interactions have become one of the critical bottlenecks in improving the overall performance of DG sets. Therefore, there is an imperative need to refine existing dynamic models to gain a more profound understanding of these interactions. This is especially pertinent in applications like shipboard and industrial microgrids, where DG sets frequently encounter transient conditions due to the limited capacity and frequent load variations [21,22]. In such environments, substantial variations in external torque can lead to severe EMC failures, thereby heightening the demands for system stability and reliability. Addressing the EMC challenges in DG sets is crucial for boosting their dynamic performance in these high-demand applications.

To address the challenges, this study proposes an electromechanical dynamic model that can be used for both steady-state and transient conditions of DG sets. This comprehensive model considers the intricate interaction between the mechanical and electrical subsystems of DG sets. Furthermore, experimental validation is conducted to verify the accuracy and reliability of this model under various operating conditions, including steady-state and transient scenarios. On this basis, a thorough investigation is carried out to analyze the impact of shafting parameters and diesel engine speed control system parameters on the dynamic characteristics of DG sets, as well as their interaction.

The paper is organized as follows: Section 2 investigates the interaction mechanism between the mechanical and electrical systems of DG sets and presents the EMC model. Section 3 conducts several experiments to verify the accuracy and effectiveness of the proposed model under various working conditions. Section 4 examines the influence of the shafting parameters and the diesel engine speed control system parameters on the dynamic characteristics of the DG set and their interaction. Section 5 provides the conclusions.

2 MATHEMATICAL MODELING

As depicted in Figure 1, a DG set primarily comprises a diesel engine, transmission components, a synchronous generator, and control systems. Based on the operational principle of the DG set, it can be equivalently divided into two main subsystems: the mechanical system and the electrical system. The mechanical subsystem encompasses the engine crankshaft, transmission components, and generator shaft. The electrical subsystem includes the generator stator winding, the engine speed control system (responsible for frequency regulation), and the generator excitation system (accountable for voltage regulation). The synergistic operation of these subsystems ensures the efficient functioning of the DG set, thus forming a mechanical-electrical coupling system.

This section develops an EMC dynamic model that accounts for the interaction between the shaft system torsional vibration and the electrical system of DG sets. First, an EMC model is developed to capture the interaction between the diesel engine speed control system and the torsional vibration of the shaft system. Then, the coupling between the generator electrical system and the shaft system torsional vibration is further discussed, with their interaction relationship deduced based on Lagrange's theory. As a result, the overall EMC model for DG sets is developed.

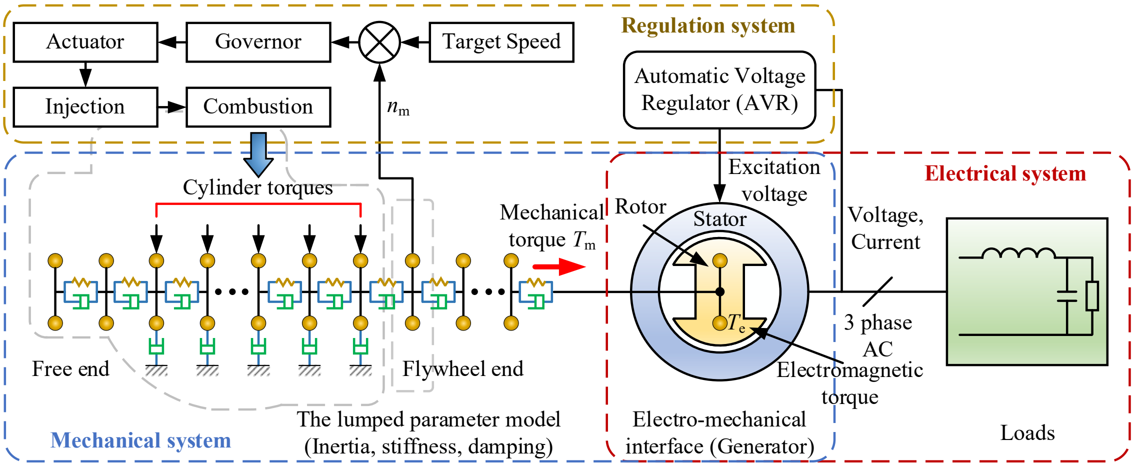


Figure 1. The EMC mechanism of the DG sets.

2.1 Modeling of diesel engine speed control system coupling with torsional vibration

Figure 2 illustrates the coupling mechanism between the diesel engine speed control system and shaft system torsional vibration. In operation, the speed control system typically employs the shaft system instantaneous speed as feedback to regulate its operation. This regulation affects the cylinder torque, resulting in variations in the instantaneous speed, which is the derivative of the torsional angle response. Thus, a closed-loop EMC exists between the diesel engine speed control system and the shaft system torsional vibration.

