EVS27
Barcelona, Spain, November 17-20, 2013

Failure Mechanism of the Transmission Shaft of a New Power Split Hybrid Vehicle

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Abstract

Power split driveline ensures a hybrid vehicle good performance in fuel economy and emissions, but might bring about serious torsional vibrations at city traffic operation. A power split driveline is addressed in this paper to understand the mechanism behind the failure of its transmission shaft. Lumped-parameter models were established for the system at various engine and motor configurations corresponding to typical working conditions. Natural frequencies and mode shapes were calculated for each configuration. Sensitivity analysis reveals the variation of the natural frequencies with respect to the inertia or stiffness of the system. It is revealed that there exists unavoidable resonance with the system during vehicle operation, which should be the main reason of the shaft failure.

Keywords: hybrid electric vehicle, powertrain, modelling, city traffic

1 Introduction

Power split hybrid vehicles make use of two or more power plants in the interests of improving the power output efficiency and fuel economy. However it may bring about a new torsional vibration problem at city traffic operations because of frequent starts and stops of ICE for seeking high fuel efficiencies. This kind of harmful vibration degrades ride comfort and even causes key component failure. In respect of traditional petrol vehicle, studies about reducing torsional vibration have been reported early and extensively. A linear torsional vibration model of an automotive driveline has been established and the steady state responses have been calculated in [1]. Dynamic behaviours of total vehicle powertrain have been analyzed in time domain considering some nonlinear effects such as gear backlash and nonlinear clutch behaviour [2]. Recently, there has been a trend towards developing torque control method for reducing vibration and improving ride comfort. Optimal algorithms and several advanced control strategies have been developed to control driveline vibrations [3-5]. *Kamran A. Gul* [6] has significantly reduced the torsional vibration amplitude by modifying inertia and stiffness properties of the driveline based on sensitivity analysis.

As for an electrical vehicle, advanced control methods applied to the motor and/or the generator can also reduce vibration dramatically [7-9], referring to the good performances of electrical motors.

Similar to the power train in a petrol-powered vehicle, the torsional vibration suffered by a hybrid driveline is also due to the torque fluctuation of the internal combustion engine, as the electrical motor has little torque ripples. Due to various power combinations, the torsional vibration in a hybrid driveline is more complex, asking for more endeavour with the system design. Otherwise the components might suffer from failures. Figure 1 presents a broken shaft in the clutch of a hybrid driveline.

Figure 1: A broken shaft in the clutch

According to the authors' knowledge, there is no publication to systematically address driveline failure in a hybrid or pure electric vehicle. With this observation and motivated by a series of actual failure problems, this paper tries to explore some possible mechanism behind the shaft failure in a hybrid driveline. A lumped-parameter model is derived firstly for the driveline. Natural frequencies and corresponding mode shapes are respectively calculated for the driveline under different operation conditions. Based on the simulations, discussions demonstrate that there exist several resonances when ICE starts/stops. During actual city traffic driving cycles, the resonances occur repeatedly as the ICE frequently starts/stops, which eventually leads to the shaft failure. Sensitivity analysis is finally carried out for a possible solution to avoid resonance.

2 Driveline modelling

A lumped parameter model is employed to describe the torsional vibration of a general driveline as shown in Figure 2.

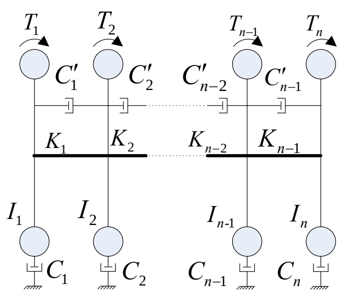


Figure 2: Schematics of a driveline with n inertia

Assuming that there are n lumped inertias in the driveline, each of which is subjected to external torque. Correspondingly there are n-1 torsional springs and n-1 relative and absolute torsional dampers. Notating the inertias and the torques by I_i and T_i with $i=1,2,\cdots,n$, while the stiffness rates and two sets damping coefficients are assigned as K_j and C_j' C_j with $j=1,2,\cdots,n-1$. Using the Newtonian second law, the equations of motion of the driveline can be derived as

$$\mathbf{I}\ddot{\boldsymbol{\theta}}+\mathbf{C}\dot{\boldsymbol{\theta}}+\mathbf{K}\boldsymbol{\theta}=\mathbf{T}\,,\quad \boldsymbol{\theta}=[\theta_1 \quad \theta_2 \quad \cdots \quad \theta_n]^{\mathrm{T}} \tag{1}$$

