TIME VARYING OUTER LOAD AND SPEED ESTIMATION BY VIBRATION ANALYSIS – APPLICATION TO PLANETARY GEARBOX DIAGNOSIS IN A MINING BUCKET WHEEL EXCAVATOR

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Summary

In this paper we attempt to retrieve information about the time varying load/speed condition of a planetary gearbox used in mining excavator system from the vibration measurement. We first define the Instantaneous Power (IP) of the signal, which reflects the global amplitude modulation. For the Instantaneous Speed (IS) an original approach is proposed based on the estimation of a local time-scaling factor, which is well adapted to systems submitted to higher speed variations. Both approaches are applied to two gearboxes, one being in a presumably faulty condition and a new one after replacement. The study of the IS showed that, contrary to the new gearbox, the higher frequency speed variation is highly dependent on the load for the used gearbox, thus indicating a backlash due to wear in the gear/bearing mechanism. Our results also show that IS estimation technique is more reliable than IP approach for outer load estimation. We conclude that while a fault could be detected from the strong amplitude modulation at the arm frequency observed for the used gearbox, the study of the instantaneous speed and outer load variation helped us to refine the diagnosis of this fault.

Keywords: planetary gearbox, diagnostics, time varying load/speed estimation.

ESTYMACJA ZMIENNEGO OBCIĄŻENIA I PRĘDKOŚCI OBROTOWEJ NA PODSTAWIE ANALIZY SYGNAŁÓW DRGANIOWYCH NA POTRZEBY DIAGNOSTYKI PRZEKŁADNI PLANETARNEJ W UKŁADZIE NAPĘDOWYM KOŁA CZERPAKOWEGO

Streszczenie

W artykule przedstawiono metody przetwarzania sygnału drganiowego w celu ekstrakcji informacji o zmienności obciążenia zewnętrznego w przekładni planetarnej w układzie napędowym koła czerpakowego. Zastosowane metody to analiza mocy chwilowej i analiza chwilowej prędkości. Ze względu na duży zakres zmienności obciążeń metoda proponowana przez Bonnardot'a nie może być zastosowana, zatem zaproponowano nowy algorytm wyznaczania prędkości chwilowej. Przeanalizowano dwie przekładnie: w złym stanie technicznym oraz nową, po wymianie uszkodzonej. Wyniki analizy prędkości chwilowej dla przekładni nowej pokazują: niewielką amplitudę i małą zmienność dla częstotliwości drugiej harmonicznej jarzma, dla przekładni uszkodzonej składowe silnie zależą od chwilowej wartości obciążenia, co wskazuje na przekroczoną wartość luzu w przekładni/łożyskach. Wykazano zalety wynikające z analizy chwilowej prędkości w porównaniu do chwilowej mocy. Zauważono również, że o ile nieprawidłową pracę jarzma można wykryć przy użyciu analizy modulacji AM to zastosowanie dodatkowo analizy chwilowej prędkości może zwiększyć wiarygodność diagnozy.

Słowa kluczowe: przekładnia planetarna, diagnostyka, estymacja chwilowej wartości obciążenia/prędkości.

1. INTRODUCTION

Usually majority of proposed analysis is dedicated to new very fault-sensitive methods of signal processing with silent assumption about stationary (in fact constant) load. In many real-life situations, gearbox during normal operation works under varying load. The crucial issue is to identify this variation. Knowledge about load variability allows making a key decision how to take into account this parameter. Load significantly influences vibration signal. In our case there is no "nominal" load because the load is strongly varying so the diagnosis must be adapted to these particular in-service conditions. It is described by Randall [6] that time varying load causes AM modulation of signal related to mesh component.

Condition Monitoring of gearboxes (especially planetary gearboxes) under time varying non stationary load condition is one of the most important task in many industrial applications (i.e helicopters, mining excavators). Some work that investigates time varying load problem in gearbox diagnostics was done by Stander [1], Bonnardot [2], Baydar [3], Zhan [4] and recently by Bartelmus [5].

To estimate load variation there are a few known approaches.