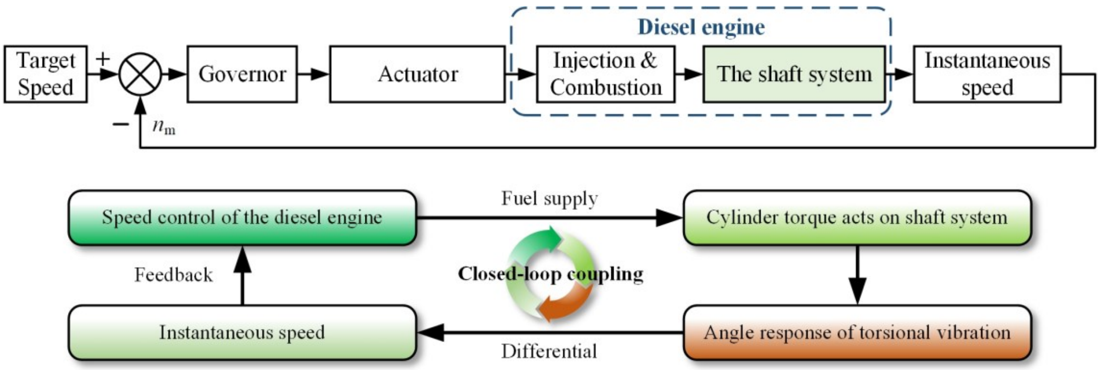


Figure 2. Mechanism of engine speed control system coupling with shaft system torsional vibration.

However, in traditional research, the diesel engine speed control system and the shaft system torsional vibration are typically studied in isolation. Neglecting the interplay between these systems may result in unforeseen vibrations and oscillations [23]. Hence, this section meticulously models the diesel engine cylinder torques and the shaft system, and develops the EMC model between the speed control system and torsional vibration.

2.1.1 Engine cylinder torque

The engine cylinder torque is mainly caused by the tangential force acting on the crankpin, primarily from the variation of the in-cylinder pressure and the inertial load of the reciprocating components. Figure 3 shows the diagram of the engine-kinematic, allowing the torques acting on the crankshaft to be derived. In this diagram, D is the cylinder diameter, S is the stroke, L is the connecting rod length, R is the crank radius, β is the conrod angular travel, $\alpha = \omega t$ is the crankshaft angle, ω is the crankshaft angular velocity, F_p the force acting on the piston, F_g is the gas force, F_i is the inertial force, F_{cr} is the force acting on the connecting rod axis, and F_t is the tangential force acting on the crankpin.

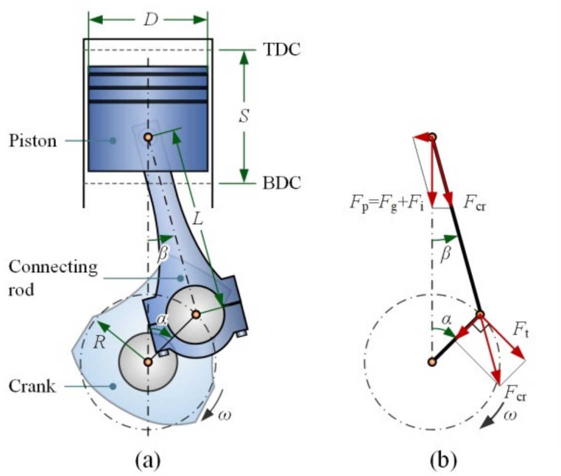


Figure 3. Diesel engine cylinder torque: (a) diagram of the engine-kinematic; (b) forces acting on crank mechanism.

The gas force F_g is generated by the in-cylinder pressure p_g. It is expressed as the product of the cylinder pressure P_g and piston area A_p.

$$F_{\mathrm{g}} = A_{\mathrm{p}} p_{\mathrm{g}} = \frac{\pi D^2}{4} p_{\mathrm{g}} \tag{1}$$

The inertial force F_i results from the inertial effect of the reciprocating components. This force is described as

$$F_{\mathrm{i}} = -mR\omega^2 (\cos\omega t + \lambda \cos 2\omega t) \tag{2}$$

Together, these two tangential forces yield the force acting in the axial direction of the piston.

$$F_{\mathrm{p}} = F_{\mathrm{g}} + F_{\mathrm{i}} \tag{3}$$

The tangential force acting on the crankpin:

$$F_{\mathrm{t}} = F_{\mathrm{p}} \frac{\sin(\alpha + \beta)}{\cos\beta} \tag{4}$$

The relationship between the crankshaft angle α and the conrod angular travel β is represented as follows:

$$\sin\beta = \lambda \sin\alpha \tag{5}$$

Therefore, Equation 4 is rewritten as follows:

$$F_{\mathrm{t}} = F_{\mathrm{p}} \sin\alpha \left(1 + \frac{\lambda\cos\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}}\right) \tag{6}$$

Finally, the instantaneous torque applied to the crankshaft is derived by multiplying the tangential force F_t by the crank radius R.

