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Detection of Running Deviation of Cardan Shaft in High-speed Train

*Cai Yi [1)], Jianhui Lin[2)], Lu Liu [3)] and Kwok-Leung Tsui [4)]

[1), 4)] *City University of Hong Kong, Kowloon Tong, Hong Kong, 999077, China*
[2), 3)] *Southwest Jiaotong University, Chengdu, Sichuan, 610000, China*
[1)] *justin.yi@163.com*

ABSTRACT

The unbalance of cardan shaft compromises operations of high-speed train. A new and engineered method is proposed to detect the running deviation by applying Hilbert-Huang Transform (HHT). The vibration acceleration of gearbox was decomposed into IMFs and the characteristic frequency band could be separated effectively by applying EMD to vibration signal. Furthermore, the corresponding IMFs component of the characteristic frequency in drive system can be got by the center frequency of each IMF. When dynamic imbalance or other faults occur in the cardan shaft, the energy of characteristic frequency would change correspondingly, so the condition information of the gearbox vibration signal can be extracted effectively from the instantaneous frequency energy spectrum. The validity of this method is supported by the bench test and in-service train monitor experience data. The analysis results show that it is feasible to estimate the work state of cardan shaft from gearbox vibration by characteristic frequency separation method and instantaneous frequency energy, and the detection mode has an exciting application effect in engineering, however, the detail mathematical proof is lacking for the detection model.

1. INTRODUCTION

The drive system of China Railway High-speed 5 (CRH5) is a special and complex mechanical system, which consists of the gearbox, cardan shaft, traction motor, other rotating parts and supporting component parts. The traction motor of power bogie adopts body suspension structure, and axle holding structure for the gearbox. In the special service surroundings of the cardan shaft in CRH5 shown in Fig.1. The cardan shaft disposed longitudinally is connected to the motor and gearbox, and plays an important role in the drive system [1]. It not only transfers traction torque but also

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[1)] Postdoctoral Research Fellow
[2)] Professor
[3)] Lecture
[4)] Professor

coordinates the complex motion relationship between two universal joints. In addition, both bending and torsion stiffness of cardan shaft are small due to its elongated structure. As a result, cardan shaft easily deform and produce eccentric loads over the long service life of high-speed train. Other factors, which contribute to worsening eccentricity over time, include the gap to the rear of the shaft between the axis of the universal joint and the looseness of the balance slide blocks. Eccentricity quickly creates additional dynamic unbalance torques and gives rise to excessive vibrations, which lead to rapid damage of power transmission components such as bearings and universal joints. If the cardan shaft occurs the dynamic imbalance, sintered bearings, cover wear and failure, too high bearing temperature, locking and other faults, all of these situations will greatly affect the normal use of the drive system, or even develop to be a direct threat to the traffic safety [2]. In practical application, there are more high performance requirements for couplings, because it must pass the traction torque and also adapt to the variety of complex motion relationships. The performance status of the coupling represents that of the entire drive system [3], [4]. Due to the poor applying environment including dust, large temperature span and high speed, the cardan shaft is always in harsh service condition, at the same time the performance status of the coupling represents that of the entire drive system. So it is of crucial importance and is a necessary measure to ensure the safety and reliability of the drive system that implementing real-time detection and condition estimation to cardan shaft.