Where, θ_i is the angle fluctuation of inertia i with respect to the steady rigid rotation, while $\mathbf{I},\mathbf{C}$ and $\mathbf{K}$ are system matrices, namely inertia matrix, damping matrix and stiffness matrix, and respectively calculated by

$$\mathbf{I}=\begin{bmatrix} I_1 & & & & \\ & I_2 & & & \\ & & \ddots & & \\ & & & I_{n-1} & \\ & & & & I_n \end{bmatrix} \tag{2}$$

$$\mathbf{K}=\begin{bmatrix} K_1 & -K_1 & & & \\ -K_1 & K_1+K_2 & -K_2 & & \\ & -K_2 & \ddots & \ddots & \\ & & \ddots & K_{n-1}+K_n & -K_n \\ & & & -K_n & K_n \end{bmatrix} \tag{3}$$

$$\mathbf{C}=\begin{bmatrix} C_1+C_1' & -C_1' & & \\ -C_1' & C_2+C_1'+C_2' & -C_2' & \\ & -C_2' & \ddots & -C_n' \\ & & -C_n' & C_n+C_n' \end{bmatrix} \tag{4}$$

The externally applied torques construct a vector as $\mathbf{T}=[T_1 \quad T_2 \quad \cdots \quad T_{n-1} \quad T_n]^{\mathrm{T}}$.

The natural frequencies and corresponding mode shapes of the driveline can be obtained free vibration analysis. Although the damping ratio of a torsional damper is quite big, the damping in the system is still neglected at this early stage, which results in a conservative predictions. By getting rid of the damping matrix and torque vector from Eq.(1), we have

$$\mathbf{I}\ddot{\boldsymbol{\theta}}+\mathbf{K}\boldsymbol{\theta}=\mathbf{0} \tag{5}$$

The natural frequency and normal modes can be obtained by solving the following generalized eigenvalue problem

$$\mathbf{K}\psi = \lambda \mathbf{I}\psi \tag{6}$$

where $\lambda = \omega^2$ is the eigenvalue and ψ is the eigenvector. As both the inertia and stiffness matrices are symmetric, the eigenpairs can be readily calculated in Matlab.

3 The targeted driveline

3.1 Configurations and running modes

Figure 3 shows the target hybrid driveline, which consists of an ICE, a flywheel, a torsional damper, an ISG, a clutch, the main motor, a propeller shaft, a differential and rear shaft assembly, and the tires.

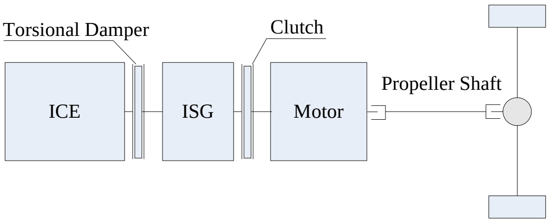


Figure 3: Driveline of a hybrid vehicle

Depending on the vehicle status, the driveline appears at various combinations of the internal combustion engine, the main motor and the ISG means different system configurations.
Among various power combinations of the driveline, two ones are of the most importance after careful investigation.

- Electric-only with charge-depleting and charge-sustaining of energy management that is pure electric vehicle mode only operates at no more than 20km/h. The battery is gradually depleted because the vehicle runs on power exclusively from the battery. Once the battery is depleted to a certain minimum, ISG drives the ICE as a starter and then ICE drives the ISG as a generator which powers the electric motor in a series configuration.
- Hybrid drive with ICE engaged in high efficiency mode operates at more than 20km/h. Both the ICE and the electric motor are involved to provide traction.

Operating on all-electric drive mode is at low speed and the clutch is disengaged. Only the main motor works to drive the vehicle, which can be called pure electric drive mode. The necessary of the motor comes from the batteries. At this status, the clutch is disengaged. As the broken shaft lies between the ICE and ISG, the pure electric drive mode has no contribution to the shaft failure. Free vibration will thus not be performed on the configuration of this mode. At the start mode, the ISG runs at first to drive the ICE. When the speed reaches to a prescribed speed, the ICE works and takes over control from the ISG, the vehicle enters a normal running stage. During the start and stop modes, there are dramatic variation within the ICE status including ignition or cut-off, which presents excitation to the driveline. As a result, the configurations of the driveline at both modes are the focus of vibration analysis.
While operating on hybrid drive mode the clutch is engaged. Both the ICE and the electric motor are involved to provide traction. Disturbances or any human orders will directly change the state of the whole driveline resulting in variances of the torsional torque between IEC and ISG. Then hybrid drive mode is another possible running mode related to shaft failure. Section 4 will focus on analyzing the above two running modes respectively.

