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RESEARCH ARTICLE

Effect of Different Combined Excitation on the Vibration Characteristics of Locomotive Gear Transmission

TAO ZHANG, XIAOHUI DANG, YONG HE, JIANLONG XIE, AND HE MA
School of Mechatronic Engineering, Lanzhou Jiaotong University, Lanzhou 730070, China

Corresponding author: Yong He (hylzjt2014@139.com)

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ABSTRACT With the continuous improvement of locomotive operating speed, increasing axle load and increasing traction power, the dynamic excitation of wheel-rail interface and gear pair interface have become more and more intense, directly affecting the operational reliability or even safety of the locomotive or the train. Based on the traditional rotor dynamics and the classical vehicle-track coupled dynamics, a locomotive-track coupled spatial dynamics model is established, in which the dynamic effects from drive system are considered. Under three different combinations of excitation, namely constant mesh stiffness, time-varying mesh stiffness and track geometrical irregularity, the vibration responses of the gear transmission are simulated during the uniform and accelerated operation of the locomotive. By comparing the difference of vibration response, mesh force and dynamic transmission errors of gear pair of traction motor under different combinations of excitation, the influence of different excitation on the vibration characteristics of gear transmission is explored. The following conclusions are drawn: 1) The time-varying mesh stiffness directly affects the vertical and longitudinal vibration of the traction motor. 2) The influence of time-varying mesh stiffness and track geometrical irregularity varies at different speed stages of locomotives. 3) During the acceleration process of the locomotive, resonance phenomenon of the traction motor and gear pair can be caused at a certain speed when considering time-varying mesh stiffness. The conclusions obtained will provide certain reference significance for the selection of design parameters, evaluation of dynamic characteristics, and life prediction of locomotive drive system.

INDEX TERMS Locomotive, gear transmission, vibration characteristics, time-varying mesh stiffness, track geometrical irregularity.

I. INTRODUCTION

Gear transmission is a key component in a locomotive drive system. Through gear transmission, traction or electric braking forces can be transmitted from motor to wheel–rail contact interface. With the continuous improvement of locomotive operating speed, increasing axle load and increasing traction power, the dynamic excitations of wheel-rail interface and gear pair interface have become more and more intense. Under the complicated excitations, the dynamic characteristics of locomotive gear transmissions are prone to be insufficient, such as vibration, noise, and slight wear, which can even lead to fatigue failure of traction motors and gear pairs, and even lead to/result in driving safety accidents. Therefore, exploring the effect of different excitations on the vibration characteristics of gear transmission is of great significance for the reliable, safe, smooth, and efficient operation of locomotive.

The vibration characteristics of gear transmission affect the quality of gear transmission. Many scholars around the

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world have paid more attention to the topics on dynamic model of gear transmission [1], [2], [3]. The dynamic model of gear transmission has gradually evolved from a simple torsional vibration model to a complex bending-torsion-shaft-pendulum coupled model. Based on different dynamic models of gear transmission, the vibration characteristics of gear transmission have been researched under different internal excitations of gear pairs (e.g. time-varying mesh stiffness, dynamic transmission errors) [4], [5], [6].

In fact, locomotive gear transmission is not only subjected to nonlinear dynamic excitation from the gear pair interface, but also to external excitation from the wheel-rail interface, such as track geometrical irregularity and changes in the parameters of the support structure under the track. Therefore, based on the operating status of locomotive gear transmission and their own working environment characteristics, many scholars have established different dynamic models to study the dynamic characteristics of the drive system. Hirotsu et al. [7], [8] established a single wheel drive system model to study the lateral stability of locomotive drive systems under saturated adhesion conditions. Kim et al. [9], [10] established a mechanical electrical coupled model of the drive system, and studied the control methods for improving wheel-rail adhesion, the effects of high-order harmonic torque as well as suspension stiffness of the traction motor on the dynamic characteristics of the locomotive drive system. Although the above scholars have revealed the coupling effect between the electrical and mechanical parts of the locomotive drive system under external excitation, the wheel-rail interface is still been oversimplified. Zhai [11] firstly proposed a classical vehicle–track coupled dynamics model called Zhai-Sun model, which fully revealed the coupling effect of the wheel-rail system. However, the drive system was simplified accordingly in Zhai-Sun model. Based on the classical vehicle–track coupled dynamics theory and longitudinal dynamics theory of the train, Liu et al. [12] established a three-dimensional coupled dynamic model for heavy-haul trains where both the dynamic impacts between vehicles and the dynamic interactions between the train and the track systems were considered. In some of the dynamic models aforementioned, effects of the gear transmission were dealt as the equivalent sprung and unsprung masses while the influence of the internal dynamic excitations of the gear pairs was not taken into account.