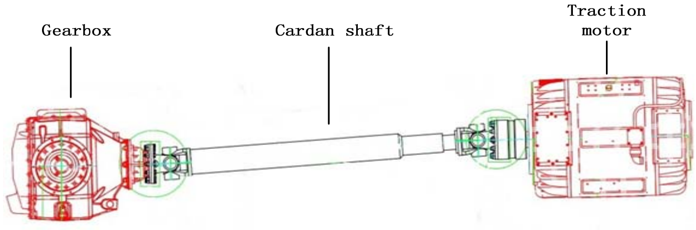

Figure 1: CRH5 drive system

There is a newly proposed signal processing method, namely Hilbert-Huang transform (HHT) developed by Huang et al [5]. It is a powerful tool to process non-stationary signals and has been widely applied to mechanical fault diagnosis such as bearing fault diagnosis [7], gearbox fault diagnosis [8], and rotor unbalance detection [9]. As we know, time scale and the corresponding energy distribution are two critical characteristic features of a signal in signal processing, in which energy distribution with the change of time and frequency could reflect the dominant features of the signal effectively. For example, when gears operate in a normal condition, the energy of vibration signal would focus on gear meshing frequency and its multiple frequencies. However, while crack fault or broken teeth occur in gears, sidebands would appear in spectrum because the vibration signals present modulation characteristic, scilicet, at this moment the energy distribution has changed. Therefore, it is possible to detect a fault by comparing the energy distribution of gearbox vibration signals with and without fault conditions in time-frequency domain, namely, the energy variation in time-frequency plane may indicate fault occurrence. When there is defect in the gearbox, the vibration signal presents the non-stationary characteristics, thus Hilbert-Huang transform has been applied to gearbox defect diagnosis widely [10]. While Hilbert spectrum that could be obtained by performing Hilbert-Huang transform exactly offers the accurate amplitude distribution with the change of time and frequency, scilicet, it provides a complete energy-frequency-time distribution. HHT, which includes EMD (empirical mode decomposition) and Hilbert transform, is based on the local characteristic time scales of a signal and could decompose the complicated signal into a number of IMFs (intrinsic mode functions). Frequency components contained in each IMF not only relate to the sampling frequency, but also change with the signal itself. Furthermore, the whole transform process of HHT would not lead to energy diffusion and leakage. Therefore, HHT is a self-adaptive signal processing method and could be applied to non-linear and non-stationary signal processing perfectly [11], [12].

Considering practical engineering application, we focus on the indirect detection method research of the cardan shaft based on the gearbox vibration acceleration as there is no effective monitoring to directly access the signal of the cardan shaft state. In this paper, the preliminary evaluation model of cardan shaft condition estimation from gearbox vibration basing on Hilbert spectrum and instantaneous frequency energy; before this, we researched the characteristic frequency separation method by EMD and explored how to determine the location of the characteristic in IMFs. All of the research work are based on the bench test and in-service train monitoring experiment, and finally the calculation model is preliminarily verified by these signal data.

2. DATA ACQUISITION

Cardan shaft is a high speed rotating component, and it is so hard to measure directly the vibration acceleration. The ends of cardan shaft are connected to the traction motor and gearbox, so we must explore and compare the vibration contribution and performance of cardan shaft to two sides separately.

According to the structure of the traction motor, there is a spiral spring below the motor to achieve elastic suspension which buffer and weaken most of the vibrational energy of the motor. The vibration contribution of dynamic imbalance of cardan shaft to the motor is relatively small, so it is negligible for the preliminary estimation of the cardan shaft condition. However, it is quite necessary to monitor the vibration of the motor when we want to detect and identify the fault of the drive system accurately. On the other hand, the contribution of dynamic imbalance of cardan shaft to the gearbox is quite significant, and the gearbox vibration contains more condition information of the drive system. To prove the effect relationship between gearbox vibration and cardan shaft state, we implemented the bench test (Fig.2) and in-service train monitoring (Fig.3) to the drive system.

There are three cardan shafts for the bench test, and the states of these shafts respectively are new (the red line represented), special repairing (the purple line represented) and close to the use limit (the black line represented); the bench test result is showed as Fig.4. It can clearly be seen that the higher the speed is, the more pronounced the dynamic imbalance state of the cardan shaft becomes; the worse the condition of the cardan shaft is, the larger the vibration of the gearbox caused by the cardan shaft is. Obviously, the gearbox vibration character to some extent reflects the dynamic imbalance state of cardan shaft.