$$\begin{aligned} M_{\mathrm{cyl}} &= F_{\mathrm{t}} R \\ &= F_{\mathrm{p}} R \sin\alpha \left(1 + \frac{\lambda\cos\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}}\right) \\ &= F_{\mathrm{p}} R \sin\omega t \left(1 + \frac{\lambda\cos\omega t}{\sqrt{1-\lambda^2\sin^2\omega t}}\right) \end{aligned} \tag{7}$$

2.1.2 Shaft system

The shaft system of a DG set comprises several key components, including the engine, flexible coupling, and generator. To analyze the torsional vibration characteristics of the shaft system, a multi-degree-of-freedom equivalent shafting model is established using the lumped-parameter theory, as shown in Figure 4. This model includes the torsional vibration damper (inertias J_1 and J_2), the flanges (inertias J_3 and J_{13}), the cylinders (inertias J_4 to J_{11}), the gear system (inertia J_{12}), the flywheel (inertia J_{14}), the flexible coupling (inertias J_{15} and J_{16}), and the generator (inertias J_{17} to J_{19}).

Based on Lagrange's theory, the differential equation that represents the torsional vibration dynamic characteristics of the shaft system is derived and expressed as

$$[\boldsymbol{J}]\{\ddot{\theta}(t)\} + [\boldsymbol{C}]\{\dot{\theta}(t)\} + [\boldsymbol{K}]\{\theta(t)\} = \{\boldsymbol{M}(t)\} \tag{8}$$

where $\{\boldsymbol{\theta}\} = [\theta_1(t), \ldots, \theta_{17}(t)]^T$ is the torsional angle vector. $\{\boldsymbol{M}\} = [0, 0, 0, M_{cyl.1}(t), \ldots, M_{cyl.8}(t), \ldots, -M_{gen}(t)]^T$ represents the excitation torque, which varies according to the engine speed and load. $[\boldsymbol{J}]$ is the inertia matrix, $[\boldsymbol{K}]$ is the torsional stiffness matrix, $[\boldsymbol{C}]$ is the damping matrix. The damping matrix $[\boldsymbol{C}]$ is composed of two parts: the absolute damping matrix $[\boldsymbol{C}_a]$ and the relative damping matrix $[\boldsymbol{C}_r]$, respectively.

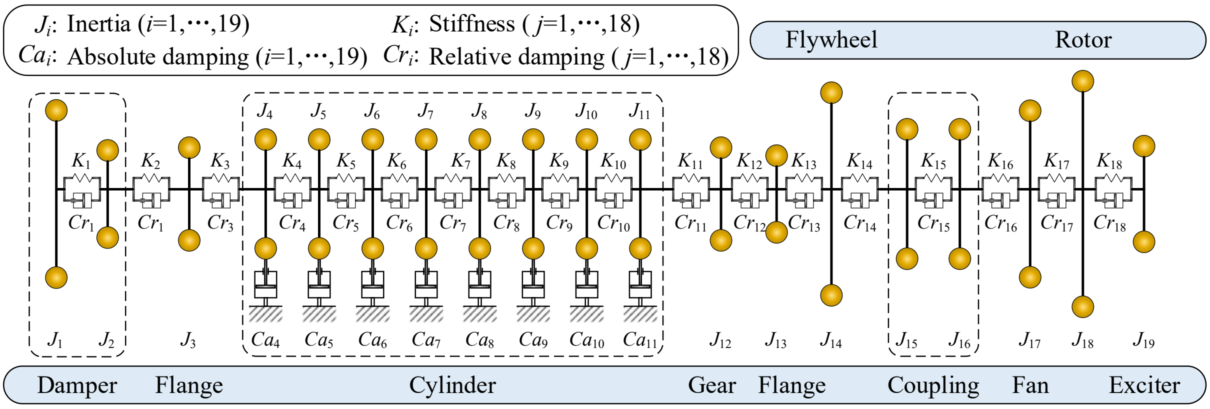


Figure 4. The multi-degree-of-freedom equivalent shafting model of the DG set.

Then, the relationship between the shaft system torsional vibration and the engine speed control system is established by replacing the angular displacement variable in Equation 8 with the shaft system instantaneous speed variable, as shown in the following equation.

$$\begin{cases} \theta_i = \int \dfrac{\pi}{30} n_i dt \\ \dot{\theta}_i = \omega_i = \dfrac{\pi}{30} n_i \qquad (i = 1, \cdots, 19) \\ \ddot{\theta}_i = \dot{\omega}_i = a_i = \dfrac{\pi}{30} \dfrac{dn_i}{dt} \end{cases} \tag{9}$$

2.2 Modeling of synchronous generator with its excitation system

The synchronous generator is the electromechanical conversion device in a DG set, which is responsible for converting mechanical energy into electrical energy. During this conversion, it produces electromagnetic torque acting on the generator rotor, influencing the performance and dynamic characteristics of the DG set. The electromagnetic torque is calculated by the generator model, which varies according to the working conditions of the DG set. In this subsection, the park transform is used to convert a three-phase synchronous machine into an equivalent two-phase machine under the d-q coordinate system (direct-quadrature system) to establish the synchronous generator analytical model. The equivalent circuits of the synchronous machine in the d-q reference frame, as illustrated in Figure 5, typically show the relationship between the voltage, current, and flux linkages in the machine.