Leva [13] established a mechanical electrical coupled model of the drive system to study the dynamic characteristics of the locomotive gear transmission. However, a constant spring damping element was used to simulate the gear tooth mesh stiffness, ignoring the time-varying mesh stiffness. Chen et al. [14], [15], [16] established locomotive–track vertical and vertical-longitudinal coupled dynamics model with gear transmission system to reveal the coupling effects between the traditional vehicle–track system and the gear transmission system from a plane perspective, where the internal dynamic excitations of the gear transmission system, the wheel–rail nonlinear creep excitation, and the wheel–rail geometrical irregularities were considered. However, this model can't calculate locomotive safety evaluation indicators such as derailment coefficient and wheel load reduction rate.

In the actual operation process of locomotives, not only are there longitudinal motion, vertical motion, and nodding motion, but also the influence of lateral motion, rolling motion, and shaking motion of key components of the locomotive should be considered. Using the SIMPACK software, Huang et al. [17] established a motor car dynamics model with gear transmission system to analyze the dynamic behavior of the gear transmission system in a high speed train and its influence on the vehicle dynamic performance, and Wang et al. [18], [19] adopted the mobile mark point technology to establish a locomotive track spatial coupled dynamic model considering gear transmission, and studied the changes in the order and amplitude of wheel polygons, as well as the time-varying mesh stiffness changes caused by tooth crack size on the dynamic characteristics of locomotives and gear transmission.

Based on rotor dynamics theory and vehicle-track coupled dynamics theory, Zhang et al. [20], [21] established a locomotive-track spatial coupled dynamics model considering gear transmission, and studied the influence of traction motor suspension parameters on the vibration characteristics of locomotives and drive system under the action of time-varying mesh stiffness and track geometrical irregularity. Wang et al. [22], [23] established a high-speed train track spatial coupled dynamics model considering gear transmission, and studied the dynamic characteristics of high-speed trains and axle box bearings under different wheel damage states. However, the above research focuses more on analyzing the dynamic characteristics of locomotives under both internal and external excitation, and does not reveal the impact of time-varying stiffness and track irregularities on the vibration characteristics of gear transmission devices from a spatial perspective. Therefore, this article will establish a locomotive track coupled dynamic model considering the driving system, simulate and analyze the vibration response of the gear transmission device under three different combinations of excitation: constant stiffness, time-varying stiffness, and track irregularity during the uniform and accelerated operation of the locomotive. By comparing and analyzing the vibration response of the traction motor, gear tooth meshing force, and dynamic transmission error of the gear pair under different combinations of excitation, the impacts of different excitations on the vibration characteristics of gear transmission are clarified, providing a theoretical reference for the selection of design parameters, evaluation of dynamic characteristics, and life prediction of locomotive gear transmission devices.

II. LOCOMOTIVE–TRACK COUPLED SPATIAL DYNAMICS MODEL WITH GEAR TRANSMISSIONS

One end of the drive system is elastically suspended on the bogie frame through a suspender and a ball type

