AMSE JOURNALS-2016-Series: Modelling A; Vol. 89; N°; pp 224- 243

Submitted July 2016; Revised Oct. 15, 2016, Accepted Dec. 10, 2016

Analyses on the Dynamic Characteristic of Power Coupling Mechanism During Engine Starting in Hybrid Electric Vehicle

Liao Lianying [1,3,a*], Zuo Yanyan [1,b], Zeng Xianren[2], Meng Haodong[3] and Yang Zhongkai[1]

[1] Institute of Noise and Vibration, Jiangsu University

China, 301 Xuefu Road, Zhenjiang, Jiangsu Province, 212013, (liaolianying@163.com)

[2] School of Mechanical & Material Engineering, Jiujiang University, China

[3] School of Mechanical & vehicle Engineering, Changzhou Institute of Technology, China

Abstract

In view of problem of obviously vibration and noise occurred during engine starting in a hybrid electric vehicle (HEV), the vibration analysis and test of its power coupling mechanism are carried out. After analyzing the structure of power coupling mechanism, the rigid-flexible dynamic model of the double planetary gear power coupling mechanism is developed based on the multi-body system dynamics theory. The multi-body system dynamics model is established under engine starting conditions using ADAMS software. The dynamic characteristics of power coupling mechanism are obtained through analyzing the time domain and frequency domain response curve of gears meshing force and bearings restraining force. At the same time, the HEV vibration and noise test is carried on under the working condition when engine is starting. Both the results of simulation and test demonstrate that the source of noise in power coupling mechanism is the engagement of gear pairs, and there is a severe shock during engine starting.

Keywords

hybrid electric vehicle, power coupling mechanism, dynamic characteristic, noise test

1. Introduction

Power transmission system of Hybrid Electric Vehicle (HEV) is a complex electromechanical system which is usually composed of engine, power coupling mechanism, motor, inverter and battery etc. Among all components, power coupling mechanism is a key component which couples engine and motor output power, and also the core part of HEV powertrain [1].

Among all types of power coupling mechanisms, planetary gear power coupling mechanism installed between the engine and the motor is most extensively applied in HEV. On the one hand, the power coupling mechanism with compact structure can efficiently couple the output power of engine and motor [2-3]; on the other hand, the working environment of dynamic coupling device is relatively complex. Power coupling mechanism is not just acted by steady load inputted from each power source, but also acted by transient impulse load in power switching process. Passengers are more sensitive to vibration impact caused by such unpredictable startup and stop, which thus influences drivability and ride comfort [4]. Moreover, if planetary gear mechanism with complex structure is not designed and manufactured rationally, severe vibration and noise will be generated.

For hot issues about dynamics of planetary gear power coupling mechanism, scholars have carried out relevant topic researches and gained certain achievements in recent years. Song et al.[5] combined the influence factors of planetary axis displacement and bearing stiffness on dynamics performance of coupling structure, and established a modified model of 2K-H planetary transmission system. Luo et al.[6] built a torsional dynamical model of two-stage planetary gear mechanism, and further analyzed its intrinsic characteristics and coupling stiffness. Zou et al.[7] set up a dynamical model of HEV transmission system and obtained intrinsic characteristics of the model through analytic method. In addition, other scholars also studied interior vibration and noise problem caused by HEV startup and stop[8-10]. However, few researches focus on the dynamic characteristics of power coupling mechanism which suffers transient impact under the operation conditions of engine startup and stop.

For this problem, this paper plans to study rigid-flexible coupling dynamical model of a HEV double-row power coupling mechanism. On this basis, simulation and experiment are combined to research and analyze dynamic characteristics of power coupling mechanism in accordance with engine startup conditions when the vehicle is motionless and operates under pure electric condition. The research result provides reference for low noise improvement design of power coupling mechanism.

2. Analysis of power coupling mechanism

To extend speed range of the motor, improve the output torque of power coupling mechanism and achieve more working modes, the number of planetary gearsets may increase, and planetary gearsets are connected in a certain way. A HEV power coupling mechanism adopts double planetary gearsets [11], and its structure is shown in Fig.1.

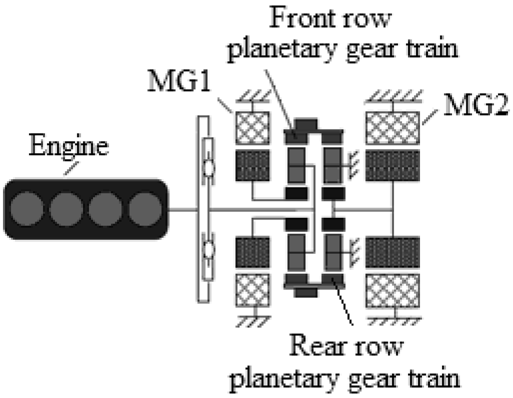


Fig.1 Double planetary gearsets power coupling system

It can be seen from Fig.1 that, this power coupling mechanism includes two planetary gearsets. The front row is power distribution planetary gear train, while the rear row is motor slowdown planetary gear train. The motor is connected with the front row of planet carrier through a torsion damper. The motor MG1 is connected with the front row of sun gear; the motor MG2 is connected with the rear row of sun gear; the rear row of planet carrier. The rear row of planet carrier is fixed on the external shell. The power outputted by the two rows of planetary gear trains is finally outputted by compound ring gear. In accordance with movement relation of planetary gear, the speed and torque of double planetary gearsets power coupling mechanism is described as

$$n_{out} = \frac{1+k_1}{k_1} n_e - \frac{1}{k_1} n_{M1} = \frac{1}{k_2} n_{M2} \tag{1}$$

$$T_{out} = T_e \cdot \frac{k_1}{1+k_1} + k_2 T_{M2} \tag{2}$$

where n_{out} is the output speed of ring gear; n_e is the engine speed; n_{M1} is the MG1 speed; n_{M2} is the MG2 speed; k_1 is the gear ratio of ring gear and the front row of sun gear; k_2 is the gear ratio of ring gear and the rear row of sun gear; T_{out} is the output torque of compound ring gear; T_e is the output torque of engine; T_{M2} is the output torque of MG2.