The easiest is to measure current taken (consuming) by motor driving gearbox. Unfortunately, there are many cases where it is (almost) impossible due technical problems related to organization of such measurements (i.e. high power of signal, U=6kV, I=80-150A). It could be good idea to think about diagnostic system at machine design stage and predict sensor location but that leads to mechatronic systems idea. Moreover, it is often the case that between motor and gearbox is mounted a coupling that could modify relationship between motor current and real load variation

Other interesting approach proposed by Stander [1] is to use demodulation techniques to get information about load variation. According to Randall [6] if load variation is periodic it causes AM modulation of mesh harmonics. Using AM demodulation technique we get back information about load. Load variation affects gearbox vibration signal also in terms of frequency structure. Change of load level for gear pair means change of working condition of motor, too.

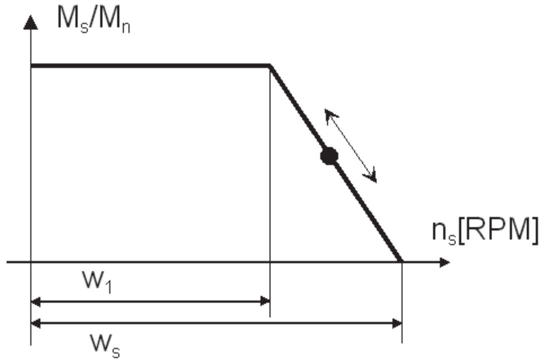


Fig. 1. Electric motor characteristic [7]

As it is presented on Fig.1 change of load causes change of working point on motor characteristic: both input speed and moment will change.

If input speed varies each component in spectrum changes its position on frequency axis. On Fig 2 frequency response function of gearbox housing obtained by impulse excitation is presented. Straight horizontal line is a spectrum of impulse excitation (it is almost flat which means vibration with all possible frequencies in this range can be excited in the same way), arrow (red) symbolize theoretical mesh frequency component. If we take into account frequency response function of gearbox structure and theoretical location of mesh component we could easily find that speed variation can modify amplitude of component (especially if mesh component will meet local resonance as on fig. 2).

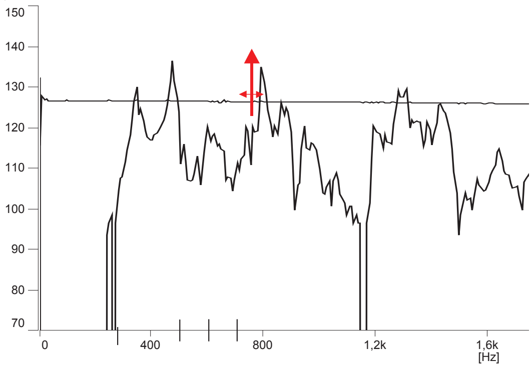


Fig. 2. Frequency response function of gearbox structure obtained by impulse excitation [8]

The crucial question is: how large is input speed variation range and what is a frequency structure of vibration signal generated by gearbox. Usually, for fixed axis gearbox harmonics from 1 to 5 are analyzed and for higher harmonic amplitudes of components decrease. For planetary gearbox we have more complicated situation: there are 8 harmonics and their amplitude almost don't significantly decrease with harmonic number (fig. 3a).

Why do we analyze it?-because higher harmonic has n times wider variation range and it can happen that harmonic of mesh component could match witch resonance frequency. For example harmonic no 7 that is located at resonance area has variation range 100Hz

It leads to conclusion that AM demodulation technique for load estimation is not a necessarily good idea. Moreover, influence of transmission path is not the only problem. It is well known that for multistage gearboxes we can find many sources of AM modulations related to improper work of shafts. After demodulation we need to separate source of modulation of the signals with quite similar frequencies.

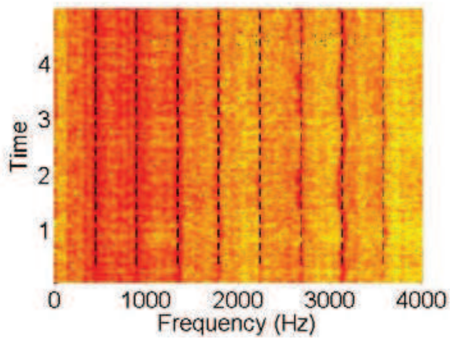
Taking into account that load variation causes also input speed variation we can use this information to estimate load condition. Let us assume that relation moment/speed is linear as it was presented on fig. 1.