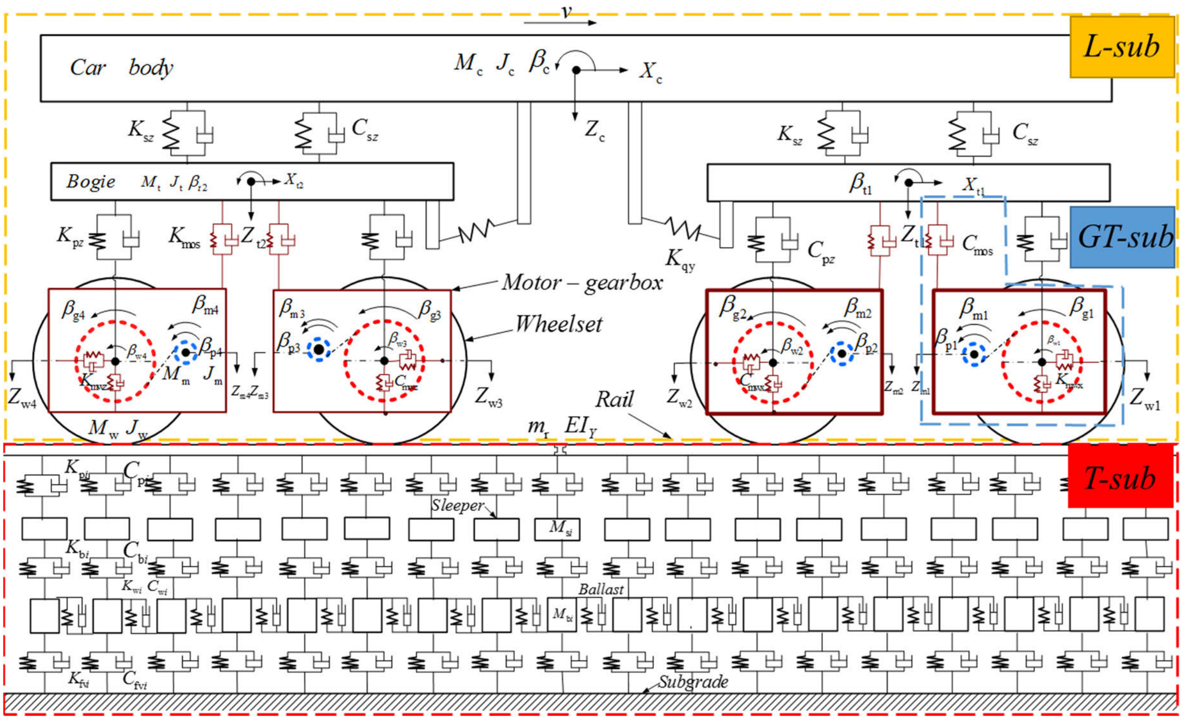


FIGURE 1. Locomotive–track coupled dynamic model with gear transmissions.

rubber joint, and the other end is supported on the axle by two axle-hung bearings. The torque of the traction motor is transmitted through the driving gear to the big gear which is installed on the axle. The big gear drives the wheelsets to rotate, thus driving the locomotive to run. Based on the classical vehicle–track coupled dynamics theory and gear dynamics theory, locomotive–track coupled spatial dynamics model with gear transmissions is established. Its schematic is shown in Fig.1. This developed dynamics model consists of three subsystems, namely the locomotive sub-model (L-sub), the gear transmission sub-model (GT-sub) and the track sub-model (T-sub). The locomotive-track coupled system dynamics model with gear transmissions is shown in Figure 1. The concrete modeling process can refer to the literature [20].

The differential equation of the entire coupled dynamics system can be expressed in the following matrix form as,

$$\mathbf{M}\ddot{\mathbf{X}}(t) + \mathbf{C}\dot{\mathbf{X}}(t) + \mathbf{K}\mathbf{X}(t) = \mathbf{P}(t) \tag{1}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ represent the mass, the damping and the stiffness matrices, respectively; $\mathbf{X}(t)$, $\dot{\mathbf{X}}(t)$, $\ddot{\mathbf{X}}(t)$ denote the displacement, velocity, acceleration vectors, respectively; while $\mathbf{P}(t)$ represents the force vector caused by the external excitation from the track geometrical irregularity and by the internal excitation from the time-varying gear mesh process.

The algorithm used for the equation adopts the fast explicit integration method (Zhai-method) [11], [20].

III. EXCITATION

The vibration of the locomotive drive system not only comes from the vibration excitation of the gear pair interface, but also bears the influence of track irregularity from the wheel rail interface.

A. MESH STIFFNESSBIPM

In fact, mesh stiffness varies with time periodically during the mesh process. Calculation of the time-varying mesh stiffness has been studied by many scholars using different models and methods. For example, Chen et al. [24], [25] proposed analytical models for gear mesh stiffness calculation based on the potential energy principle, where the bending, the shear and the axial compressive deformations of gear teeth, the fillet-foundation deformation, and the Hertz contact deformation were considered. The time-varying mesh stiffness in [24] and [25] is calculated as,

$$K_{\mathrm{m}}(t) = \frac{\sum_{j=1}^{N} K_j(t)}{1 + \sum_{j=1}^{N} K_j(t) E_{ij}(t)/F(t)} \tag{2}$$

where N represents the number of tooth pairs in mesh; E denotes the errors of tooth profile; The subscripts i and j denote the number of the tooth pairs; F is the total mesh force of the gear pair; K is the single-tooth mesh stiffness, which can be calculated as follows [24], [25],

$$\frac{1}{K_j(t)} = \frac{1}{K_{top}(t)} + \frac{1}{K_{ff\mathrm{p}}(t)} + \frac{1}{K_{tog}(t)} + \frac{1}{K_{ff\mathrm{g}}(t)} + \frac{1}{K_h(t)} \tag{3}$$

TABLE 1. Main design parameters of the GT sub-model.