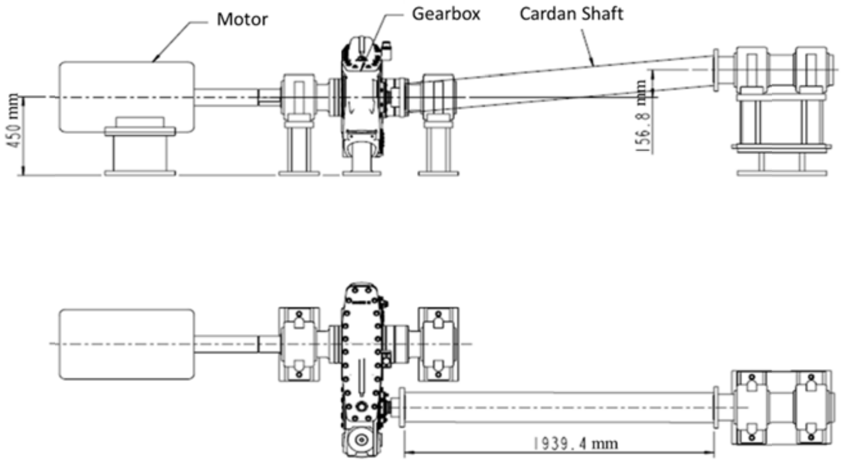

Fig.2 Bench test

Fig.3 In-service train monitoring

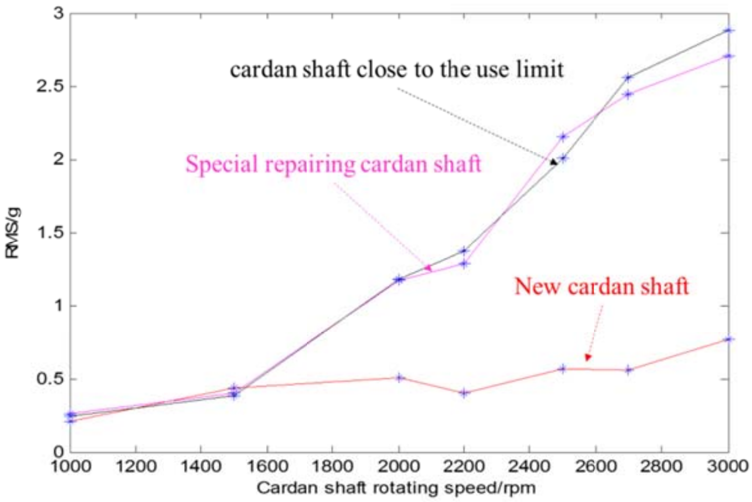

Fig.4 Gearbox vertical vibration effective-value and cardan shaft rotating speed

For exploring the mapping relationship between the gearbox vibration and the cardan shaft running deviation state in practical application, we conducted an in-service train monitoring experiment and picked up the real-time detecting data while the train was running at the different speed. To avoid changing the structure of the drive system and bringing additional risks to the train, The sensor was seated on the auxiliary hole where is in upper of the gearbox to monitor the vibration acceleration of the gearbox, and as shown in Fig.3 can see that an advantage of gearbox acceleration measurement device is their simple structures, which make it easier to carry out maintenance. However, the gearbox acceleration waveform contains too much vibration information, and the amplitude greatly depends on the vehicle speed. This paper to estimate the cardan shaft condition from gearbox vibration mainly is based on the real-time detecting data, which is more persuasive and has more practical significance.

3. THE BASIC THEORY

3.1 Hilbert-Huang Transform

Hilbert-Huang transform includes empirical mode decomposition (EMD) and corresponding Hilbert transform. EMD method is developed from the simple assumption that any signal consists of different simple intrinsic modes of oscillations. EMD can eliminate the signal of riding waves (small-scale waves “riding” on the large-scale waves) by several moving processes, and each linear or non-linear mode will have the same number of extreme and zero-crossings. There is only one extra-mum between successive zero-crossings. Each mode should be independent of the others. In this way, it can smooth un-even signals, and each signal could be decomposed into a number of intrinsic mode functions (IMFs), each of which must satisfy the following definition [5]:

(1) In the whole data set, the number of extreme and the number of zero-crossing must either equal or differ at most by one.
(2) At any point, the mean value of the envelope defined by local maxima and the envelope defined by the local minima is zero.

An IMF represents a simple oscillatory mode compared with the simple harmonic function. With the definition, any signal $x(t)$ can be decomposed as:

$$x(t) = \sum_{i=1}^{n} c_i + r_n(t) \quad . \tag{1}$$

The original can be expressed as the sum of all the IMFs and the residue. The IMFs include different frequency bands ranging from high to low. The frequency components contained in each frequency band are different and they change with the variation of signal $x(t)$, while $r_n(t)$ represents the central tendency of signal $x(t)$.

For each IMF $c_i(t)$, we can always have its Hilbert transform, and $f(t)$ can be expressed by convolution of $f(t)$ and $1/\pi x$ as:

$$\hat{c}_i(t) = c_i(t) * \frac{1}{\pi t} = \int_{-\infty}^{+\infty} c_i(t') \frac{1}{\pi(t-t')} dt = \int_{-\infty}^{+\infty} c_i(t-t') \frac{1}{\pi t'} dt \ . \tag{2}$$

Then the analytical signal of the original signal is obtained by:

$$z_i(t) = c_i(t) + i\hat{c}_i(t) = a_i(t)e^{j\theta_i(t)} . \quad (3)$$

$$a_i(t) = \sqrt{c_i(t)^2 + \hat{c}_i(t)^2} , \quad (4)$$

$$\theta_i(t) = \arctan(\frac{\hat{c}_i(t)}{c_i(t)}) . \quad (5)$$

Instantaneous amplitude and instantaneous phase are expressed by formula (4) and (5). In formula (5), we can have the instantaneous frequency as:

$$\omega_i(t) = \frac{d\theta_i(t)}{d(t)} . \quad (6)$$

Then

$$z_i(t) = c_i(t) + i\hat{c}_i(t) = a_i(t)e^{j\theta_i(t)} = a_i(t)e^{j\int_0^T \omega_i(t)dt} \quad (7)$$