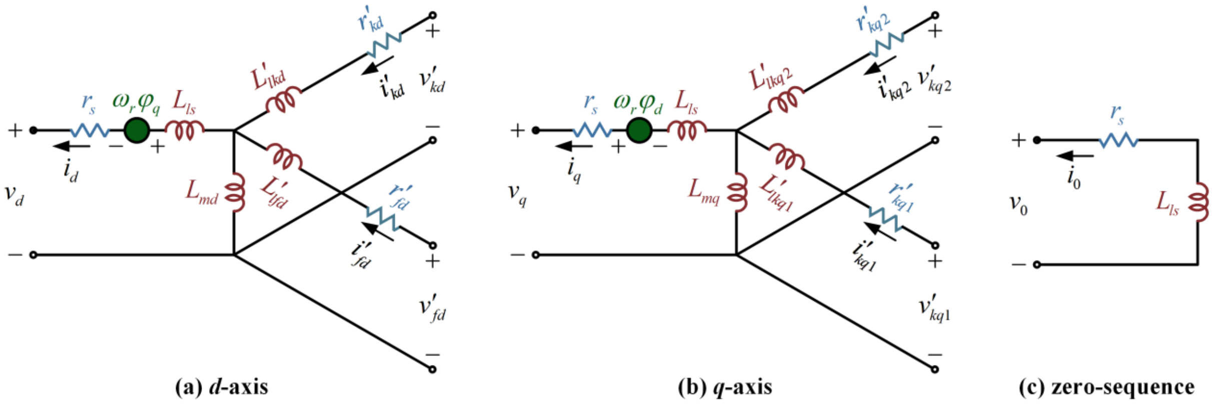


Figure 5. Equivalent circuits of the synchronous machine.

By applying Faraday's law of induction, the electrical part of the synchronous generator is based on the following voltage and flux equations [24, 25]:

$$\begin{bmatrix} v_d \\ v_q \\ v_0 \\ v'_{fd} \\ v'_{kd} \\ v'_{kq1} \\ v'_{kq2} \end{bmatrix} = \begin{bmatrix} r_s & & & & & & \\ & r_s & & & & & \\ & & r_s & & & & \\ & & & r'_{fd} & & & \\ & & & & r'_{kd} & & \\ & & & & & r'_{kq1} & \\ & & & & & & r'_{kq2} \end{bmatrix} \begin{bmatrix} -i_d \\ -i_q \\ -i_0 \\ i'_{fd} \\ i'_{kd} \\ i'_{kq1} \\ i'_{kq2} \end{bmatrix} + \begin{bmatrix} \dot{\varphi}_d \\ \dot{\varphi}_q \\ \dot{\varphi}_0 \\ \dot{\varphi}'_{fd} \\ \dot{\varphi}'_{kd} \\ \dot{\varphi}'_{kq1} \\ \dot{\varphi}'_{kq2} \end{bmatrix} + \begin{bmatrix} -\omega_r \varphi_q \\ \omega_r \varphi_d \\ 0 \\ 0 \\ 0 \\ 0 \\ 0 \end{bmatrix} \tag{10}$$

$$\begin{bmatrix} \varphi_d \\ \varphi_q \\ \varphi_0 \\ \varphi'_{fd} \\ \varphi'_{kd} \\ \varphi'_{kq1} \\ \varphi'_{kq2} \end{bmatrix} = \begin{bmatrix} L_{md}+L_{ls} & & & L_{md} & L_{md} & & \\ & L_{mq}+L_{ls} & & & & L_{mq} & L_{mq} \\ & & L_{ls} & & & & \\ L_{md} & & & L_{md}+L'_{lfd} & L_{md} & & \\ L_{md} & & & L_{md} & L_{md}+L'_{lkd} & & \\ & L_{mq} & & & & L_{mq}+L'_{lkq1} & L_{mq} \\ & L_{mq} & & & & L_{mq} & L_{mq}+L'_{lkq2} \end{bmatrix} \begin{bmatrix} -i_d \\ -i_q \\ -i_0 \\ i'_{fd} \\ i'_{kd} \\ i'_{kq1} \\ i'_{kq2} \end{bmatrix} \tag{11}$$

where v is the voltage, r is the resistance, i is the current, φ is the magnetic flux, and L is the inductance. Additionally, $\omega_r = p\omega_{\mathrm{mec}}$ is the rotor electrical angular velocity, where p is the number of pole pairs and ω_{mec} is the mechanical angular velocity. The subscripts d and q represent the axes of the dq frame, while s, r, m, l, k, and f represent the stator, rotor, magnetizing inductance, leakage inductance, damper winding, and field winding, respectively.