Notation	Specification	Value	
		Pinion	Gear
m	Module (mm)	8	
α0	Pressure angle (°)	20	
β0	Helical angle (°)	0	
ha*	Addendum coefficient	1	1
cn*	Tip clearance coefficient	0.25	0.25
W	Face width (mm)	175	136
Zto	Number of teeth	23	120
Xn	Tooth profile shift coefficient	0.362	0.151

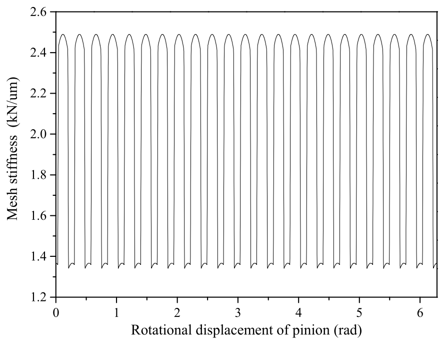


FIGURE 2. Time-varying mesh stiffness curve.

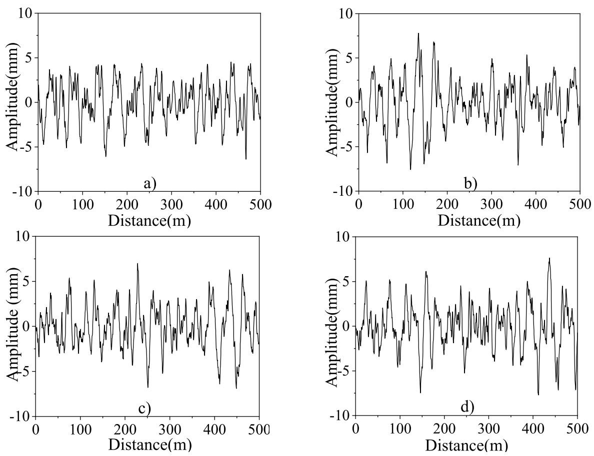


FIGURE 3. Track geometrical irregularity.

where K_{to}, K_{ff} and K_h indicate the tooth stiffness, the fillet foundation stiffness and the Hertz contact stiffness, respectively. The subscripts, p and g represent the pinion and the gear, respectively.

Based on the time-varying stiffness method mentioned above, combined with the gear pair parameters shown in Table 1 [20], the time-varying stiffness curve of the gear pair is shown in Fig.2.

B. TRACK GEOMETRICAL IRREGULARITY

The actual geometric contact state of the wheel-rail is not only related to the structural parameters of the vehicle, but also directly constrained by the line conditions. Therefore, the worse the line conditions are, the stronger the interaction between the wheel and rail will be, and the vibration of the locomotive-track coupled system will also intensify. The vibration of the wheel-rail interface can be further transmitted to the drive system. This paper adopts track geometrical irregularity as shown Fig.3.