How to measure input speed?

Again, the simplest method is to measure rotation of input shaft. It is well known approach that uses optical sensor for speed measurement. Unfortunately it leads to additional hardware requirements (sensor, wires, additional channel) and additionally, in some mechanical devices, rotational speed measurement using a tachometer or a shaft encoder may not be available due to difficult ambient conditions or just shaft is not accessible.

This is the case of the mining system studied here. Therefore the speed needs to be estimated from the vibration measurement recorded on the gearbox casing. This topic will be presented in details in section 3.

a)



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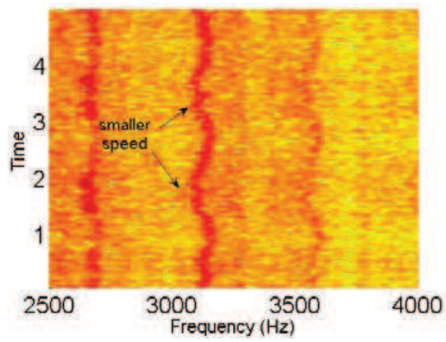

Fig. 3 a) Whole map of time-frequency signal presentation b) restricted map of time frequency signal presentation

2. DESCRIPTION OF THE SYSTEM

The diagnosed device is presented in Fig. 4. (see other work related to object done by Bartelmus and Zimroz [5, 9]).

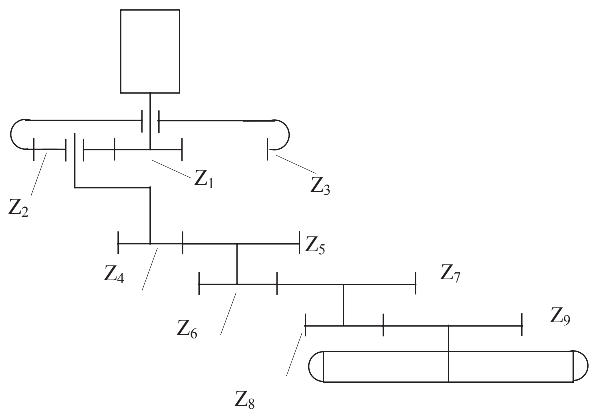

Fig. 4. Driving system for bucket wheel with planetary gearbox (gears: z1 – sun, z2 – planet, z3 – standstill rim, z4 – z9 three stage cylindrical gearbox

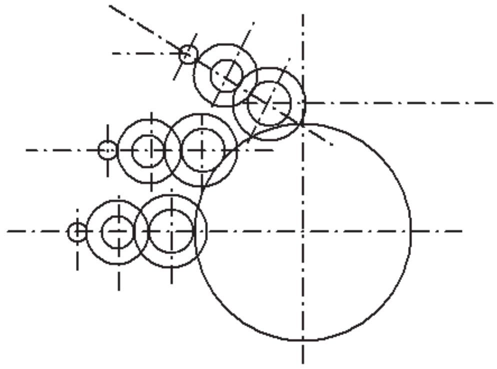

Fig. 5. Three independent drives of bucket wheel

Fig. 4 shows a gearbox whose first stage is a planetary gearbox with a standstill rim. Considered planetary gearboxes consist of: a sun gear z1, planetary gear z2 and rim gear z3. There are three cases of planetary gear element rotation. In the considered case where the sun gear is rotating, the planet gear makes planetary movement and the rim-gear is standstill as it is for the case given in Fig. 4. Taking into consideration some basic parameters of system (design and operation factors [7]) one can calculate meshing and other characteristic frequencies (see table 1).

We are considering the system given in Fig. 4 and we are using notation f12 as meshing frequency for a pair of gear wheels marked in Fig.4 as z1 and z2. Some calculated frequencies are presented in Table1.

These frequencies are calculated to get some knowledge about frequency structure of signal.