3.2 Other aspects

The equivalent parameters of stiffness, inertia can be obtained using the principle introduced in [10]. Inertias of those small rotating accessories are either neglected or respectively transferred to their host components.
The torque-angle relation of the clutch is depicted in Figure 4. It is noted from the figure that three different torsional stiffness rates exist with the clutch either for a negative or positive rotation. As the clutch stiffness is piecewise linear, free vibration analysis should be performed independently for each linear stage of the clutch to determine the natural frequency and corresponding mode shapes.

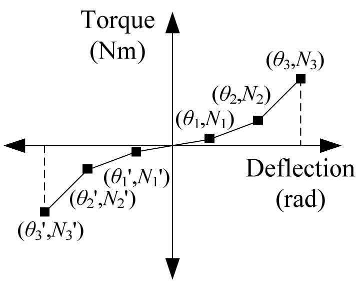


Figure 4: Torque - angle relation of the clutch

It should be noted that the torsional damper in Figure 3 can be regarded as a combination of springs and dampers. In other words, it acts as a stiffness device after neglecting its damping. The stiffness is also piecewise linear as shown in Fig. 4.

4 Vibration analysis

4.1 ICE start/stop

At either of the start or stop mode, only the ICE and ISG works. In other words, the driveline is reduced to a system of two degrees of freedom as shown in Figure 5. In the figure, θ_1 and I_1 are the angular displacement and inertia of the ICE, θ_2 and I_2 are corresponding variables of the ISG, while K_1 is the equivalent stiffness of the shaft between ISG and ICE.

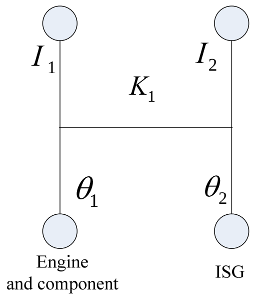


Figure 5: Two degrees of freedom model of driveline

It is clear that the system shown in Fig. 5 is a semi-definite system. In another word, the system has a zero natural frequency corresponding to the rigid rotation of the whole system. Considering the piecewise linearity of the torsional damper, six natural frequencies are obtained through a series of modal analysis, which are listed as follows

Table 1: Torsional natural frequencies of two degrees of freedom model (order 1 =0Hz)

K_1 (Nm/(°))	30	33	88	100	190	230
Natural frequencies (Hz)	7.62	7.99	12.44	13.91	19.17	21.10

Different K_1 represents various equivalent stiffness namely different working situations of the torsional damper between ISG and ICE.

The normalized mode shape is plotted in Figure 6 where nodes 1 and 2 represent the ICE and the ISG, respectively. Torque fluctuation of the ICE varies with the rotational speed of ICE. For ICE with six cylinders and four cycles, the 3rd and the 6th harmonics are dominant. Figure 7 presents an interference diagram in an engine speed range from 0 to 800 rpm. The figure clearly depicts all natural frequencies and the variation of the two harmonics, from which one can figure out the critical speed of the engine.