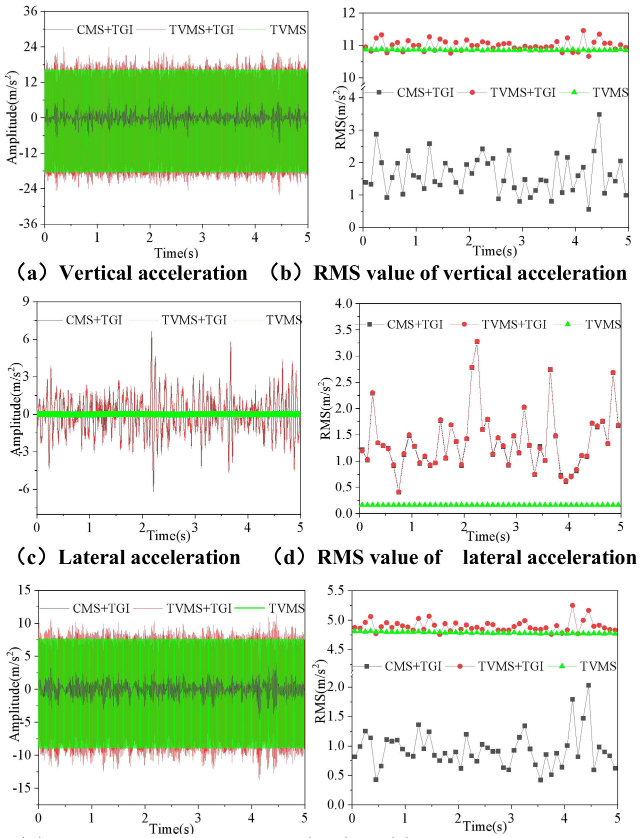
IV. DYNAMIC RESPONSE ANALYSIS OF DRIVE SYSTEM

Based on the established locomotive-track coupled model with drive system, the effect of internal and external motivation on the dynamic characteristics of locomotive drive system is investigated. In order to more intuitively express the impact of the track geometrical irregularity and time-varying mesh stiffness on dynamic response of locomotive drive system, the vibration responses of drive system were simulated under three kinds of combined excitations including the constant mesh stiffness and the track geometrical irregularity combination (CMS+TGI), the time-varying mesh stiffness and the track geometrical irregularity combination (TVMS+TGI), only the time-varying mesh stiffness (TVMS). In addition, we contrasted and analyzed the vibration responses of the drive system, the mesh force of the gear teeth and the dynamic transmission errors (DTE) of the gear pair during uniform and accelerated operation of the locomotive, and explored the effect of different excitations on the vibration characteristics of the drive system. Due to the uncoupling of the driving gear and traction motor, driven gear and wheelsets in the vertical direction, the vertical vibrations of the driving and driven gears are respectively attached to the traction motor and wheelsets, while the vibrations of the traction motor are more severe than that of the wheelsets. Therefore, this paper only considers the vibration responses of traction motor and the torsional vibration of gear pair when analyzing the vibration responses of the drive system.

A. UNIFORM SPRRD CONDITION

When the locomotive is running at 70 km/h and the track geometrical irregularity adopts AAR6, the dynamic responses of the locomotive drive system are simulated under different combinations of excitation during the uniform speed process.

The vibration responses of the traction motor are shown in Fig.4. As can be seen from Fig.4 (a), (c), and (e): when the combined excitation is CMS+TGI, the vertical, lateral and longitudinal acceleration of the traction motor fluctuate in the range of -0.8 g∼0.6 g, -0.6 g∼0.6 g and -0.4 g∼0.3 g; when the combined excitation is TVMS+TGI, the vertical, lateral and longitudinal acceleration of the traction motor fluctuate approximately between -2.6 g∼2.4 g, -0.6 g∼0.6 g, and -1.4 g∼1.1 g, respectively; when the combined excitation is TVMS, the vertical and longitudinal acceleration of the traction motor fluctuate between -1.8 g∼1.5 g and -0.9 g∼0.75 g, respectively. The amplitude of the lateral acceleration is very small. It is obvious that time-varying mesh stiffness affects the amplitude of



(a) Vertical acceleration (b) RMS value of vertical acceleration

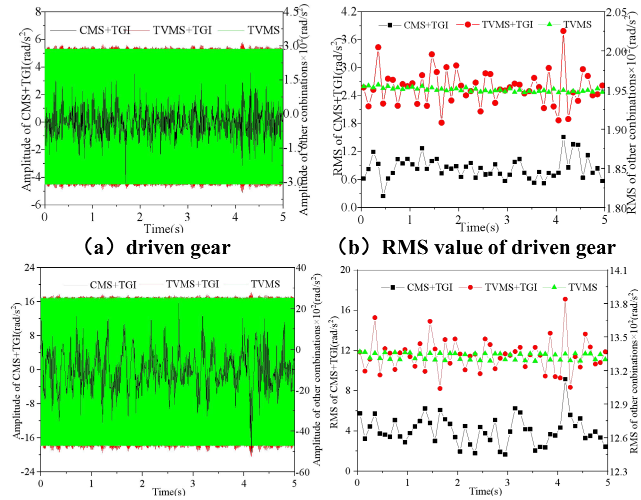
(c) Lateral acceleration (d) RMS value of lateral acceleration

(e) Longitudinal acceleration (LA) (f) RMS of LA

FIGURE 4. Vibration responses of traction motor (uniform speed).

the vertical and longitudinal vibration acceleration of the traction motor, while it has no effect on the lateral vibration of traction motor. The track geometric irregularity affects the vibration fluctuation of traction motor. The RMS values of motor vibration under three kinds of combined excitations shown in Fig.4 (b), (d) and (f). Fig.4 (b), (d) and (f) indicate that the fluctuation trend of motor vibration depends on the track irregularity. Time-varying mesh stiffness has no relation with the lateral vibration of the traction motor, but it directly affects the fluctuation of the vertical and longitudinal vibration of the traction motor. Therefore, it is necessary to consider the influence of time-varying mesh stiffness of gear pair when analyzing the vibration of locomotive traction motor.

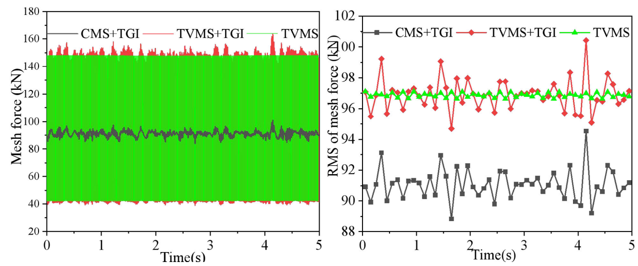
Fig.5 shows the torsional vibration response of the gear pair. From Fig.5, it can be seen that when considering time-varying mesh stiffness, the torsional vibration responses of the driving gear and driven gear are significantly intensified, and the fluctuations of the torsional vibration acceleration amplitudes of the driving and driven gear are related to track geometrical irregularity. The RMS values of the torsional vibration response of the driving and driven gear clearly show that the fluctuation trend of the gear pairs torsional vibration depends on the track geometrical irregularity, while the average value of the fluctuation is related to the time-varying mesh stiffness.