Table 1. Characteristic frequencies

Name of component	Frequency value [Hz]	Comments
f12=f23	435	Mesh
f2	11,43	rotation frequency of second gear z2
fa	4,67	Arm frequency
f45	158,7	Meshing frequencies for three stage cylindrical gearbox
f67	57	
f89	13,75	

All calculation has been done for n1=950RPM, where n1 - input rotation velocity

The described driving system Fig. 5 of bucket wheel excavator consists of three subsystem as it is in Fig. 4. The system is used for driving a bucket wheel. The bucket wheel diameter is 17.5m and have 11 evenly distributed buckets of capacity 4.6 m3. The rotation speed of the bucket wheel is about 3RPM and it gives the rotation cycle, 20s. General picture of diagnosed object is presented on Fig. 6.

Fig. 6. Overview of diagnosed system

Two experiments were done for diagnostics purposes. The first was done for gearbox approximately 2 months before its replacement (further "old gearbox"), then second trial was done for new gearbox ("new gearbox"). The reason of replacement was connected with too big noise generated by gearbox but it should be noted that in fact the real detailed report is still unknown because according to warranty conditions gearbox has to be sent to manufacturer.

3. INSTANTANEOUS POWER AND SPEED ESTIMATION FROM VIBRATION MEASUREMENT

3.1. Literature survey

In order to recover the low speed fluctuation from vibration measurement a method has recently been proposed by Bonnardot [2] by using phase demodulation of the mesh frequency (or one harmonic). However limitation of this method is the maximum speed fluctuation allowed which depends on the number of teeth of the gears in mesh (this maximum fluctuation is 0.5% in our case). This method is so restricted to systems under steady loading conditions for which the speed remains approximately constant. This is not the case for the mining gearbox studied here which is subject to important load variations due to the bucketing process. Moreover we aim to monitor not only the low speed variations but also speed variations related to internal components of the gearbox occurring at a higher frequency.

It should be noted that while the instantaneous speed (IS) has a clear mechanical definition, the instantaneous frequency (IF) is more problematic to define (in theory a "pure" frequency is defined on an infinite duration!). A review of instantaneous frequency estimation techniques can be found in [10]. It is shown that the phase difference estimator (phase demodulation of analytic signal and derivation) is optimal for a relatively high SNR (>~16dB). For lower SNR, the noise is amplified due to the derivation and phase unwrapping problems may also arise. In that case other techniques such us peak extraction in time-frequency representations or adaptive techniques (Phase Lock Loop) can be applied.

In the mechanical context of this paper however, the gear mesh vibration generally contains many harmonics, whereas the above IF estimation techniques need to be applied to one particular frequency component only. Since estimation of IF is better for a high modulation frequency, a high rank harmonic of the mesh should be selected. However the higher harmonics are also weaker and consequently have a lower Signal to Noise Ratio (SNR). Moreover it appears that harmonic components can exhibit strong amplitude modulation (AM). For example figure 7 shows evolution of the first 3 harmonics of the planetary gear mesh frequency (~440Hz) on the spectrogram (Short Time Fourier Transform) of the old and new gearbox signals. Strong AM can be observed especially for old gearbox. This AM will highly affect the speed estimation from a given harmonic because its SNR evolves in time. Therefore in this context a new approach must be performed for IS estimation.

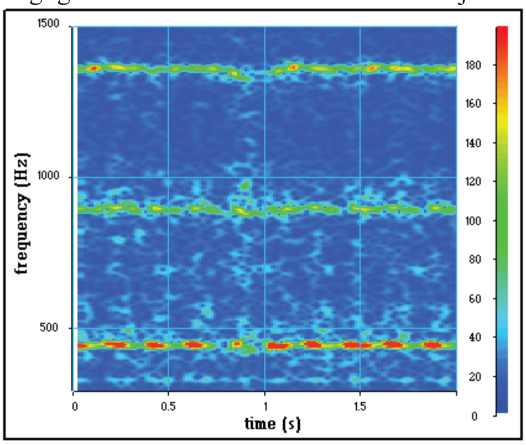

Fig. 7 (a). Old gearbox (Presel_2): spectrogram of part of the signal showing the first 3 harmonics of the planet mesh frequency (440Hz). A strong amplitude modulation is observed for all mesh harmonics