Since a row of slowdown planetary gear is added to the motor MG2, speed range of MG2 becomes wider, and the output torque of coupling mechanism is also larger. In addition, it is known from Formula (1) and Formula (2) that, engine speed can be regulated by MG1 and MG2. The engine can be maintained in the optimal working range all the time. The output torque of compound ring gear can be supplied by the engine and MG2 alone or jointly according to different working conditions of HEV.

Engine startup conditions of HEV include parking startup condition and EV mode startup condition [12]. For the former, under parking condition, the driving wheel of vehicle does not rotate, and compound ring gear does not move, either. If the motor MG2 stops, the motor MG1 as the starting motor will drive the sun gear to rotate and start up the engine through the planet carrier. For the latter, under EV mode, when the state of charge (SOC) drops to the set value, or when the vehicle needs large power, the engine starts up. After the engine starts up, output power is divided into two parts by power distribution planetary gear row. Some power flows to main retarder and drives the vehicle through the ring gear; the other power drives MG1 through the sun gear, and MG1 as the generator charges the battery.

3. Modeling of dynamic coupling system

3.1 Theoretical basis.

According to mechanical properties of bodies, the system can be classified into multi-rigid-body system, multi-flexible-body system and rigid-flexible coupling multi-body system. And the multi-flexible-body system is usually applied to analyze some mechanical systems with light weight, large volume or high-velocity motion. Rigid-flexible coupling multi-body system is the most common system.

Dynamical equation of multi-rigid-body system and dynamical equation of multi-flexible-body system is described as

$$\begin{Bmatrix} F(q,\dot{q},\ddot{q},\ddot{\lambda},t) \\ \psi(q,t) \end{Bmatrix} = 0 \tag{3}$$

$$M\ddot{\xi} + \dot{M}\dot{\xi} - \frac{1}{2}[\frac{\partial M}{\partial \xi}\dot{\xi}]^T\dot{\xi} + K\xi + f_g + D\dot{\xi} + [\frac{\partial \Psi}{\partial \xi}]^T\lambda = Q \tag{4}$$

where F is the generalized force; ψ is the constraint equation; q, $\dot{q}$, $\ddot{q}$ indicate the generalized coordinates and time derivative respectively; λ is the Lagrange multiplier array, ξ, $\dot{\xi}$, $\ddot{\xi}$ indicate the generalized coordinates and time derivative respectively; M, $\dot{M}$ indicate the mass matrix and time derivative respectively; $\frac{\partial M}{\partial \xi}$ is the partial derivative of mass matrix for generalized coordinates.

3.2 Solid modeling

UG is used to set up assembly model of HEV power coupling mechanism[13]. The core component of the power coupling mechanism is the fornt and rear double rows of helical gearsets.The parameters of the gear system are shown in Tab.1.

Tab.1 The parameters of planetary gear system

Parameters	Front row			Rear row		
	Sun gear	Planet gear	Ring gear	Sun gear	Planet gear	Ring gear
Number of teeth	30	23	78	22	18	58

Module (mm)	1	1	1	1.5	1.5	1.5
Normal pressure angle (°)	20	20	20	20	20	20
Derection of teeth	Left	Right	Right	Left	Right	Right
Helix angle (°)	25	25	25	30	30	30
Width of tooth (mm)	27.5	25	25	50	35	35
Normal addendum coefficient	1	1	1	1	1	1
Normal tip clearance coefficient	0.25	0.25	0.25	0.25	0.25	0.25
Tooth fillet radius(mm)	0.5	0.5	0.25			

The parts of helical planetary gear solid models are built based on the parameters given in Tab.1. The front row planetary gearset solid model is shown in Fig.2.

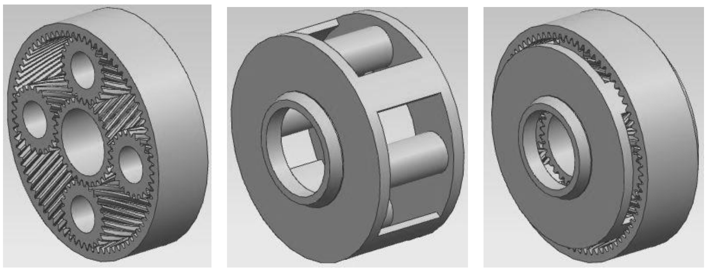


Fig.2 Front gearsets solid model

The rear row planetary gearset solid model is shown in Fig.3.