(a) driven gear (b) RMS value of driven gear

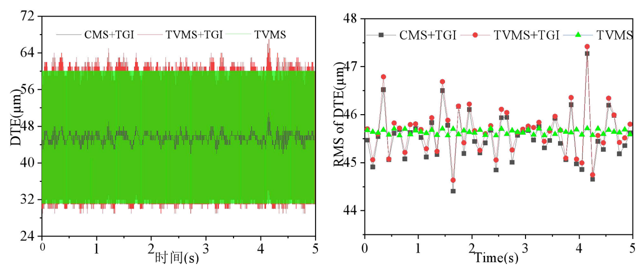
(c) driving gear (d) RMS value of driving gear

FIGURE 5. Torsional vibration responses of gear pair (uniform speed).



(a) Mesh force (b) RMS value of mesh force

FIGURE 6. Mesh force of gear teeth (uniform speed).



(a) DTE (b) RMS value of DTE

FIGURE 7. DTE of gear pair (uniform speed).

The mesh force and the DTE of the gear pairs are shown in Fig.6 and Fig.7, respectively. As can be seen from Fig.6 (a) and Fig.7 (a): when the combined excitation is CMS+TGI, the mesh force and the DTE of the gear pair fluctuate in the range of 84 kN∼101 kN and 42 μm∼51 μm, respectively; when the combined excitation is TVMS+TGI, the mesh force and the DTE of the gear pair fluctuate in the range of 39 kN∼164 kN and 29 μm∼67 μm, respectively; when the combined excitation is TVMS, the mesh force and DTE of the gear pair fluctuate in the range of 42 kN ∼ 147 kN and 31 μm∼60 μm, respectively. Compared with the mesh force and DTE calculated under CMS+TGI and TVMS combined excitation, the maximum mesh force calculated

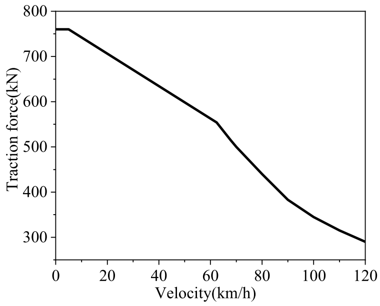


FIGURE 8. Locomotive traction characteristic curve.

under TVMS+TGI combined excitation increase about 62% and 12% respectively, and the maximum DTE of gear pair increases about 31% and 17% respectively. RMS values of mesh force and DTE under different combinations of excitation shown in Fig.6 (b) and Fig.7 (b). It can be seen from Fig.6 (b) and Fig.7 (b) that the fluctuation trend of gear pair mesh force and DTE depends on the change of track irregularity, and the mean value of fluctuation is related to time-varying mesh stiffness.

B. ACCELERATION CONDITION

When the locomotive starts and accelerates to 100 km/h,the track geometrical irregularity adopts AAR6. The locomotive traction characteristic curve is shown in Fig.8. The dynamic responses of locomotive drive system are simulated under different kinds of combined excitation during acceleration of the locomotive. The vibration responses of the traction motor and gear pairs are shown in Fig.8 ∼ Fig.11.

The vibration responses of the traction motor are shown in Fig.9 (a), (c) and (e). As can be seen from Fig.9 (a), (c) and (e): when the combined excitation is CMS + TGI, the vertical, lateral and longitudinal acceleration of the traction motor fluctuate in the range of −1.1 g∼1.1 g, −0.6 g∼0.6 g and −0.4 g∼0.5 g, respectively; when the combined excitation is TVMS+TGI, the vertical, lateral and longitudinal acceleration of the traction motor fluctuate in the range of −6.6 g∼3.9 g, −0.6 g∼0.6 g and −1.4 g∼1.5 g, respectively; when the combined excitation is TVMS, the vertical, lateral and longitudinal acceleration of the traction motor fluctuate in the range of −2.8 g∼1.9 g, −0.2 g∼0.2 g and −0.7 g∼1.2 g respectively. Vibration characteristics of the traction motor under acceleration are similar to those of traction motor under uniform speed. Time-varying mesh stiffness affects the amplitude of vertical and longitudinal vibration acceleration of traction motor, but has no effect on lateral vibration of traction motor. Meanwhile, it can be seen that resonance phenomenon of the traction motor will appear because the gear pair resonance caused by the parameters of the gear pair or sudden change in traction power when locomotive speed is about 9 km/h, 20 km/h or 60 km/h, and only the overall variation trend of locomotive longitudinal vibration is consistent with the variation trend of locomotive traction characteristic curve. During the whole acceleration process, the traction motor vibration is dominated by the time-varying mesh stiffness when the locomotive speed is less than 40km/h, while the traction motor vibration turns out to be more susceptible to the track geometrical irregularity when the locomotive speed is more than 40 km/h.