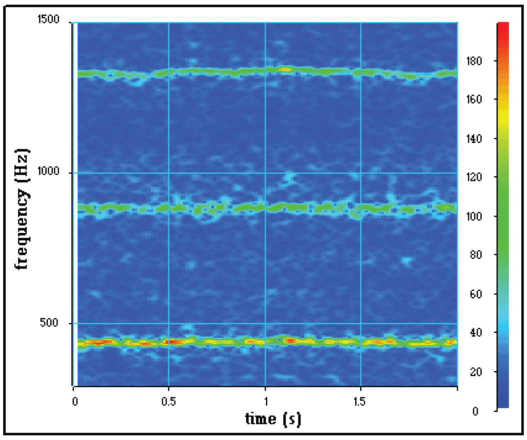

Fig. 7 (b). New gearbox (y12): The spectrogram appears less noisy but amplitude modulation of the mesh harmonic is also present

3.2. Novel approach for instantaneous speed estimation

For a good "robustness" of the instantaneous speed estimation the idea here is to use not only one particular harmonic of the mesh frequency but all harmonic components together. By doing so we will take into account all the information available in the signal.

Suppose we captured two signals x1 and x2 on the same system but at slightly different operating speeds V1 and V2 respectively. Intuitively the effect of a small speed gap between two signals can be approximated by a time scaling, or "stretching" effect. This can be written:

$$x_2(t) = x_1(a(t-\tau)) \tag{1}$$

where a is the time scaling factor and τ the time delay between the signals. The time scaling factor is related to the speed gap by:

$$a = \frac{V2}{V1} = 1 + \frac{\Delta V}{V1} = 1 + \Delta v \tag{2}$$

where Δv is called the relative speed gap between the 2 signals. Note that the time scaling effect in eq. (1) is not exactly true since the measured signal is generally the result of a "source" signal due to mechanical contacts, which is phase-locked with the speed, and then convoluted by the system response or transmission path, which is a feature of the mechanical structure. Relation (1) is however approximately valid provided that the relative speed gap Δv remains small compared to the damping factor(s) of the system response. For example if the most resonant natural frequency in the system hasa 10% damping factor, the time-scaling approximation remains valid for speed variations not higher than about 2%. Under this condition a method has been proposed in order to estimate the time-scaling factor, or relative speed gap, between two signals based on the scale transform [11].

The idea here is to extend this method for estimation of the IS in a signal. In that purpose, the signal is first segmented into overlapping segments, on which a windowing function is applied, and the speed gaps are estimated between them. Similarly to the Short Time Fourier Transform, a trade-off must be achieved for the segment size. Long segments will give a good accuracy of the estimation but speed fluctuations within the segment length will be averaged out. In order to detect non-stationarity events like rapid speed variations, segments must be shortened, which also implies a higher variance of the estimation. Here segment length is set to 0.1s and a Hanning window is applied. The first zero of the corresponding filter is at ~20Hz. Thus speed variations up to the 3rd harmonic of the arm shaft can be captured while variance of estimation remains acceptable.

3.3. Results of instantaneous power and speed estimation

Fig. 8 presents the time waveform and Power Spectral Density for both old and new gearbox signals. The "instantaneous power" (IP) is defined by the RMS value of the signal computed on the same segments as for the speed estimation (IP reflects the global amplitude modulation of the signal). Thus, similarly to the IS, the IP includes all frequency components and reflects the overall amplitude modulation in the signal. A strong power variation appears on fig. 8 (a) for old gearbox at a frequency corresponding to the arm shaft speed (4.7Hz). On the contrary, the IP remains more or less constant for new gearbox.

Results of IS are shown on fig. 9 for both signals. The speed is affected by a low frequency excitation (at ~0.5Hz) due to the outer load variations (bucketing) and by a higher frequency excitation (>4Hz) due to the internal transmission errors in the gearbox. In order to separate the effect of load variations the speed is low-pass filtered at ~3Hz cut-off frequency and the low speed variation is shown on the same figure. It can be compared with the low variations of power obtained by applying the same low-pass filter to the IP.

The Fourier Transform magnitude shows the spectral content of the speed variations. However this may not be appropriate here since speed variations appear rather non-stationary especially for old gearbox. So the time-frequency spectrogram of the IS has been computed and is shown on fig. 10. Non-stationary is evident here especially for old gearbox. Note that second harmonic of the arm component seems to be correlated with load variations, which is not the case for new gearbox.