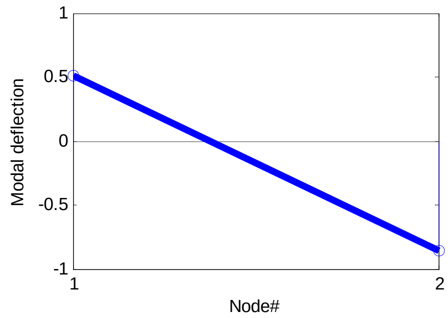


Figure 6: The second mode shape

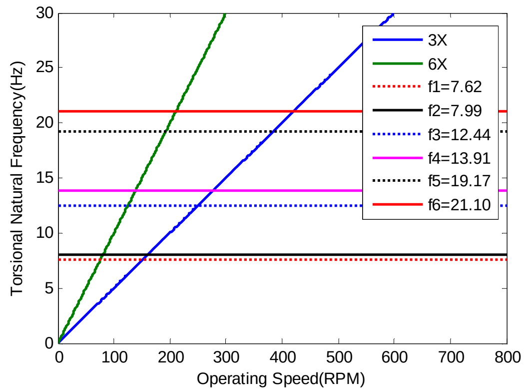
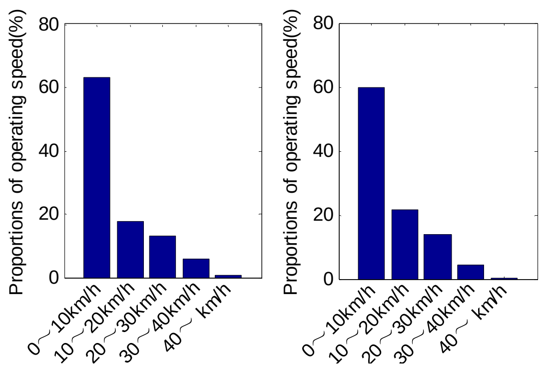


Figure 7: Interference diagram (0–800RPM)

It can be found in Figure 7 that there are 12 critical speeds varying from 76.0 to 422.0 rpm. During the start mode, only the first two sets of springs of the torsional damper will play a role in either direction. Recalling Fig. 4, one will note that the starting torque lies between N_2' and N_2. As a result, possible resonance will occur when the speed of ICE is under 300rpm derived from the second stage stiffness of torsional damper.

Figure 8 illustrates a statistic analysis on speed proportion of a city bus in actual city traffic driving cycles.



(a) Forward (b) Backward

Figure 8: Speed proportions of a city bus in actual city traffic driving cycles

Fig. 8 reveals that a city bus usually operates at a speed less than 30km/h, with a proportion up to

95%. It implies that the ICE starts or stops very frequently.

Although a resonance lasts just for a very short period of time, high repetition make it possible for the driveline to suffer from fatigue problem. In engine start-up the speed sweep range of 0-1000RPM considering 200RPM overshoot so the maximum major excitation frequency is approximate 50Hz (six cylinders and four cycles). A straightforward solution to avoid resonance is to adjust the driveline parameters so as to have the natural frequencies bigger than the maximum exciting frequency. The applicability of such a solution will be discussed in section 3.

4.2 Hybrid drive mode

In this mode the clutch is engaged then most components are involved. Disturbances or any human order will change the state of the whole driveline resulting in variances of the torsional torque between IEC and ISG. So we need to analyze the vibration situation under hybrid drive mode. A model with six degrees of freedom can be constructed as shown in Figure 9. The computed natural frequencies are listed in Table 2. It should be noted that the first order frequency is 0Hz corresponding to rigid rotation.

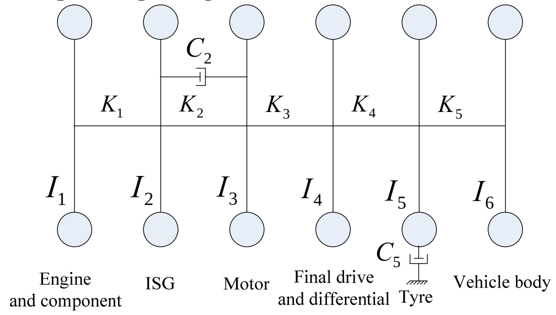


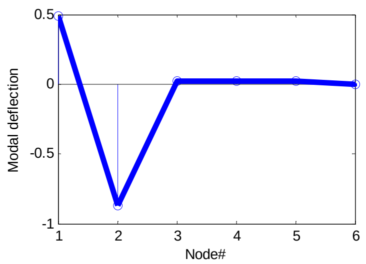
Figure 9: Six degrees of freedom model of driveline