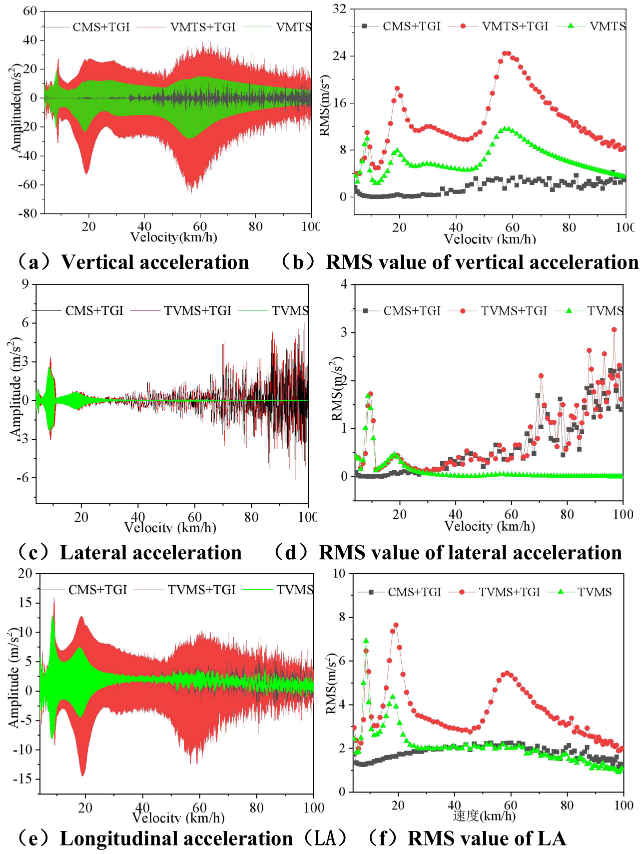


FIGURE 9. Vibration responses of traction motor (acceleration condition).

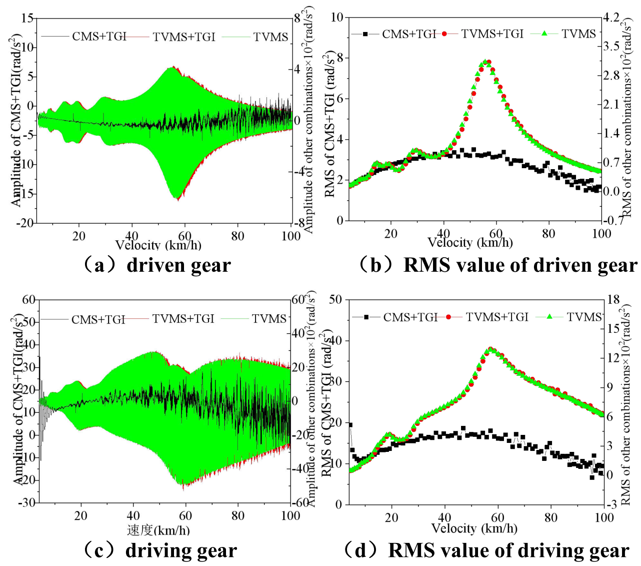


FIGURE 10. Torsional vibration responses of gear pair (acceleration condition).

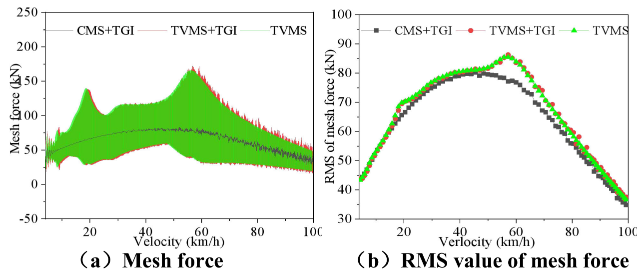


FIGURE 11. Mesh force of gear teeth (acceleration condition).

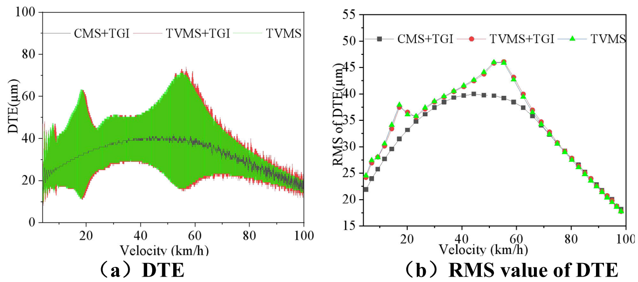


FIGURE 12. DTE of gear pair (acceleration condition).