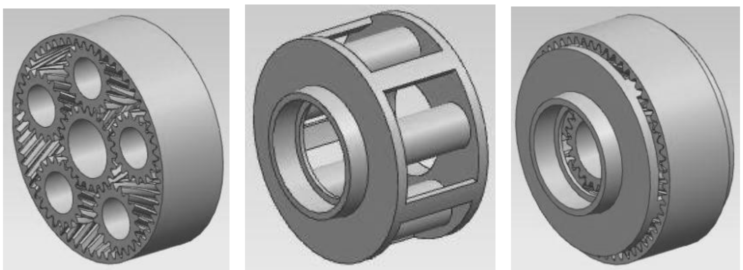
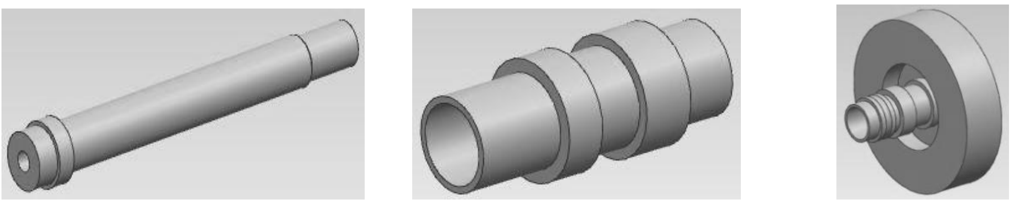
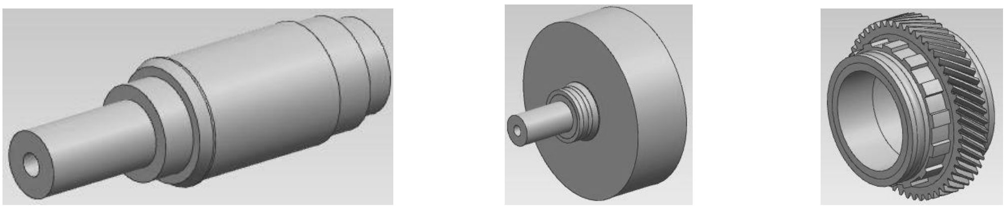


Fig.3 Rear gearsets solid model

Except the front and rear double rows gearsets, the power coupling system also includes the parts of the engine input shaft, the MG1 input shaft, the MG1 rotor, the MG2 input shaft, the MG2 rotor and compound ring gear. Based on the parameters, their solid models are built respective, as shown in Fig.4.



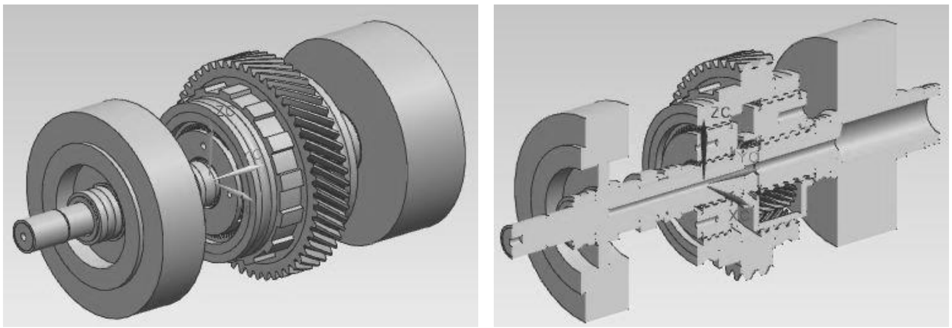
(a)Engine input shaft (b)MG1input shaft (c) MG1 rotor



(d)MG1input shaft (e) MG1 rotor (f)compound ring gear

Fig.4 Other parts solid models

After building the parts of power coupling mechanism, the assembly model of the power coupling system is built based on the assembly function of UG software. The whole assembly model and sectional drawing of the power coupling system is shown in Fig.5.



(a)assembly model (b)sectional drawing

Fig.5 Power coupling system assembly model and sectional drawing

3.3 Multi body dynamics modeling.

After the solid model format is transformed, the model is imported in ADAMS, where each component is mutually independent [14].

Definition of material property is conducted for each component of power coupling mechanism, as shown in Tab.2.

Tab.2 Material properties of power coupling mechanism

Parts	Material	Elasticity modulus (N/m2)	Poisson ratio	Density (kg/m3)
Small bearing	GCr15	2.19e+11	0.3	7.83e+3
Large bearing	GCr15SiMn	2.16e+11	0.3	7.82e+3
Motor rotor	Stalloy	1.97e+11	0.26	7.65e+3
Motor input shaft	40Cr Steel	2.11e+11	0.277	7.87+3
Gears	42CrMo	2.1e+11	0.28	7.85e+3
Engine input shaft	20CrMnTi	2.12e+11	0.289	7.86e+3

Each component is connected together through imposing restrictions. Relative to the ground, engine input shaft, MG1 rotor, MG2 rotor, compound ring gear are revolutes. Besides, relative to the front and rear rows of planet carriers, the front and rear rows of planet gears are also revolutes. Except gear engagement, fixed joints are among other components. Contact force F among gear pairs is defined

$$F = \begin{cases} kx^e + F_s(x,0,0,d_{\max},C_{\max})\dot{x}; & x < 0 \\ 0; & x \geq 0 \end{cases} \tag{5}$$

where K represents contact stiffness coefficient, x is the distance variable between two objects, e is the nonlinear contact force exponent, F_s refers to Step function, d_{max} is maximum breakdown depth, C_{max} is the maximum damping coefficient.

The crankshaft of the engine is connected with the output shaft of the engine through a torsion shock absorber, which is needed to add in the ADAMS. The stiffness coefficient K_d of the torsion shock absorber is 3.93e+5N·mm/deg. The damping coefficient C_d is 23.33N·mm·s/deg. A certain preload is allied to the damping springs in the torsional damper. The value of preload T_n is 14200N·mm.