After performing the Hilbert transform to each IMF component, the original signal can be expressed as the real part (Re) in the following form:

$$x(t) = \sum_{i=1}^{n} c_i(t) = \mathrm{Re}\sum_{i=1}^{n} z_i(t) = \mathrm{Re}\sum_{i=1}^{n} a_i(t)e^{j\theta_i(t)} = \mathrm{Re}\sum_{i=1}^{n} a_i(t)e^{j\int_0^T \theta_i(t)dt} \quad (8)$$

Here we left out the residue r_n on purpose, for it is either a monotonic function or a constant. The formula (8) gives both amplitude and frequency of each component as functions of time. This frequency-time distribution of the amplitude is designated as the Hilbert spectrum $H(\omega,t)$.

$$H(\omega,t) = \mathrm{Re}\sum_{i=0}^{n} a_i(t)e^{j\int_0^T w_i(t)dt} \quad (9)$$

That T is the total data length. With the Hilbert spectrum defined, the Hilbert marginal spectrum can be shown as:

$$h(\omega) = \int_{-\infty}^{+\infty} H(\omega,t)dt . \quad (10)$$

Obviously, the Hilbert spectrum offers a measure of amplitude distribution from each frequency and time, while the marginal spectrum gives a measure of the total amplitude distribution from each frequency.

Overall，HHT is a self-adaptive signal processing method and could be applied to non-linear and non-stationary signal processing perfectly [13],[14]， which was demonstrated to be superior to wavelet analysis in many applications. In this paper, we applied this method for time-frequency analysis of measured data.

3.2 Characteristic frequency separation based on EMD

According to the motion transmission principles and structure of the drive system, there are some characteristic frequencies which have close correlation with cardan shaft condition are shaft rotation frequency, pinions rotating frequency, big gear rotating frequency and gear mesh frequency. All of these characteristic frequencies are calculated with the real-time train speed (v), wheel diameter (d) and transmission ratio (i).

According to the structure of the cardan shaft and gearbox, the cardan shaft rotation frequency is approximately equal as pinions rotating frequency, which is calculated as:

$$fw = \frac{v}{3.6 \times \pi \times d} \times i \tag{11}$$

According to the wheel turning repair record, the wheel diameter d is 0.88m, and the transmission ratio i is 2.22. When the train is running at the speed 250km/h, the real-time train speed v is 69.44m/s; Calculated by formula (11), the cardan shaft rotation frequency is 55.76Hz.

The gear mesh frequency is calculated as following:

$$fn = fw * n \tag{12}$$

The big gear rotating frequency (approximately equal as train-wheel rotation frequency) is calculated as:

$$fc = \frac{v}{3.6 \times \pi \times d} \tag{13}$$

When the train is running at the speed 250km/h, the gear mesh frequency fn is 1505.7Hz, and the big gear rotating frequency fc is 25.12Hz.

In order to verify the accuracy of the theoretical calculations, we select one set gearbox vibration acceleration data while the train is running at the speed 248km/h to spectral analysis. It is seen that in-service train monitoring experimental values are fairly consistent with the theoretical values by Fig.5.

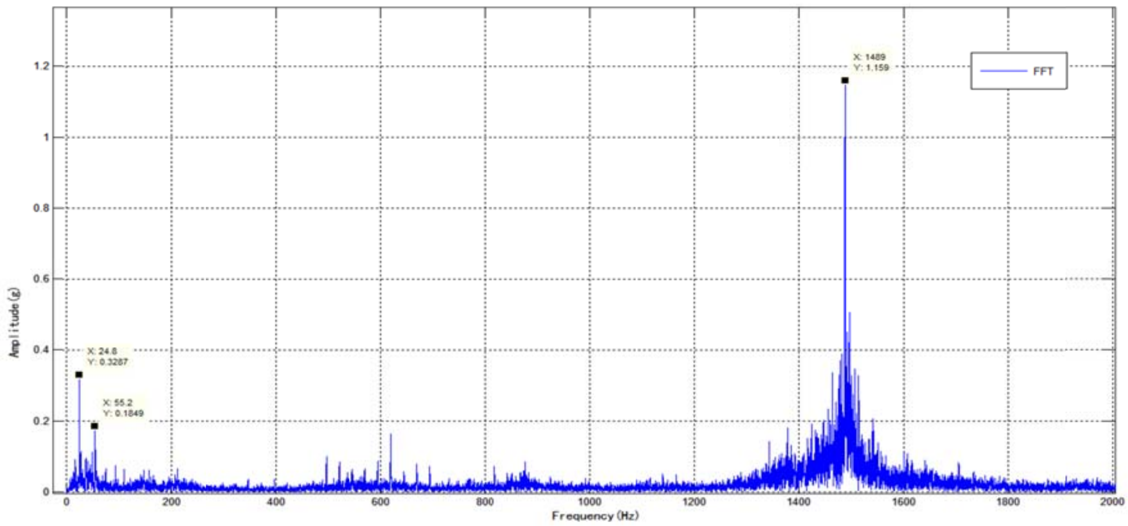

Fig.5 Gearbox vibration spectrum while speed 248km/h

When the cardan shaft with the dynamic imbalance or the gears with fatigue crack are meshing, both the amplitude and phase of vibration signal would be modulated. Leaving out the effect of transport function, the gearbox vibration signal picked up by sensor that is seated on the gearbox can be expressed as following [15]:

$$y_i(t) = \sum_{m=1}^{M} X_m [1 + d_m] \cos[2\pi mzf_w + \phi_m + b_m(t)]. \tag{14}$$