When the shaft system is simplified as a rigid rotor, the torsional dynamics are neglected, meaning that the system is assumed to behave as a single, rigid body without any internal twisting or oscillatory motions. In this case, the electromagnetic torque and motion equation can be expressed in the following form:

$$M_{\mathrm{gen}} = \frac{3}{2} p \begin{bmatrix} -\varphi_q & \varphi_d \end{bmatrix} \begin{bmatrix} i_d \\ i_q \end{bmatrix} \tag{12}$$

$$M_{\mathrm{gen}} + \begin{bmatrix} J_{\mathrm{s}} & C & M_{\mathrm{mec}} \end{bmatrix} \begin{bmatrix} \ddot{\theta}_{\mathrm{mec}} \\ \dot{\theta}_{\mathrm{mec}} \\ -1 \end{bmatrix} = 0 \tag{13}$$

where J_{s} is the total inertia of the shaft system, C is the damping coefficient, θ_{mec} is the mechanical angle of the generator rotor, M_{mec} is the mechanical torque from the engine, and M_{gen} is the electromagnetic torque from the generator.

In fact, the shaft system of a DG set is not an ideal rigid drivetrain but a flexible one. This means that, during operation, the shaft undergoes torsional, bending, or vibrational deformations, especially under uneven load conditions or rapidly changing external forces. The flexibility of the shaft allows for more complex dynamic behavior, including torsional vibrations and shaft deflections.

By applying Lagrange's theory, the equation of motion Equation 12 can be rewritten to reflect the torsional dynamics of the shaft system. The resulting equation, Equation 14, includes terms representing the shaft system torsional stiffness and damping, as well as the inertia of the rotating components.

$$\begin{aligned} J_{18}\ddot{\theta}_{18} &= k_{17,18}(\theta_{17}-\theta_{18}) + c_{\mathrm{r}17,18}(\dot{\theta}_{17}-\dot{\theta}_{18}) \\ &\quad - k_{18,19}(\theta_{18}-\theta_{19}) - c_{\mathrm{r}18,19}(\dot{\theta}_{18}-\dot{\theta}_{19}) \\ &\quad - c_{\mathrm{a}18}\dot{\theta}_{18} - M_{\mathrm{gen}} \end{aligned} \tag{14}$$

The revised equation of motion demonstrates the interrelationship between the shaft system torsional vibration and the generator electrical system. In this coupled system, mechanical torsional vibrations influence the electromagnetic behavior of the generator, and changes in the electrical load or generator torque, in turn, affect the mechanical vibration response.

As a result, the coupling relationship between the shafting torsional vibration and the electrical system is finally obtained.

Moreover, a phase-compounding brushless alternating current excitation system is commonly employed to maintain the stability of the DG set.

This system is pivotal in regulating the synchronous generator performance, especially under varying load conditions. By dynamically adjusting the excitation, it ensures smooth and stable operation, mitigating the impact of load fluctuations and supporting consistent, reliable power generation [26].

3 MODEL VALIDATION AND ANALYSIS

In this section, the EMC model established in Section 2 is verified using the experiment data of a DG set. The verification covers both the steady-state operation conditions and the abrupt load transient operation conditions. The parameters of the DG set are presented in Table 1, and the results are illustrated in Figure 5, Figure 6, and Table 2. Specifically, Figure 5 depicts the flywheel speed fluctuation under 100% load condition, Figure 6 shows the transient process of 100% load sudden unloading to 67% load, and Table 2 presents the peak-to-peak values (PPV) of the flywheel instantaneous speed. The results indicate that the simulation results of the flywheel instantaneous speed fluctuation are highly consistent with the experimental results under the steady-state and transient conditions, with errors below 5.0%. The model is used for further EMC characteristic analysis.

Table 1. Basic Parameters of the DG set.

Parameters	Values	Units
Diesel engine		
Nominal power P_m	2800	kW
Nominal speed n_m	750	r/min
Flexible coupling		
Torsional stiffness C_{Tdyn0}	260	kN·m/rad
Damping ψ_0	0.7	-
Vibratory torque T_{KW}	12.5	kN·m
Synchronous generator		
Nominal power P_e	2688	kW
Nominal speed n_e	750	r/min

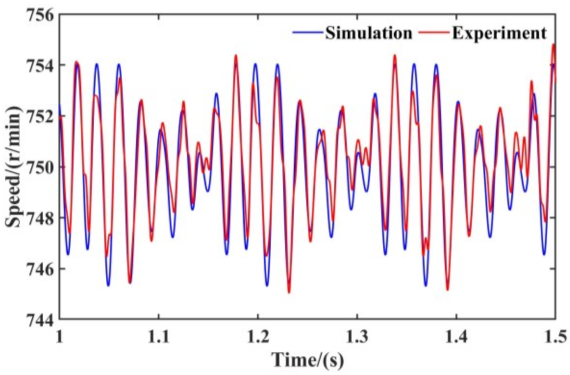


Figure 5. n_f under steady-state conditions.