For this HEV power coupling mechanism, its operation conditions are complex and varied. The engine, the motor MG1 and the motor MG2 should frequently start up and stop according to working conditions. The maximum speed of MG1 and MG2 is over 10000rpm. This inevitably generates large impulse excitation to each part in power coupling mechanism. For such complex mechanical system with high-speed operation, flexible treatment is usually carried out for the parts of which elastic deformation may easily happen in the system. Dynamical model of rigid-flexible coupling multi-body system is established in order to improve computational accuracy. Flexible treatment is conducted for engine input shaft and input shaft of MG1 and MG2 in dynamic coupling system, and other transmission parts are set to the rigid body.

Through analyzing modal frequency and vibration mode of engine input shaft and input shaft of MG1 and MG2, it is found that high-order mode which infleunces calculation result little loses efficacy. After rigid body is repalced by fleixble body, rigid-flexible coupling dynamical model is set up, as shown in Fig.6.

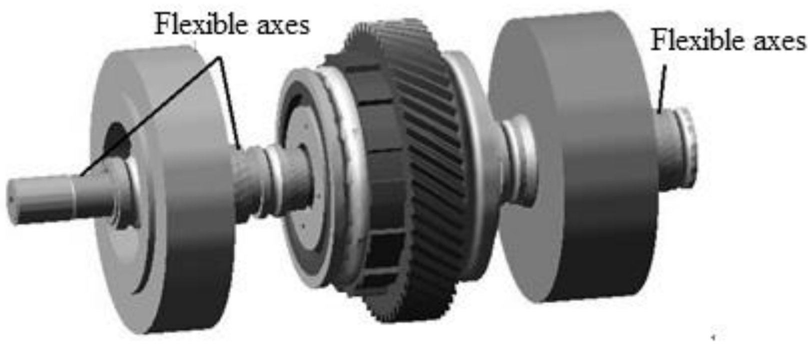


Fig.6 The rigid-flexible coupling dynamics model

4. Analysis of vibration characteristics of power coupling mechanism

4.1Acting loads and drives

Dynamic characteristics of power coupling mechanism under engine startup condition can be analyzed through imposing load and drive on rigid-flexible coupling dynamical model. For this HEV, its operation condition is very complex, and the drive and load under each working condition are different. To gain speed and torque data of HEV engine, MG1 and MG2 under engine startup condition, HEV detector is adopted for road detection test. The speed and torque data of each power component are shown in Table 3.

Tab.3 Testing data of each power element under engine starting conditions

Operation condition	v (km/h)	n_{Eng} (rpm)	n_{MG2} (rpm)	T_{MG2} (Nm)	n_{MG1} (rpm)	T_{MG1} (Nm)	SOC (%)	Gas peddle (%)
Engine startup under parking condition	0	0~325	0	0	0~1170	0~18.5	40	0
Engine startup under pure electric operation	30	0~600	1500	25~8	-1480~ 680	0~27.6	47	0~100

According to the measured data, the driving force is acted on dynamic coupling mechanism under engine startup condition. The power coupling mechanism finally transmits power to vehicle wheels through compound ring gear. Thus, it is required to act corresponding load torque on the compound ring gear according to working conditions. Based on driving equation (Formula 6) of HEV, the load torque on the compound ring gear can be figured out according to vehicle speed and known parameters under different working conditions.

$$\frac{T_v \cdot i_g \cdot i_0 \cdot \eta_t}{r} = mgf\cos\alpha + \frac{C_d \cdot A \cdot V_a^2}{21.15} + mg\sin\alpha + \delta \cdot m \cdot \frac{dV}{dt} \tag{6}$$

Where, T_V is the load torque on compound ring gear of power coupling mechanism, i_g is the gear ratio, i_0 is the final drive ratio, η_t is the gear transmission efficiency , r is the radius of driving wheel , m is the gross mass of vehicle, g is the gravitational acceleration, f is the rolling resistance coefficient, α is the road surface gradient, C_d is the air resistance coefficient, A is the windward area, V_a is the vehicle speed, δ is the vehicle mass conversion ratio, dV/dt is the vehicle acceleration.

4.2 Selection of simulation observation points

Dynamic excitations generated by gear transmission system in the transmission process can be classified into two types: internal excitation and external excitation [15]. In accordance with dynamic excitation characteristics of transmission, measurement points are chosen on multi-body dynamical model to study dynamic characteristics of each component under engine startup condition.

Vibration noise characteristics of dynamic coupling system are closely related to engaging transmission of two planetary gearsets. Hence, the engaging force of gears are first selected as the observation targets.

External vibration excitation of dynamic coupling system is ultimately transmitted to the bearing through a compound device, and the bearing transmits excitation to external shell so as to trigger shell vibration. Thus, it is still necessary to select mass center of compound device as the observation point to study its constraining force characteristic on the bearing.

4.3 Dynamic characteristics analysis under engine startup condition

Under parking condition, after the vehicle starts up, if SOC or engine temperature is too low, MG1 as the starting motor will drive the sun gear to rotate and drive the engine through the planet carrier. At this moment, the driving wheel, compound ring gear, MG2 and the rear row of planetary gearsets are all stationary. This process is transient acceleration process. According to testing data in Table 2, the speed n_{MG1}=Step (time, 0,0,0.25,-7020d) is acted on MG1 to drive. Load moment T_{Eng}=34000 N•mm on the input shaft is figured out according to rotational inertia of engine. The simulation time is 0.25s, and the number of simulation steps is 500. Engaging force characteristics of front row of sun gear and planet gear, and engaging force characteristics of planet gear and ring gear are shown in Fig.7 and Fig.8 respectively.