The formula (14) can be also expressed as:

$$y(t)=\sum_{m=1}^{M}p_m(t)\cos\theta_m(t)\,. \tag{15}$$

The commonly used demodulation methods in engineering include Hilbert transform and traditional envelope analysis. However, in these methods it is needed to choose the band-pass filter to separate the frequency families shown in the formula (14) before the envelope signal is obtained. Unfortunately, both the centre frequency and bandwidth of the band-pass filter are usually needed to determine by experience in advance, which would inevitably make great subjective influence on the analysis results [15]. In addition, according to the formula (3) - (5), each IMF resulted from EMD of the gearbox vibration signal can be expressed as:

$$c_i(t)=a_i(t)\cos\phi_i(t)\,. \tag{16}$$

As the envelope amplitude function $a_i(t)$ obtained by the formula (4) is a slowly changing signal compared with the phase function $\phi_i(t)$ obtained by the formula (5), each IMF $c_i(t)$ resulted from EMD can be the signal which contains the frequency and phase information. Therefore, omitting the residual r_n, the formula (1) can be expressed as:

$$x(t)=\sum_{i=1}^{n}a_i(t)\cos\phi_i(t)\,. \tag{17}$$

Comparing the formula (15) and (17), we know that gearbox vibration signal consists of a number of frequency family components, each of which is an amplitude modulation signal. On the other hand, the gearbox vibration signal consists of a number of IMFs, each of which is also exactly a modulation signal. The representation forms in the formula (15) and (17) are different, however, the frequency components contained in gearbox vibration signal are same. Therefore, it is viable to apply EMD method to decompose the gearbox vibration signal into a number of IMF components, in which it contains the information of the cardan shaft dynamic unbalance condition and other fault in the drive system.

Fig. 6 gives the IMFs of a set gearbox vibration acceleration data and the residue produced by the EMD. The IMF_{1-11} are the effective frequency components, and the IMF_{12} is the residual frequency component that the whole signal deduct IMF_{1-11}, represented a trend. However, there are so many frequency components, and it is very difficult to identify the real or pseudo component, in another word we can't discern which IMF or frequency component is useful for further analysis and calculation.

If we grasp the position of the characteristic frequency of the drive system in which IMF, maybe can we discover the target frequency range to emphatically analyze. In order to identifying the position of the characteristic frequency in which IMF, we calculated the center frequency of each IMF showed as Table. 1.

When the train running speed is 248km/h, and wheel diameter is taken as 0.88m, based on the formula (11) ~ (13), the meshing frequency of the drive system is 1494Hz; the cardan shaft rotation frequency is 55.3Hz. There is a very high alignment between the cardan shaft rotation frequency and IMF_9 centre frequency, so IMF_9 is the position of the cardan shaft rotation frequency; however, the gear meshing frequency is in the middle of the IMF_3 and IMF_4, and how to analysis and identify? IMF_3 and IMF_4 are exerted to Hilbert transform to get the Hilbert marginal spectrum, and the spectrum features can be surveyed from Fig.7. In Fig.7, the vibration amplitude is expressed by different colors, and by comparing the gear meshing frequency mainly concentrated in IMF_3.

Obviously, it is a quietly effective method to separate and extract the frequency components from the gearbox vibration by using EMD, centre frequency and Hilbert marginal spectrum.