Table 2: Natural frequencies for hybrid operational mode

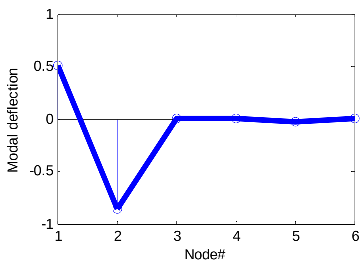
Case	Equivalent stiffness of K_1, K_2 (Nm/(°))	Order 2 (Hz)	Order 3 (Hz)	Order 4 (Hz)	Order 5 (Hz)	Order 6 (Hz)
1	30, 3.3	1.11	3.02	7.79	20.28	248.66
2	33, 3.3	1.11	3.02	8.15	20.28	248.66
3	80, 3.3	1.12	3.03	12.54	20.28	248.66
4	100, 3.3	1.12	3.03	14.00	20.28	248.66
5	190, 3.3	1.12	3.03	19.24	20.28	248.66
6	230, 3.3	1.12	3.03	20.28	21.16	248.66
7	30, 180	1.90	5.56	17.46	21.07	248.66
8	33, 180	1.91	5.78	17.54	21.08	248.66
9	80, 180	1.94	8.16	18.75	21.40	248.66
10	100, 180	1.94	8.79	19.20	21.66	248.66
11	190, 180	1.95	10.38	20.19	23.95	248.66
12	230, 180	1.95	10.75	20.32	25.26	248.66

Different cases represent various stiffness combinations namely different working situations of clutch and torsional damper. From Table 2, it can be found that the highest value of the first 5 natural frequencies is 25.26Hz

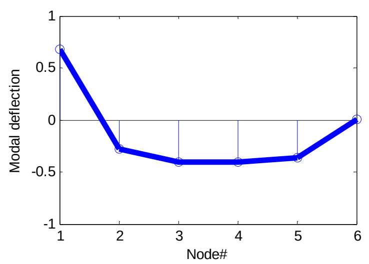
In order to figure out the maximum relative torsional angle between ICE and motor, which is proportional to the shaft stress, we can check the normalized mode shapes as shown in Figure 10.



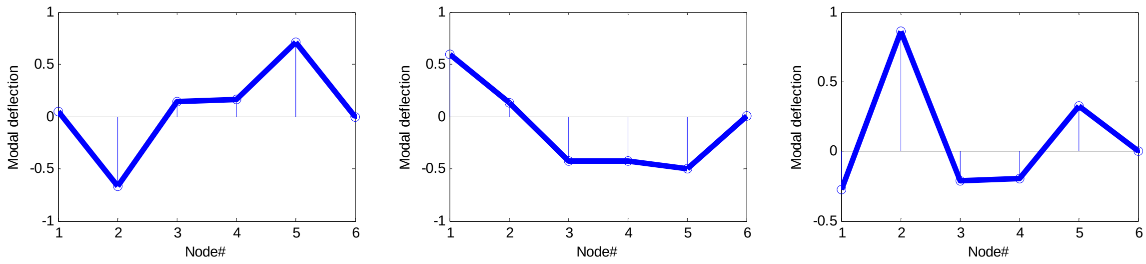
Case 1−5 order4



Case 6 order 5



Case 7−10 order3



Case 7−10 order4 Case 11−12 order3 Case 11-12 order5

Figure 10: Normalized mode shapes

Table 2 shows the 6th torsional frequency is much higher than the 5th. Looking at the 6th mode shape plotted in Figure 11, we can find that Node 4 or the assembly of the final drive and differential dominate this vibration mode. This agrees with an observation that the natural frequency of the assembly itself is higher than those of other components.

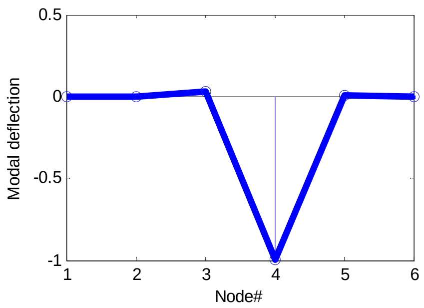


Figure 11: The 6th mode shape

Fortunately the 6th vibration mode cannot be excited because the corresponding critical speed of 5000rpm is far beyond the operating speed range of a diesel ICE. As a sequence, the highest natural frequency that should be reviewed for possible resonance is about 26Hz (Order 5 in Case 12).

An interference diagram is plotted in Figure 12 covering a speed range of 0-1200rpm. The highest critical speed is lower than 600rpm. But it is assumed that the hybrid power drive could only be applicable at a speed no less than 800rpm. Therefore no resonance will occur in hybrid drive.