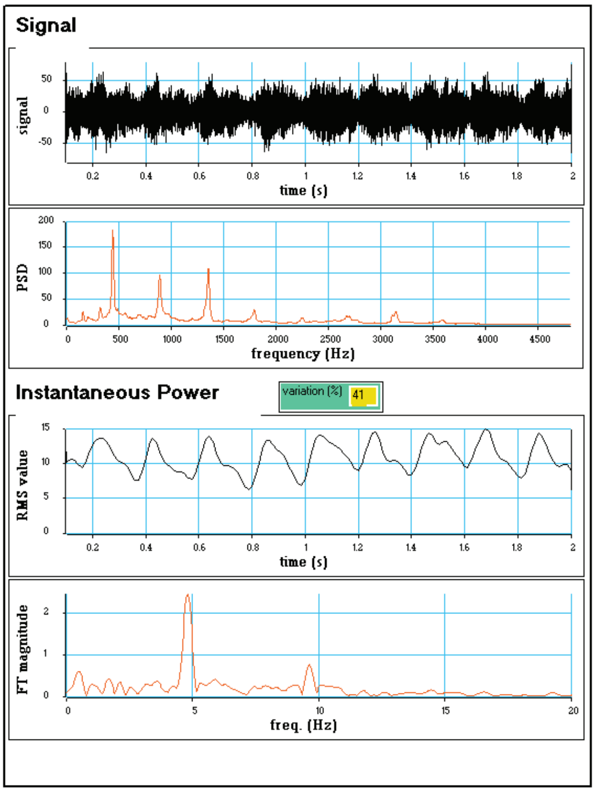

Fig. 8 (a). Old gearbox: signal, Power Spectral Density and instantaneous power (RMS value) computed on each signal segment. A strong power variation appears at the arm shaft speed (4.5Hz)

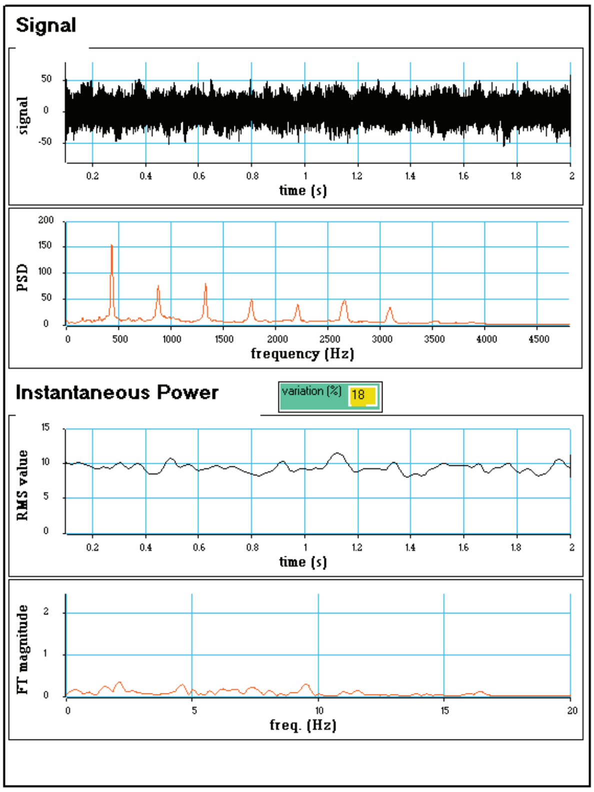

Fig. 8 (b). New gearbox: signal, Power Spectral Density and instantaneous power (RMS value) computed on each signal segment. The signal power remains approximately constant here