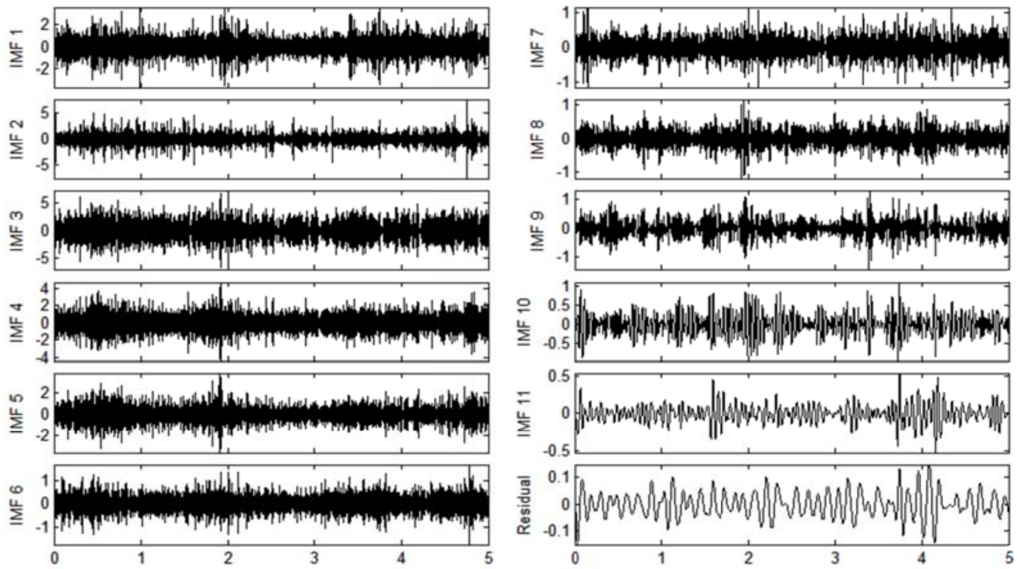

Fig.6 The IMFs and residue of gearbox vibration acceleration

Table.1 Center frequency of each IMF

Number	IMF_1	IMF_2	IMF_3	IMF_4	IMF_5	IMF_6	IMF_7	IMF_8	IMF_9	IMF_{10}	IMF_{11}
Center frequency/Hz	6204	3199	1702	1128	661.2	422.8	217.7	118.1	54.58	30.59	17.79

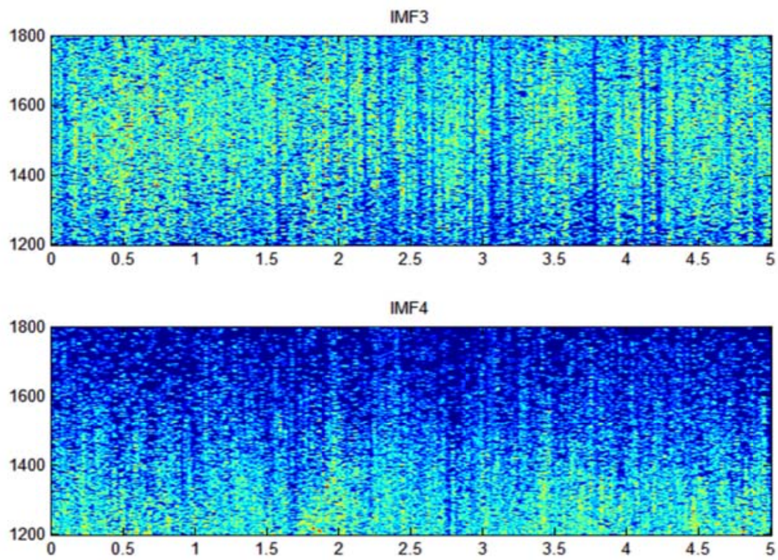

Fig.6 The Hilbert marginal spectrums of IMF_3 and IMF

4. RUNNING DEVIATION DETECTION MODEL

4.1 Detection model

When dynamic imbalance or other faults occur in the cardan shaft, the energy of the gearbox vibration signal would change strongly in some frequency bands, but in other frequency bands the energy maybe change weakly. Therefore, it is possible to identify the condition according to energy variation of the gearbox vibration signals. Analysing the whole process of Hilbert-Huang transform, it is obvious that both frequency and amplitude of each IMF are the function of time. So the Hilbert spectrum $H(\omega, t)$ offers a measure of amplitude distribution with change of every time and frequency [14]. If $|x(t)|^2$ is regarded as energy density of the vibration signal $x(t)$, then $H^2(\omega, t)$ obtained by performing HHT owns the same physical meaning. In additionally, the Papseval's theorem also proves that $H^2(\omega, t)$ is equal to $|x(t)|^2$. Leaving out the residue r_n, the HHT of $x(t)$ should be energy conservation, namely, the following formula could be obtained:

$$\int_{-\infty}^{\infty} |x(t)|^2 dt = \int_{-\infty}^{\infty}\int_{-\infty}^{\infty} H^2(\omega, t) d\omega dt \tag{18}$$

We can define instantaneous energy $E(t)$ as following:

$$E(t) = \int_{-\infty}^{\infty} H^2(\omega, t) d\omega \tag{19}$$

However, in most time the noise in the gearbox vibration signal is so strong that the condition feature cannot be extracted directly from the Hilbert energy spectrum $H^2(\omega, t)$ or the instantaneous energy $E(t)$ which provides the energy distribution of the time domain. Thus, we can select some IMF that we are interested in to perform Hilbert transform and local Hilbert spectrum $H'(\omega, t)$ can be got as:

$$H'(\omega, t) = \mathrm{Re}(\cdots + a_i(t)^{j\int \omega_i(t)dt} + a_k(t)^{j\int \omega_{ki}(t)dt} + \cdots) \tag{20}$$