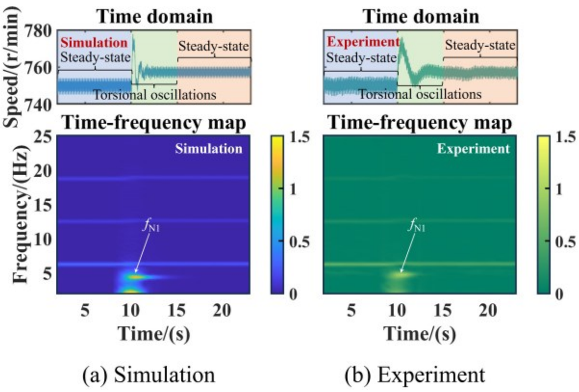


Figure 6. n_f under transient conditions.

Table 2. The peak-to-peak values of the flywheel instantaneous speed.

Condition	Experiment (r/min)	Simulation (r/min)	Maximum error (%)
100% load	9.77	9.31	4.71
100% load sudden unloading to 67% load	31.95	32.66	2.22

As shown in Figure 6, a significant transient torsional vibration occurs under the sudden load change condition. It is seen from the time-frequency diagram in Figure 6 that the first-order torsional vibration mode of the shaft system is excited due to the impact of the step load, and there is an obvious amplitude at the first-order natural frequency (f_{N1}). As further shown in Figure 7 and Figure 8, there are significant oscillations in the generator electromagnetic torque (T_e), flywheel instantaneous speed (n_f), diesel engine torque (T_M), and flexible coupling additional torque (T_W) during the transient process. Especially, the oscillation at the first-order torsional vibration node (flexible coupling) is especially obvious, and the oscillation frequency is equal to f_{N1}. Moreover, the additional torque of the flexible coupling significantly increases during the oscillation, which exceeds the permissible amplitude of a periodic vibratory torque that can be continuously endured by the coupling (T_{KW}). Shortening the duration of oscillations under transient load conditions and avoiding the DG set from long-term oscillation are of great importance in preventing mechanical component aging and damage, as well as enhancing the DG set performance.

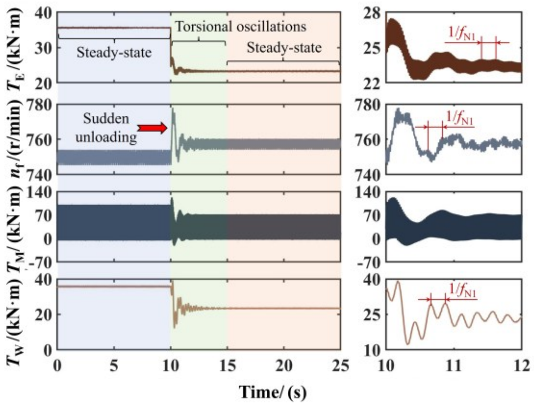


Figure 7. Transient oscillation of the DG set.

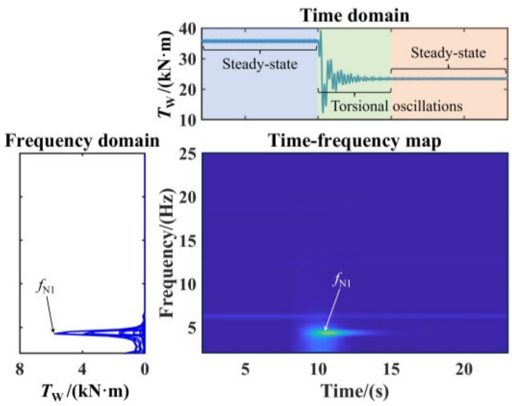


Figure 8. Time-frequency diagram of T_W.

4 PARAMETRIC ANALYSIS

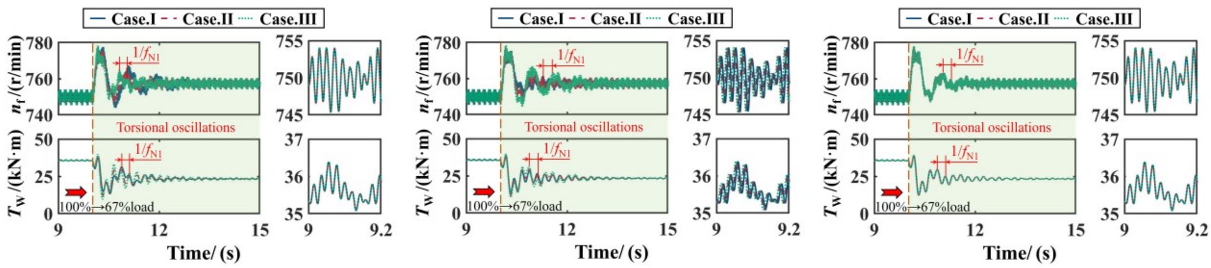
In this section, the influence of PID parameters of diesel engine speed control system and shafting parameters on torsional vibration characteristics, as well as the interaction between the controller parameters and shafting parameters are analyzed.