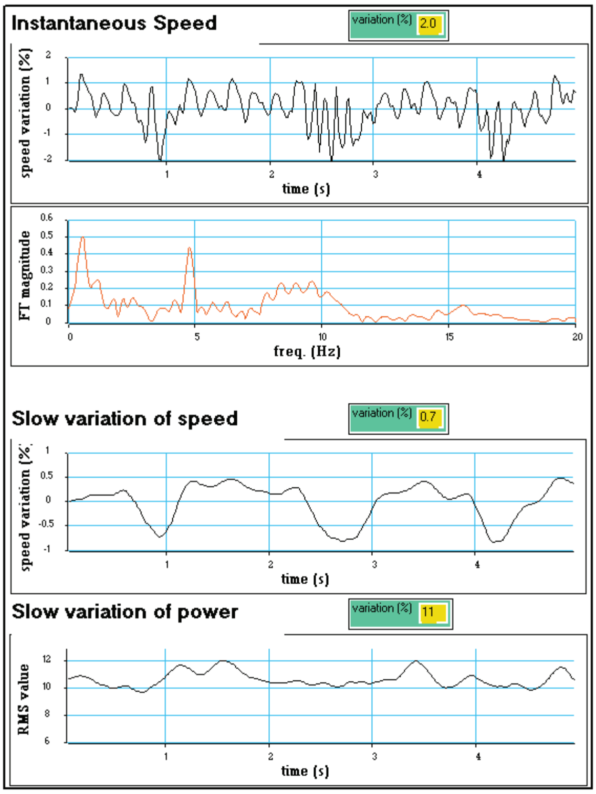

Fig. 9 (a). Old gearbox: instantaneous speed variation and comparison of low speed and power variations. Low variation of speed reflects variations of load. Slow variation of power is noticeable (11%)

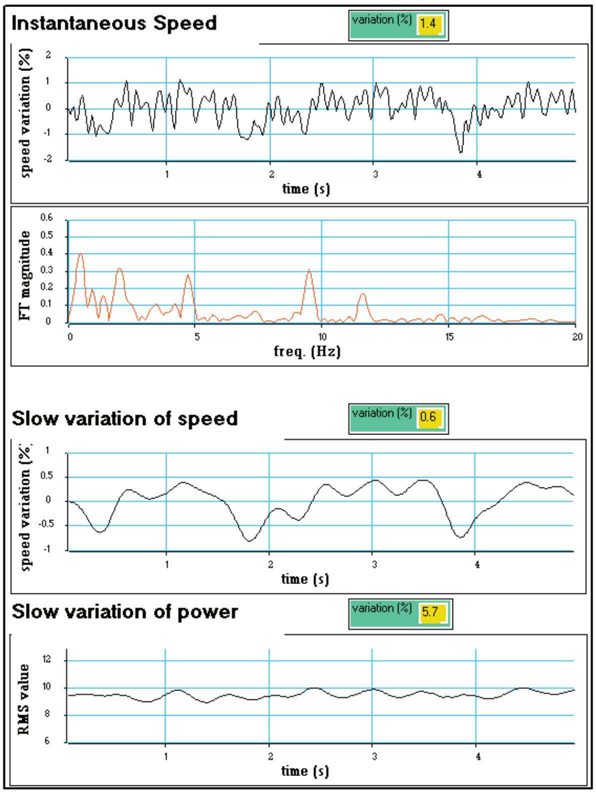

Fig. 9 (b). New gearbox: instantaneous speed variation and comparison of slow speed and power variations. Low speed variation is similar to old gearbox (similar load) but power variation is about twice smaller

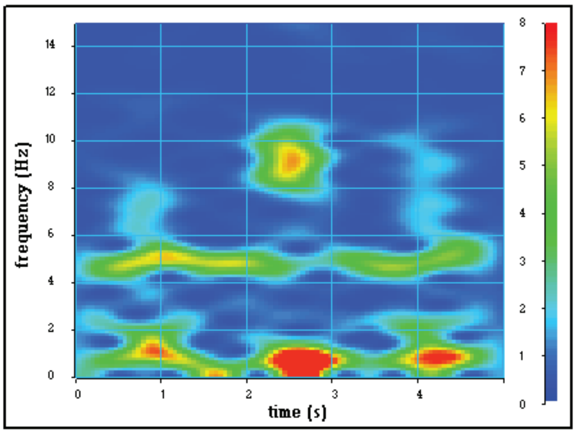

Fig. 10 (a). Old gearbox: spectrogram of IS. Non-stationarity is apparent in the speed: arm shaft 2nd harmonic (~9Hz) seems to be correlated with load variations