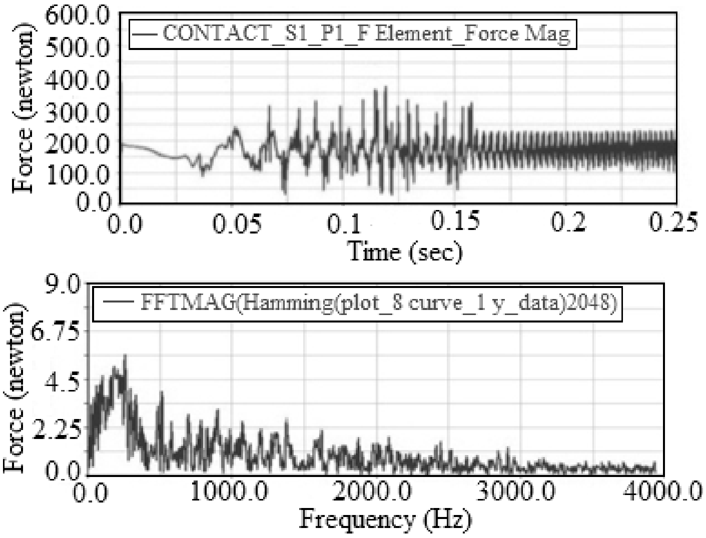


Fig.7 Time and frequency response of sun gear and planetary gear meshing force of front row

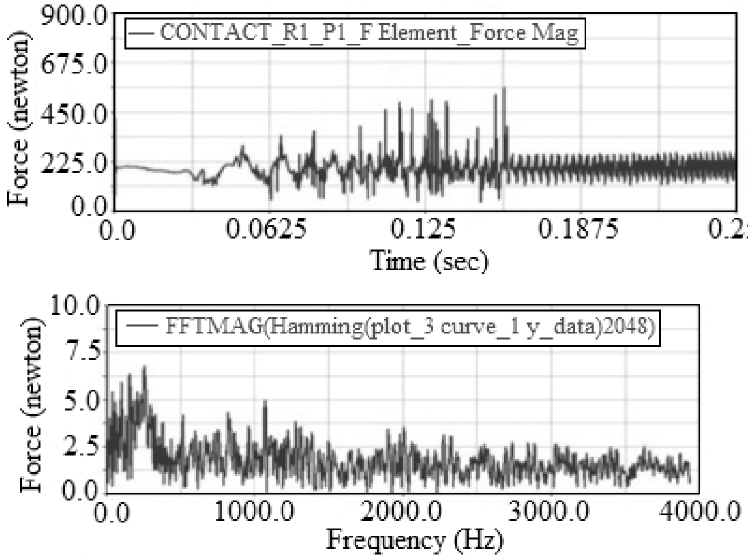


Fig.8 Time and frequency response of planetary gear and ring gear meshing force of front row

It is known from engaging force response curves in Fig.3 and Fig.4 that there are engaging forces in the front row has many transient impulse excitations in the first half time of engine startup. As the engine starts up, engaging force gradually tends to be stable. According to frequency domain response curve, no prominent peak value appears, but the amplitude value in the low-frequency band is large.

In the pure electric operation, the vehicle runs at a constant speed (v=30km/h) under pure electric mode. When SOC drops to the set value or the accelerator pedal is stepped on, the engine will start up quickly under the drive of MG1. MG1 transforms to positive rotation from reverse immediately. According to testing results in Table 2, the speed n_{MG2}=135d/s is acted on MG2 to drive. The speed n_{MG1}= Step(time,0.05,13314d,0.25,-3967.8d) is acted on MG1 to drive. This means before 0.05s, the vehicle is under pure electric condition, and the speed of MG1 is a constant value 13314d/s. From 0.05s to 0.25s, the vehicle is under engine startup state, and the speed of MG1 changes to -3967.8d/s from 13314d/s. Since engine startup process is very transitory and vehicle speed changes little, load of compound ring gear remains unchanged. According to Formula (6), the torque on compound ring gear T_V is 16853N•mm. Moreover, to overcome rotational inertia of engine, load T_{Eng}=34000 N•mm should be acted on the engine shaft. The simulation time is 0.25s, and the step number is 500. Engaging force characteristics of front and rear rows of planetary gearsets are shown in Fig.9, Fig.10, Fig.11 and Fig.12 respectively.

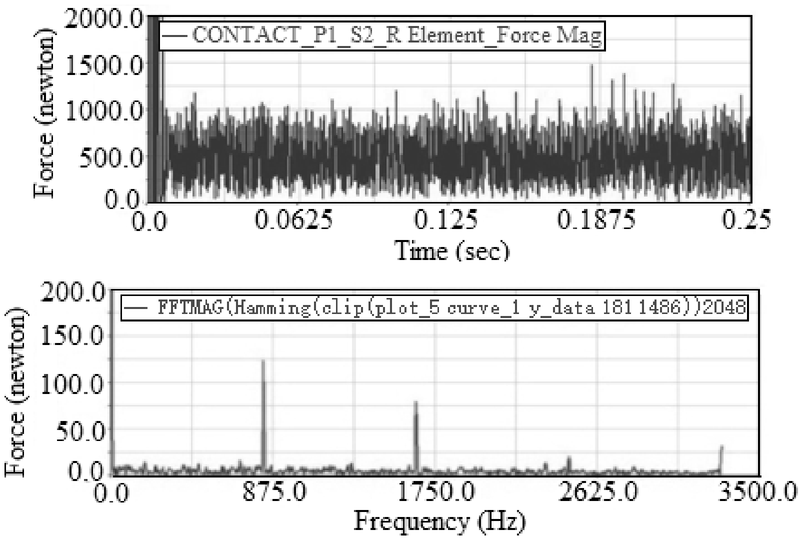


Fig.9 Time and frequency response of sun gear and planetary gear meshing force of rear row