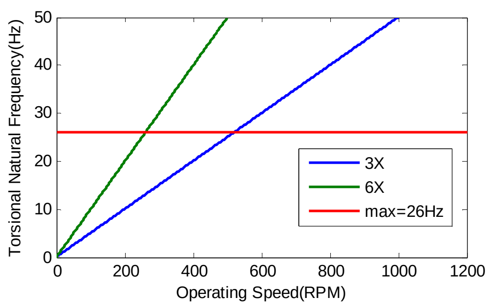


Figure 12: Interference diagram (0-1200rpm)

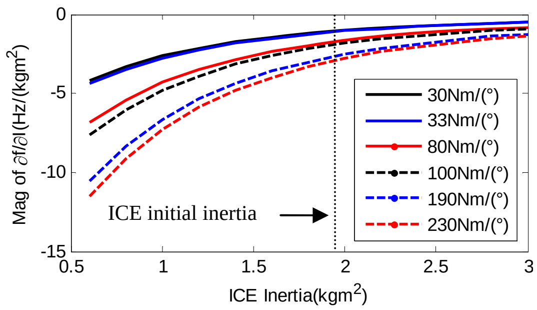
5 Sensitivity analysis

A sensitivity analysis is conducted to figure out the main component related to an abnormal vibration problem, and to give a clue to possible solutions to improve design with the driveline.

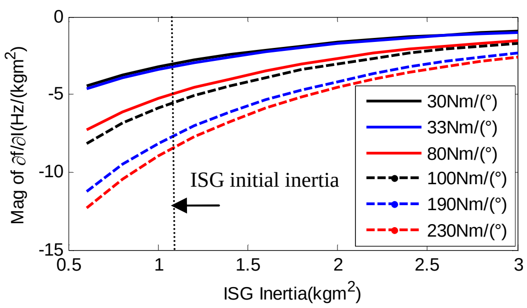
Using Eq. (6) we can derive an explicit formula of the sensitivity of each natural frequency of the system with respect to a design variable b as

$$\frac{\partial \omega_i}{\partial b} = \frac{1}{2\omega_i}\psi_i^{\mathrm{T}}\frac{\partial \mathbf{K}}{\partial b}\psi_i - \frac{\omega_i}{2}\psi_i^{\mathrm{T}}\frac{\partial \mathbf{I}}{\partial b}\psi_i \qquad (7)$$

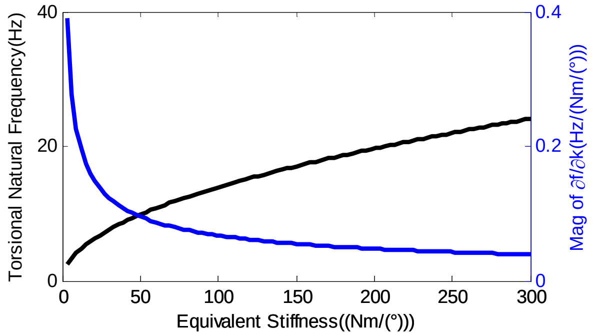
Based on the previous discussions, only the sensitivities of natural frequencies of the driveline at ICE start/stop mode is illustrated in Figure 13 because a resonance is less likely to occur in hybrid drive, which we have mentioned before.



(a) ICE inertia



(b) ISG inertia



(c) Stiffness between ICE and ISG

Figure 13: Frequency sensitivity analysis

Figure 13 clearly show that the natural frequencies become more and more sensitive to the inertia and stiffness when the inertia decreases or the equivalent stiffness increases. For a predefined driveline, the inertia of both an ICE and a ISG can only be changed within a small range. Thus the equivalent stiffness dominates the natural frequency. However, the biggest natural frequency can be increased up to 30Hz for the targeted driveline when the equivalent stiffness taken the a pretty big value 300Nm/(°). Referring to Fig.7, the corresponding critical speed is 600rpm which is still lower than the idle speed of the ICE. It is thus impossible to avoid resonance by changing the ICE inertia, ISG inertia or the equivalent stiffness. As a sequence, it is a reliable solution to properly introduce dampers into the driveline so as to effectively reduce the torsional vibration of the driveline.

6 Conclusions

Torsional vibration models are presented for a power split hybrid driveline. Natural frequencies and mode shapes are calculated for two main combinations of the power sources. Sensitivity analysis is then performed on the natural frequencies with respect to the lumped inertia or equivalent stiffness of the system. Discussions reveal that the shaft failure is resulted from resonance occurring with high repetition which is hardly avoided by changing inertia or stiffness parameters.

Acknowledgments

This research was sponsored by Jing-Jin Electric Technologies (Beijing) Co., Ltd. The authors greatly appreciate the support of all people who helps with the research.

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