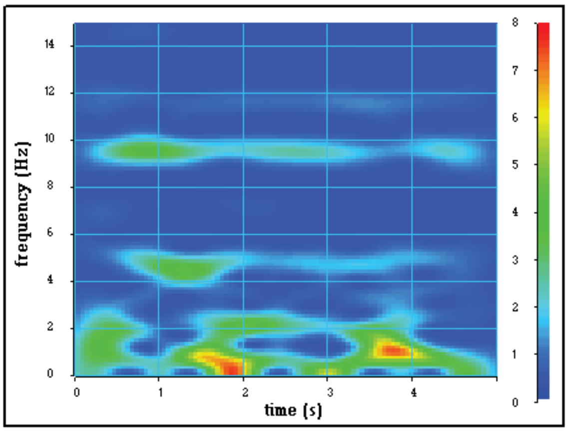

Fig. 10 (b). New gearbox: spectrogram of IS. Speed appears more stationary than for old gearbox and less dependent on load variations. 2nd harmonic of arm shaft speed is also present

Table 2. Comparison of obtained results for good and bad condition of gearbox

	Old gearbox	**New gearbox**
Slow variations (load)		
Instantaneous Power	± 11%	± 5.7%
Instantaneous Speed	± 0.68%	± 0.65%
Arm (4.7Hz)		
Instantaneous Power	strong variation (± 40%)	steady (peak is 12x smaller)
Instantaneous Speed	2nd harm. Dependent on load	stationary arm components

4. DISCUSSION OF RESULTS

Main purpose of this paper was to get information about load variation based on vibration signal processing. We applied two techniques:
important when load is higher (speed is lower), thus indicating a strong "susceptibility" to load variations. The reason for this is suspected to be a backlash in the gear mechanism which has developed due to the wear of the gear/bearing.

Instantaneous Power variation analysis (IP which reflects the global amplitude modulation of the signal) and Instantaneous Speed analysis. Table 2 summarizes the results obtained.

Slow variations at frequencies less than the arm frequency (4.7Hz) are related to load variations. For the old gearbox the IP variation is about twice that for the new gearbox. However the slow IS variation is similar for both gearboxes. As we argue that slow speed variations are linked to load variations, the load variation is so similar for both gearboxes (experimental conditions were approximately the same). The slow IP variation does not seem to be correlated with the load variation.

For higher frequency variations, related to the planetary gearbox components, a strong variation of the IP is detected for old gearbox, thus clearly indicating a problem. The study of the IS shows that while both gearboxes exhibit speed variations at twice the arm shaft speed (second harmonic), this variation remains small and steady for new gearbox, whereas it is highly dependent on the load for old gearbox (fig. 9 (a) & 10 (a)).

This second harmonic component could be an indicator of a slight misalignment in the new gearbox. For old gearbox this component becomes.

A possible reason of such situation could be related with improper arm condition in old gearbox that was detected and diagnosed during analysis.

Taking into account a possible influence of the transmission path on amplitude modulation (fig. 2), IS estimation technique is more reliable than amplitude based approach for outer load estimation.

5. CONCLUSION

The purpose of this paper was to get information about the outer load and speed instantaneous variations based on signal processing of the vibration signal, in order to find indicators that could help for the diagnosis of a planetary gearbox submitted to important load variations. For estimation of the instantaneous speed we have proposed a method that models the speed variation as a local time-scaling factor estimated on each segment of the signal. Its advantage, compared to phase demodulation of one particular mesh harmonic, is to exploit all harmonics of the meshing vibration together, thus providing more robustness in the estimation. We have also defined the instantaneous power as the RMS value of the signal computed on each segment, which reflects the global amplitude modulation of the signal. Both IP and IS have been separated in a low-frequency variation, related to the outer load, and a higher frequency variation related to the planetary stage of the gearbox. The results have been compared between two gearboxes, one being in a presumably faulty condition and a new one after replacement. The study of the IS showed that the higher frequency speed variation remains small and stationary for the new gearbox, whereas it is highly dependent on the load for the used gearbox, thus indicating a backlash due to wear in the gear/bearing mechanism. We conclude that while a fault could be detected from the strong amplitude modulation observed for the used gearbox, the study of the instantaneous speed and outer load variation have helped us to refine the diagnosis of this fault.

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