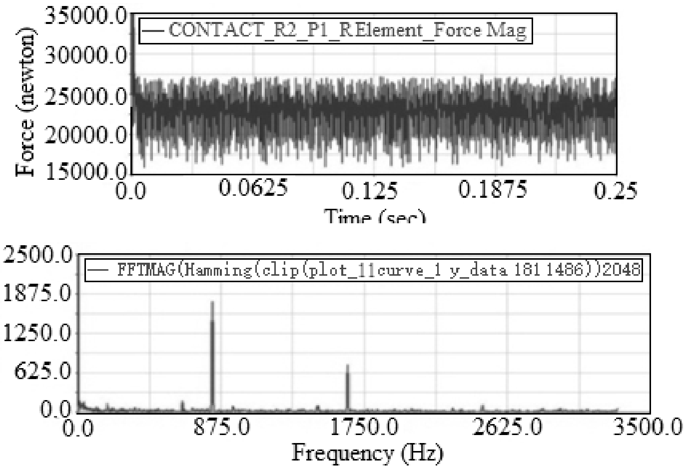


Fig.10 Time and frequency response of planetary gear and ring gear meshing force of rear row

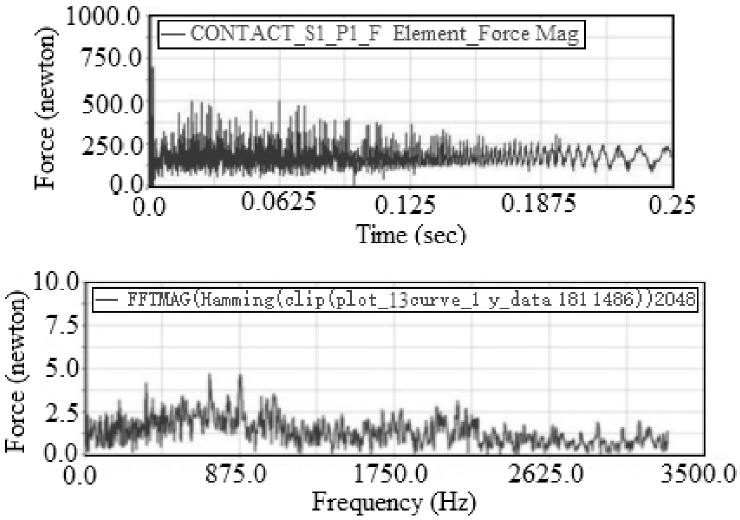


Fig.11 Time and frequency response of sun gear and planetary gear meshing force of front row

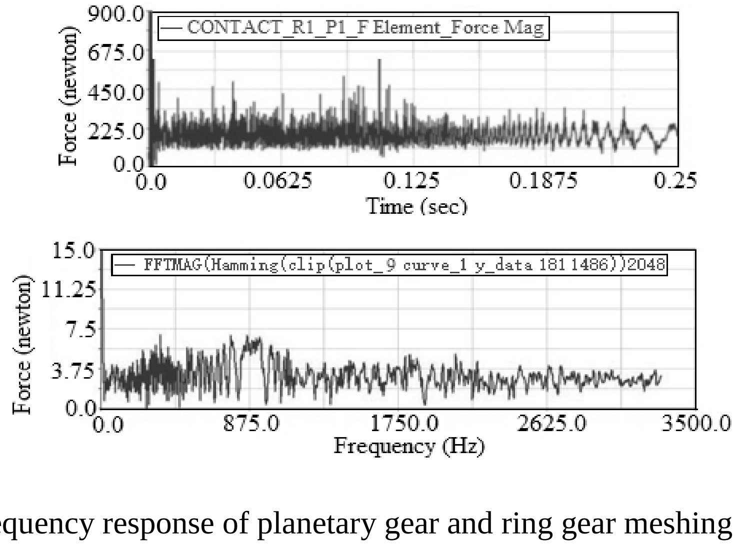


Fig.12Time and frequency response of planetary gear and ring gear meshing force of front row

According to the time domain and frequency domain response curve of engaging force, it is known that when the engine stars up under pure electric condition, engaging force characteristics in the rear row is not influenced seriously. Prominent peak values still exist at engaging frequency at 825Hz and twice of engaging frequency at 1650Hz. Since the speed of MG1 and engine changes, engaging force characteristics in the front row change greatly. At the beginning of engine startup, engaging force signal in the front row is not very steady. At the moment of engine startup, large transient impact exists, and there is no prominent peak value in frequency domain characteristics.

5. Vibration test and analysis

The test is conducted under two working conditions: HEV parking mode startup, EV mode startup. Acceleration sensors are arranged at the right front suspension of engine and the shell of power coupling mechanism respectively. A microphone is arranged in the engine compartment, as shown in Fig.13.

Fig.13 Arrangement of measuring points

LMS test equipment is used to gather dynamic parameters of system under engine startup condition. Fig.14 shows dynamic response curve of system under HEV parking mode engine startup condition.