4.1 Interaction characteristics between controller parameters and shafting parameters

During the diesel operation, the speed control system ensures the stable operation of the diesel engine by regulating its working process, and the speed control system performance is directly influenced by PID parameters. To investigate the influence of speed controller parameters on EMC vibration characteristics, simulations are conducted with different PID parameters. The values of PID, the stable time of n_f (T_s), and the peak-to-peak values of T_W during the transient process (PPV_{CT}) are listed in Table 3. Figure 9 are the results of n_f and T_W. The results show that variations in P, I, and D do not significantly affect the steady-state vibration characteristics. However, the effect on transient vibration characteristics varies. Specifically, an increase in the P parameter leads to a reduction in T_s and a slight increase in PPV_{CT}. As the I parameter increases, T_S increases significantly, and PPV_{CT} increases slightly. And changes in the D parameter have no significant effect on transient characteristics.

Table 3. The controller parameters (P_0 = 7.50, I_0 = 20.0, D_0 = 2.0).

Group No.	A: P			B: I			C: D		
	Parameters	T_s (s)	PPV_{CT} (kN·m)	Parameters	T_s (s)	PPV_{CT} (kN·m)	Parameters	T_s (s)	PPV_{CT} (kN·m)
Case. I	80%P_0, I_0, D_0	1.11	25.49	P_0, 80%I_0, D_0	0.68	25.72	P_0, I_0, 80%D_0	0.72	27.18
Case. II	P_0, I_0, D_0	0.72	27.19	P_0, I_0, D_0	0.72	27.19	P_0, I_0, D_0	0.72	27.19
Case. III	120%P_0, I_0, D_0	0.54	30.38	P_0, 120%I_0, D_0	1.04	29.31	P_0, I_0, 120%D_0	0.72	27.20



(a) Group. A (b) Group. B (b) Group. C

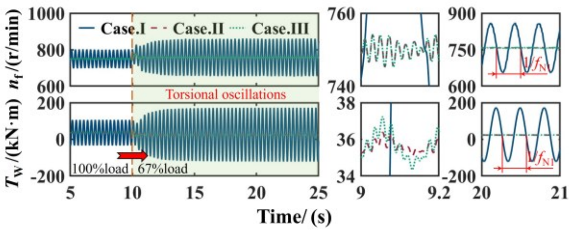
Figure 9. The influence of speed controller parameters.

4.2 Analysis of the influence of shafting parameters

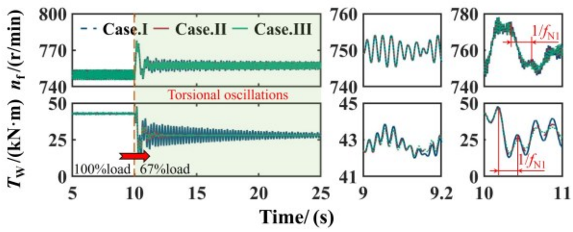
Modifying shaft system parameters of the DG set is an effective method for reduce torsional vibrations. However, neglecting electromechanical interaction during the modification may cause intensive oscillations and vibrations. To study the effect of typical shafting parameters on the EMC vibration characteristics of the DG set, simulations are conducted using the parameters shown in Table 4, and the n_f and T_W results are shown in Figure 10. It is evident that as the flexible coupling dynamic torsional stiffness (C_{Tdyn}) decreases, the transient torsional vibration gradually increases. And once the value is reduced to 50% of the design value (Group. D Case. I), the DG set enters a state of continuous periodic oscillation, with T_W exceeding T_{KW}. This continuous oscillation will cause damage to the flexible coupling. With the decrease of the flexible coupling damping coefficient (ψ), the transient oscillations increase, and when the value decreases to 50% of the design value (Group. E Case. I), significant oscillations with gradually decaying amplitudes are observed in T_W. When the flywheel inertia (J_f) increases from 50% to 150% of the design value, the peak-to-peak values of n_f and T_W are significantly reduced under both steady-state and transient conditions. On the other hand, an increase of the generator inertia (J_g) leads to amplified transient vibrations, and when it reaches 150% of the design value (Group. G Case. III), the DG set enters a state of continuous periodic oscillation with T_W exceeding T_{KW}. To summarize, the improper design of the shaft system may lead to oscillation and vibration. Equally, in the case of oscillation and vibration, the DG set can be restored to stable operation by adjusting the shafting parameters.

Table 4. The shafting Parameters (C_{Tdyn0} = 260 kN·m/rad, ψ_0 = 0.7, J_{f0} = 402.5 kg·m^2, J_{g0} = 468.49 kg·m^2).