With the definition of local Hilbert energy spectrum $H'^2(\omega, t)$, we could get:

$$E'(t) = \int_{-\infty}^{\infty} H'^2(\omega, t) d\omega \tag{21}$$

The $E'(t)$ offers a measure of local instantaneous energy that reflects signal energy distribution at certain frequency band with change of each time. When dynamic imbalance or other faults occur in the cardan shaft, it is so difficult to diagnosis based on variation of vibration signal energy in the practical engineering. However, the energy variation resulted from the dynamic imbalance always is always decentralized and small compared with the total energy of the gearbox vibration signal. Thus, there is a certain difficulty to detect the fault by comparing this kind of variation. While the local energy spectrum $H'(\omega, t)$ could exactly provide precise energy-frequency–time distribution of signal with the change of time and some frequencies, it is possible to extract cardan shaft work state characteristic by analyzing signal energy distribution with change of frequency and time. The simplest method to get the local energy spectrum $H'(\omega, t)$ is to implement Hilbert transform with the IMFs, in this way we can pick up each IMF's energy spectrum $H_i'(\omega, t)$, and according to Eq. (21), we also could pick up each IMF's instantaneous energy $E_i'(t)$.

There is coupling effect and influence between cardan shaft and the gearbox, so it is unreasonable to judge the work sate of cardan shaft according to a certain local energy or the variation. Considering the characteristic frequency of the cardan shaft and gearbox, they are cardan shaft rotation frequency, train-wheel rotation frequency and gear mesh frequency. When the train running speed is higher, namely the train-wheel rotation frequency is higher, and the vibration amplitude of the gearbox is larger, which is showed as Fig.4. When dynamic imbalance or other faults occur in the cardan shaft, the shaft rotation frequency energy and the gear mesh frequency energy both would be increased, and what is showed as Fig.7-8, the blue curve represented the new cardan shaft experience result, and the red one represented the old cardan shaft which occur dynamic imbalance to some degree.

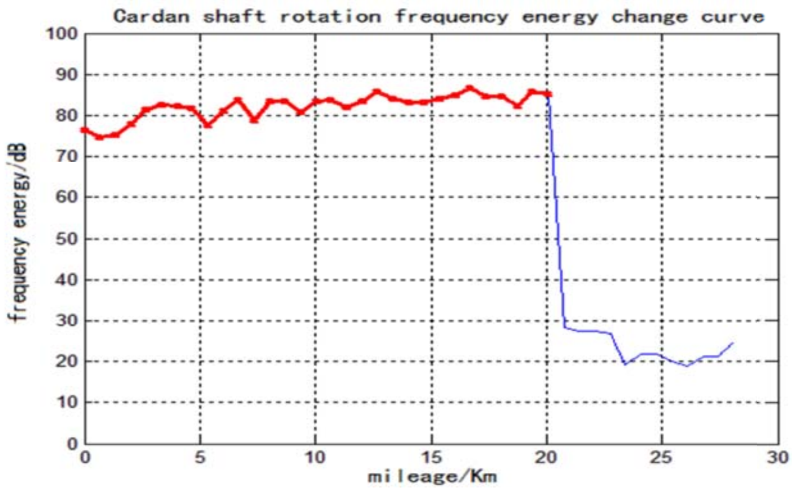

Fig. 7 Cardan shaft rotation frequency energy comparison

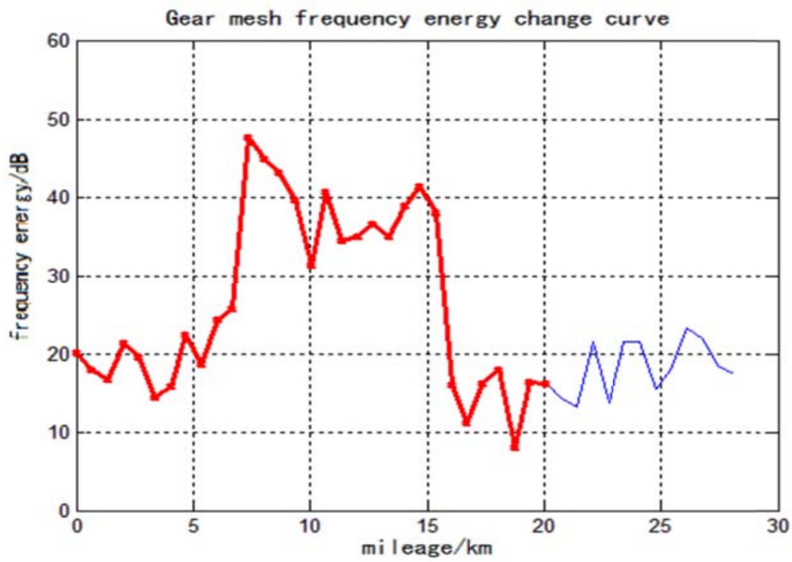

Fig. 8 Gear mesh frequency energy comparison

So maybe we can use such a function to express the instantaneous working state of the cardan shaft:

$$WS = \sqrt{aE'^{2}_{f_\omega}(t) + bE'^{2}_{f_n}(t) + cE'^{2}_{f_c}(t)} + d \quad (22)$$

where WS is a performance dimension of the work state of the cardan shaft; a is weight coefficient of the cardan shaft rotation frequency instantaneous energy; b is weight coefficient of the gear mesh frequency instantaneous energy; c is weight coefficient of the train-wheel rotation frequency instantaneous energy; d is the correction coefficient which is based on the actual engineering test.

To verify the feasibility of the calculation model, we used the bench test and in-service train monitoring experiment data which are picked up from two cardan shaft one new and the other is old to calculate, and the result is basically in accordance with the cardan shaft state. The detection mode has an exciting application effect in engineering, however, the detail mathematical proof is lacking for the detection model.

4.2 Other evidence

If we apply the frequency domain method to analysis the same question, maybe can we obtain more proof. In Mathematics, the Plancherel theorem is a result in harmonic analysis, proven by Michel Plancherel in 1910. It states that the integral of a function's squared modulus is equal to the integral of the squared modulus of tis frequency spectrum, and it can be expressed as

$$\sum_{n=0}^{N-1} \left| X(n) \right|^2 = \frac{1}{N} \sum_{k=0}^{N-1} \left| X(k) \right|^2 \quad (23)$$

The frequency spectrum of the gearbox vibration acceleration every second was used to obtain the frequency domain energy. Due to the speed variation in running process, the frequency bands around characteristic frequencies were collected to calculate the energy peak of power spectrum, and the process can be expressed as

$$E_f = \frac{1}{4N} \left\{ \max\left(\sum_{n=0}^{fs-1} x(n) e^{-j\frac{2\pi}{N}nk} \right) \right\}^2 \quad \left(k = \frac{fs}{N} f_d \ldots\ldots \frac{fs}{N} f_T \right) \quad (24)$$

where E_f is the energy of the characteristic frequency, N is the sampling points in unit time, $x(n)$ is the vibration acceleration, k is the frequency interval of the frequency domain, f_d and f_T is the lower and upper frequency limit of the frequency brand respectively, and f_s is the sampling frequency. In order to reduce the influence of random vibration of the track, the characteristic frequency energy curve of the cardan shaft in the whole trail was smoothed, thus the curve would be expressed as

$$E_{f\_p}(k) = \frac{1}{T} \sum_{t=0}^{T} E_f(t+k) \quad \left(k = 1,2,3\ldots\ldots T_{all} - T \right) \quad (25)$$

where E_{f_p} is the stationary characteristic frequency energy, T is the smoothing time, T_{all} is the whole time length of the data of the whole trail.

In the presented data, the correlation values are N=20000, k=1, f_d =50Hz, f_T =58Hz, f_s =20000, and the calculation result is close to the gear meshing frequency. It is shown that the vibration energy of the running deviation of the cardan shaft would reflect on the vibration energy of the gear meshing frequency mentioned above.

5. CONCLUSTION

A detection model of running deviation of the cardan shaft based on the instantaneous energy using HHT was proposed in this paper. Based on the bench test and in-service train monitor experience, characteristic frequency separation method by EMD and instantaneous frequency energy analysis method by Hilbert energy spectrum were verified, finally, a preliminary condition estimation model of the cardan shaft was set up. From the theory analysis and application results, it can be concluded that:

1. Hilbert-Huang transform can decompose the signal into a number of intrinsic mode functions; each IMF contain the sampling frequency and also changes with the signal itself. Furthermore, there is no energy leakage in HHT. So Hilbert-Huang transform method has shown great recognition performances in analysing the non-linear and non-stationary signals, and applied to effectively extract the condition feature of the cardan shaft in CRH5 drive system.
2. EMD method could be used to separate frequency component of gearbox vibration signal and corresponding IMFs component can be got by the centre frequency. Thus it is possible to analysis each IMF component to extract the drive system abnormal characteristic information.
3. It is feasible to detect the running deviation of the cardan shaft from gearbox vibration by the Hilbert spectrum and instantaneous frequency energy, and the detection mode has an exciting application effect in engineering, however, the detail mathematical proof is lacking for the detection model.

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