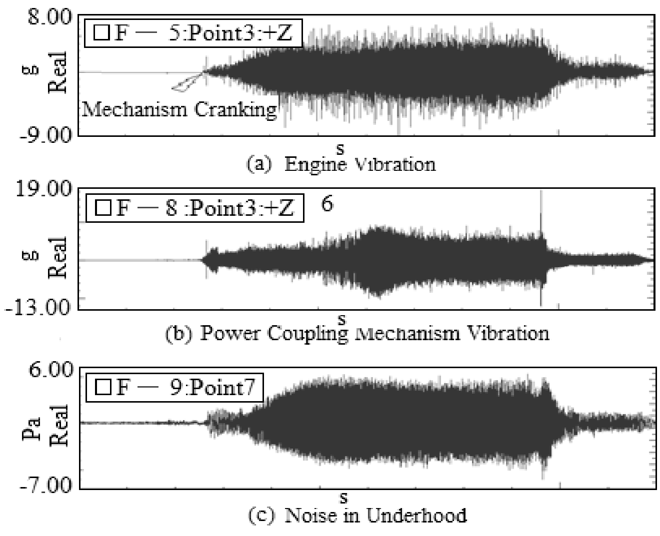


Fig.14 Time analysis of signal during engine starting in the parking condition

It can be seen from Fig.14 that, large impacts exist at the moment when MG1 drives the engine to rotate. When engine speed rises to 1100rpm, the engine starts to inject fuel and ignite. Then the vibration at the engine is intensified, and the noise in underhood is enhanced. But, the vibration at power coupling mechanism is relatively steady and smooth. This result agreed with the simulation result under engine startup condition.

After the vehicle operates for a time at a low speed under pure electric mode, dynamic response curve of system measured under sudden engine startup condition is shown in Fig.15.

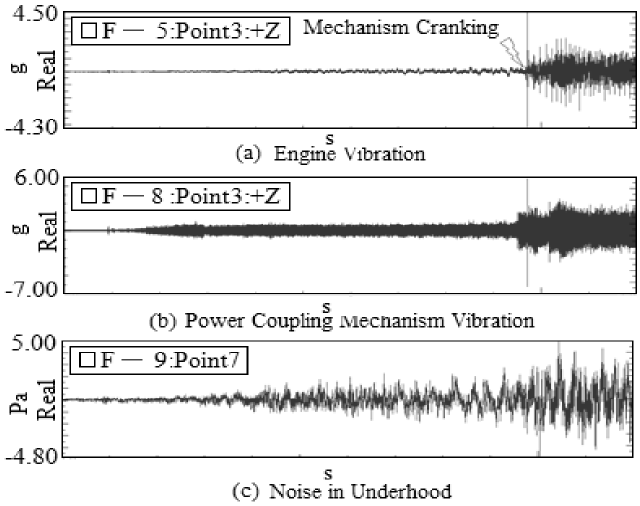


Fig.15 Time analysis of signal during engine starting under EV mode

Based on Fig.15, the vibration at power coupling mechanism is weak, and the noise level in the underhood is low when the vehicle operates under pure electric mode. The vibration at power coupling mechanism intensified suddenly at about 0.2s before MG1 drives the engine to rotate. Large transient impact exists at the moment when the engine is driven to rotate. Then the vibration tends to be steady.

Discussion

This paper mainly focuses on the study of the HEV power coupling mechanism dynamic characteristic analysis under the working condition when the engine is starting. The solid model of power coupling mechanism is established based on UG software. The dynamic characteristics of power coupling mechanism are analyzed based on ADMAS software. After that, the HEV vibration and noise test is carried on. And the results of test match with the results of simulation. Currently, a great deal of researches pays close attention to the power coupling mechanism dynamic characteristic analysis.

Hwang’s re research simulates the dynamic behaviors associated with the neutral starting and stopping of a power-split hybrid vehicle. His research results reveal that unsmooth engine start transitions can cause driveline vibrations, making the ride uncomfortable and the customer dissatisfied with the vehicle.[8] Yang et al established the 3D solid model of power coupling mechanism with two planetary gear rows and analyzed the natural frequencies and natural modes of vibration of the system, The results reveal that the vibrations of both helical planetary gear trains are mainly planet gear torsional vibration or twist vibration. [16]Liu et al performed and analyzed experiments on the vibrations of the driver’s seat track to evaluate quantitatively the vibrations during the engine start. The measurement results show that vibrations in the longitudinal direction are the most severe. [17]

These results are very similar to ours’, and indicate that the source of noise in power coupling mechanism is the engagement of gear pairs. An argument just like the suggestion pointed out by our paper is given that, transient impact exists in power coupling mechanism at the moment during the engine starting up. The driver and passengers have no preparation and precognition for the engine start. Therefore, they are more sensitive to the accompanying vibration which will affect the ride comfort and NVH performance.

This paper has many contributions and advantages. It illuminated how to build a solid model of power coupling mechanism used in HEV. It also presented a rigid-flexible dynamic model and method for analyzing the dynamic characteristics of power coupling mechanism under the special working condition when the engine starting up. The principle of this method is clear and is simple for calculation. So it can be used widely and can provide a useful tool for dynamic characteristics of HEV power coupling mechanism in analysis. From the perspective of practice, the method proposed here is of important value for… Facing the shortage of impetus and uncertainty in prospects of HEV development, the main idea of this paper is timing, and maybe can help to broaden the possibility of HEV development.