In additional, when the locomotive is operating at a constant power stage (with a speed greater than 60 km/h), the vibration caused by time-varying mesh stiffness decreases as the locomotive speed increases. RMS values of the motor vibration under the three kinds of combined excitations are shown in Figs.9 (b), (d) and (f). They are also known that the time-varying mesh stiffness has no effect on the lateral vibration of the traction motor during the acceleration of the locomotive. In additional, the time-varying mesh stiffness dominates the vertical and longitudinal vibration of the traction motor in the low-speed operation stage, however, the track geometrical irregularity in high-speed operation has a greater impact on the traction motor vibration.

Fig.10 shows the torsional vibration response of the gear pair. From Fig.10 (a) and (c), it can be seen that when considering time-varying mesh stiffness, the torsional vibration responses of driving gear and driven gear are significantly intensified. Under different combined excitation, resonance phenomena of the driven gear and driving gear will appear, which are caused by the parameters of the gear pairs, when locomotive speed is less than 50 km/h, however resonance phenomena of the driven gear and driving gear have a short duration and low frequency when the combined excitation is CMS + TGI. When locomotive speed reaches at about 60 km/h, resonance phenomena of the driven gear and driving gear caused by sudden change in traction power occur, excluding CMS + TGI combined excitation. Similar results can be found in Fig.10 (b) and (d).

Mesh force is shown in Fig.11. As can be seen from Fig.11 (a): resonance phenomenon will appear for mesh force when the speed is about 9 km/h, 20 km/h and 60 km/h, and the overall variation trend of the mesh force is consistent with the variation trend of locomotive traction characteristic curve. During the whole acceleration process, the amplitude of mesh force is dominated by the time-varying mesh stiffness when the locomotive speed is less than 50km/h, while the amplitude of mesh force is more affected by the track geometrical irregularity when the locomotive speed is more than 50 km/h. RMS values of mesh force under different combinations of excitation shown in Fig.11(b). The time-varying mesh stiffness dominates mesh force in the low-speed operation stage, however, the track geometrical irregularity in high-speed operation has a greater impact on the traction motor vibration. DTE of gear pair is shown in Fig.12. DTE and mesh force have similar regularities.

V. CONCLUSION

Based on locomotive-track coupled spatial dynamics model considering drive system, the vibration responses of the gear transmission are simulated under three different combinations of excitation, namely constant mesh stiffness, time-varying mesh stiffness and track geometrical irregularity when locomotive operates in the uniform and accelerated conditions. By comparing the difference of vibration response, mesh force and dynamic transmission errors of gear pair of traction motor under different combinations of excitation, the influence of different excitation on the vibration characteristics of gear transmission is explored. The following conclusions are drawn:

(1) The time-varying mesh stiffness directly affects the vertical and longitudinal vibration of the traction motor.

(2) The time-varying mesh stiffness plays a leading role in the locomotive running at low speed, and the track geometrical irregularity plays a leading role in the high-speed stage.

(3) When the locomotive speed is about 9 km/h, 20 km/h or 60 km/h, resonance phenomenon of the traction motor and gear pair can be caused when considering time-varying mesh stiffness.

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TAO ZHANG received the B.S. and M.S. degrees in vehicle engineering from Lanzhou Jiaotong University, Lanzhou, China, and the Ph.D. degree in vehicle engineering from Southwest Jiaotong University, Chengdu, China. His research interests include vehicle-track coupled dynamics, rotor dynamics, and fault diagnosis.

XIAOHUI DANG received the B.S. degree in vehicle engineering from Lanzhou Jiaotong University, Lanzhou, China, where he is currently pursuing the master's degree in mechanical engineering. His research interest includes gear system dynamics.

YONG HE received the B.S. degree in vehicle engineering from Dalian Jiaotong University, Dalian, China, and the M.S. and Ph.D. degrees in vehicle engineering from Lanzhou Jiaotong University, Lanzhou, China. His research interests include maintenance decision, fault diagnosis, and reliability engineering.

JIANLONG XIE received the B.S. degree in vehicle engineering from Lanzhou Jiaotong University, where he is currently pursuing the master's degree in mechanical engineering. His research interest includes gear system dynamics.

HE MA received the B.S. degree in mechanical and electronic engineering from the Guangdong University of Technology, Guangzhou, China. He is currently pursuing the master's degree in mechanical engineering with Lanzhou Jiaotong University. His research interest includes rotor dynamics.