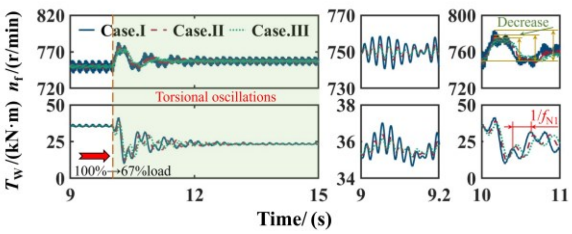
Conditions	D: Coupling stiffness	E: Coupling damping	F: Flywheel inertia	G: Generator inertia
Case. I	50%C_{Tdyn0}	50%ψ_0	50%J_{f0}	50%J_{g0}
Case. II	C_{Tdyn0}	ψ_0	J_{f0}	J_{g0}
Case. III	150%C_{Tdyn0}	150%ψ_0	150%J_{f0}	150%J_{g0}



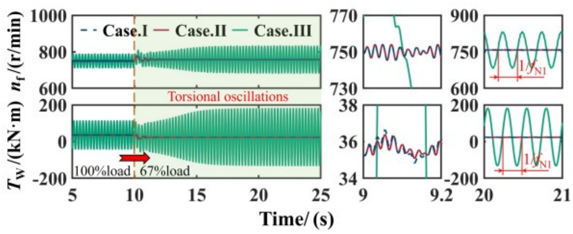
(a) Group. D



(b) Group. E



(c) Group. F



(d) Group. G

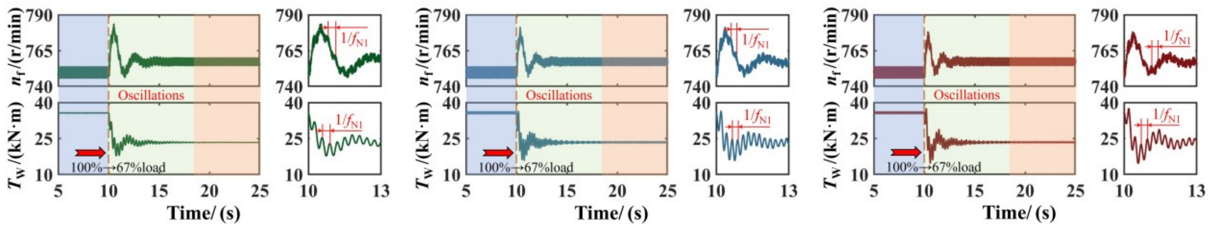
Figure 10. The influence of shafting parameters.

4.3 Interaction characteristics between controller parameters and shafting parameters

To analyze the interaction between the controller parameters of the speed control system and the shafting parameters, the interaction is analyzed by adjusting the controller parameters under the oscillating conditions (Group. D Case. I, Group. E Case. I, Group. G Case. III) in Section 4.2. The results are illustrated in Figure 11. The results show that a stable operation condition of the DG set is achieved by adjusting the P and I parameters according to Table 5.

Table 5. Optimized controller parameters. (P_0 = 7.50, I_0 = 20.0, D_0 = 2.0).

Conditions	Group. D Case. I	Group. E Case. I	Group. G Case. III
Optimized parameters	40%P_0, 25%I_0, D_0	50%P_0, 40%I_0, D_0	60%P_0, 55%I_0, D_0



(a) Group. D Case. I (b) Group. E Case. I (c) Group. G Case. III

Figure 11. Torsional vibration response of the DG set shafting.

Combining the research findings in this section as well as in sections 4.1 and 4.2, it is concluded that in the case of oscillation caused by mismatching of controller parameters and shafting parameters, the DG set is restored to stable operation by adjusting both the controller parameters and the shafting parameters.

5 CONCLUSIONS

To investigate the electromechanical coupling (EMC) dynamic behaviors of the diesel generator (DG) set, the present study establishes an EMC model that incorporates the interaction between the electrical system and the shafting system of the diesel generator set, which is validated through experiments. Furthermore, the influence of the controller parameters of the diesel engine speed control system and the shafting parameters on the torsional vibration characteristics, as well as the interaction of the two types of parameters are analyzed. The following conclusions are obtained:

1. The EMC model is verified under both the steady-state and transient conditions. The maximum errors of the flywheel instantaneous speed peak-to-peak values between the simulated and experimental results are 4.71% and 2.22%, respectively, demonstrating a high accuracy of the model.

2. The transient process results show that the first-order torsional vibration mode of the shaft system is excited and short-term transient torsional oscillation occurs under the sudden load change condition.

3. The changes of P, I and D parameters have no significant effect on the steady-state vibration. Effects on the transient vibration are various. Specifically, an increase in the P parameter leads to a reduction in the stable time of the flywheel instantaneous speed, while an increase in the I parameter has an opposite effect. Besides, the D parameter have no significant effect on transient characteristics.

4. The reduction in dynamic torsional stiffness or damping coefficient of the elastic coupling induces oscillations. Increasing the flywheel inertia is able to partially mitigate both steady-state and transient vibrations. An increase of the generator inertia leads to intensified transient vibrations, and the DG set enters a state of continuous oscillation when it reaches 150% of the design value.

5. In case of oscillation caused by mismatch between controller parameters and shafting parameters, the DG set is restored to a stable operation by modifying the shaft system parameters or re-adjusting the controller parameters of the diesel engine speed control system.

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