Of course, there are still some aspects that need to be improved. The vibration and noise test is carried on the vehicle instead of on the platform. So the vibration and noise signals gain from the test are not only cause by the power coupling mechanism but also cause by other compants. That makes the signals complex. For the reasons, frequency domain response of the vibration and nosise in the test are not analyzed. In addition, antivibration control strategies are not studied in this paper. These are also the main issues requiring in-depth study in the future.

Conclusion

Gear engagement is a major excitation source of power coupling mechanism noise. Transient vibration impact is severe at the beginning of engine startup. Characteristic frequency of peak value is the engaging frequency of two planetary gearsets and its multiple frequencies.

Large transient impact performance exit in power coupling mechanism under engine startup condition based on parking condition. After the engine injects fuel and ignites, the power coupling mechanism appears relatively stable vibration performance.

Under engine startup condition based on pure electric mode, dynamic characteristics of front row of planetary gear train are not coupled with dynamic characteristics of the rear row of planetary gear train and compound ring gear. Power coupling mechanism can switch working mode in a smooth and steady way.

The combination of simulation and experiment method can effectively acquire dynamic characteristics of double-planet-gearsets power coupling mechanism under both steady state and non-steady state.

Acknowledgements

This work was financially supported by National Natural Science Foundation of China (51575238), the Jiangsu Colleges and Universities Natural Science Research Project

(15KJB580001) and Science and Technology Program of Jiangxi Province in China (2011BBE50011).

References

1. ML Kuang, An Investigation of Engine Start-Stop NVH in A Power Split Powertrain Hybrid Electric Vehicle, 2006, SAE Technical Paper, 2006-01-1500.
2. K Yu, M Mukai, T Kawabe, Model predictive control of a power-split hybrid electric vehicle system, 2012, Artif Life & Robotics, vol.17, no.2, pp.221-226.
3. L Humbert, P Pellerey, S Cristaudo, Electromagnetic and Structural Coupled Simulation to Investigate NVH Behavior of an Electrical Automotive Powertrain, 2012, SAE Technical Paper, 2012-01-1523.
4. D Liu, H Yu, J Zhang, Multibody dynamics analysis for the coupled vibration of a power split hybrid electric vehicle during the engine start transition, 2015, Proceedings of the Institution of Mechanical Engineers Part K Journal of Multi-body Dynamics, vol.0, no.0, pp.1-14.
5. Y Song, W Xu, C Zhang, S Wang, Modified torsional model development and natural characteristics analysis of 2K-H epicyclic gearing, 2006, Journal of Mechanical Engineering, vol.42, no.5, pp.16-21.
6. Y Luo, Y Chen, Dynamic Modeling and Analysis of Power Coupling System with Two-stage Planetary Gear Trains for hybrid System, 2012, Journal of Mechanical Engineering, vol.48, no.5, pp.70-75.
7. L Zou, X Tang, H Yu, J Zhang, Analyses on the Torsional Vibration and Noise of Transmission System in Hybrid Electric Cars, 2014, Automotive Engineering, vol.36, no.6, pp.709-714.
8. HY Hwang, Minimizing seat track vibration that is caused by the automatic start/stop of an engine in a power-split hybrid electric vehicle, 2013, Journal of Vibration & Acoustics, vol.135, no.6, pp.2627-2637.
9. DD Kum, HH Peng, Control of engine-starts for optimal drivability of parallel hybrid electric vehicles, 2013, Journal of Dynamic Systems Measurement & Control, vol.135, no.2, pp.450-472.

10. JS Chen, HY Hwang, Engine automatic start–stop dynamic analysis and vibration reduction for a two-mode hybrid vehicle, 2013, Proceedings of the Institution of Mechanical Engineers Part D Journal of Automobile Engineering, vol.227, no.9, pp.1303-1312.
11. TA Burress, SL Campbell, CL Coomer, Evaluation of the 2010 Toyota Prius Hybrid Synergy Drive System, 2010, Oak Ridge National Laboratory, publish date March 2011.
12. H Sugimura, Development of HEV Engine Start-Shock Prediction Technique Combining Motor Generator System Control and Multi-Body Dynamics (MBD) Models,2013, SAE Technical Paper, 2013-01-2007
13. F Bi, X Cui, N Liu, Computer Simulation for Dynamic Meshing Force of Involute Gears, 2005, Journal of Tianjin University, vol.38, no.11, pp.991-995.
13. R Meng, C Zhang, J Wang, T Li, Q Cui, UG8.0 in the Simple Application of Automatic Shoes-Washing Machine Design, 2016, Mathematical Modelling of Engineering Problems, vol.3, no.2, pp57-62
14. L Piancastelli, L Frizziero, I Rocchi, An innovative method to speed up the finite element analysis of critical engine component, 2012, International Journal of Heat & Technology, vol.30, no.2, pp.127-132
15. T Li, R Zhu, H Bao, Nonlinear Torsional Vibration Modeling and Bifurcation Characteristic Study of a Planetary Gear Train, 2011, Journal of Mechanical Engineering, vol.47, no.21, pp.76-83.
16. Z Yang, L Liao, Y Zuo, M Ni, Dynamic Modal of Power Coupling Planetary Gear System for Hybrid Electric Vehicle, 2015, Journal of Henan University of Science & Technology, vol.36, no.2, pp.14-17.
17. D Liu, J Zhang, D Zhang, G Liu, H Yu, Experimental and numerical analysis of the seat track vibrations caused by engine starts in a power-split hybrid electric vehicle, 2016, Proceedings of the Institution of Mechanical Engineers Part D Journal of Automobile Engineering, vol.230, no.3.