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Comparison of methods for separating vibration sources in rotating machinery

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ABSTRACT

Vibro-acoustic signatures are widely used for diagnostics of rotating machinery. Vibration based automatic diagnostics systems need to achieve a good separation between signals generated by different sources. The separation task may be challenging, since the effects of the different vibration sources often overlap.

In particular, there is a need to separate between signals related to the natural frequencies of the structure and signals resulting from the rotating components (signal whitening), as well as a need to separate between signals generated by asynchronous components like bearings and signals generated by cyclo-stationary components like gears. Several methods were proposed to achieve the above separation tasks. The present study compares between some of these methods. The paper also presents a new method for whitening, Adaptive Clutter Separation, as well as a new efficient algorithm for dephase, which separates between asynchronous and cyclo-stationary signals. For whitening the study compares between liftering of the high quefrencies and adaptive clutter separation. For separating between the asynchronous and the cyclo-stationary signals the study compares between liftering in the quefrency domain and dephase. The methods are compared using both simulated signals and real data.

1. Vibration signals in rotating machinery

The diagnostics of rotating machinery via vibro-acoustic signatures requires separation of signals excited by different rotating components. Some of the rotating components (e.g. gears) excite high vibration levels that are spread over a wide frequency band. These high wide band vibration levels may mask other rotating components which excite lower level of vibrations. Therefore, in many cases, e.g. bearings, and especially when the diagnostics is done automatically, the separation task cannot be achieved by frequency separation alone. In the case of bearings, the generated signals are relatively weak, often at high harmonics of the bearing tones. These signals may be masked by gear or blade pass frequencies.

A typical vibration signal measured on rotating machinery such as a gearbox or a jet engine is composed of several types of signals excited by the basic rotating components: gears, shafts, bearings, and rotors/blades. These signals arrive to the sensor through the structure of the machine and are therefore convolved by an impulse response function, reflecting the transmission path.

Shafts generate vibrations at several harmonics of the rotating speed with the highest amplitude at the first harmonic. The amplitudes depend on the balancing and alignment of the shaft.

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Gears generate vibrations at several harmonics of the gearmesh rate with a certain amount of amplitude and frequency modulation of both gearwheel rotating speeds. The extent of modulation depends on the gear status. The frequency modulation generates a relatively large amount of sidebands that spread over a wide frequency band.

Rotor blades generate several harmonics of the blade pass frequency which is the harmonic of the shaft speed corresponding to the number of blades on the rotor.

Bearings with localized faults generate impulses for every contact occurrence with the deteriorated surface. The impulses excite resonances of the structure between the bearing and the sensor. The impulse amplitude is modulated when the fault is on a rotating surface that is subject to variations of load and/or changes of the transmission path to the sensor. When no slippage is assumed, the rate of impulse occurrences (kinematic frequencies) can be calculated from the bearing geometry. The kinematic frequencies are non-integer multiples of the rotating speed of the shaft. Due to slippage, the impulse rate may change randomly with about 1–2% tolerances around the kinematic rates. Therefore the signals generated by a faulty bearing are asynchronous to the rotating speed.

The signals generated by shafts, gears and blades will be manifested in the spectrum at discrete frequencies which are integer multiples of the corresponding shaft speed, i.e. synchronous to the rotating speed. All these signals are strong compared to the signals generated by defective bearings.

The measured signal is thus composed of a superposition of signals corresponding to the rotating components multiplied in the frequency domain by a structural transfer function.

2. Separation of discrete frequency noise

In bearing diagnostics, one of the first tasks is to separate the bearing signals from the discrete frequency noise [1–4], i.e. signals generated by shafts, gears, rotors, etc. Several methods were proposed for this task [2]: linear prediction, dephase (removal of the time synchronous average) [1,4], discrete/random separation, and liftering in the quefrency domain. The goal is to remove the synchronous elements of the vibration signal, leaving the asynchronous vibrations, which are usually related to the structure and to the bearings. In [3] the methods for discrete frequency separation were qualitatively compared based on their theoretical characteristics. In the present study the relevant capabilities of the compared algorithms were quantified for scenarios of diagnostics of complex machinery like gearboxes and jet engines.

In this study two methods for separation of discrete frequency noise are compared, dephase and liftering in the quefrency domain [2]. The rationale for selecting these two methods is their ability to remove specific peaks synchronous to specific shafts, in a very controlled manner, while keeping in all the information which is not synchronous to the specific shafts. This ability is crucial when diagnosing very complex machines. In such machines, the richness of excitation sources and the large number of bearings makes the separation in the order or frequency domains problematic, while there is a need to ensure that all the signals belonging to the many bearings remain.

A common practice in bearing analysis is to use envelope demodulation. An important preliminary step in the procedure is filtering the band of frequencies carrying the bearing tones in order to isolate the bearing related phenomena. The determination of the appropriate frequency band requires prior data from all types of damaged bearings which, in many cases, is not available. The two selected methods for separation of discrete frequency noise allows to work around the lack of data problem, and proceed with envelope analysis without bandpass filtering.

2.1. Dephase

The method is based on removal of the phase averaged signal (or the time synchronous average – TSA) [1,4]. The time synchronous average is the mean value of segments representing one rotation of a shaft. The TSA removes the asynchronous components by averaging the resampled signal over a cycle of rotation. All the signal elements that are not in phase with the rotating speed are eliminated, leaving the periodic elements represented in one cycle, i.e. the elements corresponding to harmonics of the shaft rotating speed.

In the cycle domain the STA y_n corresponding to rotation frequency R_1 of the cycle history x is calculated as follows:

$$y_n := \frac{1}{M}\sum_{m=0}^{M-1} x_{n+mN} \quad n = 1, \cdots, N \tag{1}$$

where $N = 1/R1$ and M is the number of cycles in the signal. Note that y is a vector in $\mathbb{R}^N$ representing a single cycle.

The phase average y is replicated in order to get a vector of the same size as x (the resampled signal):

$$z_n = y_{\mathrm{nmod}N} \quad n \in \mathbb{N} \tag{2}$$

Now removing the phase average, thereby generating the dephased signal, removes all the effects which are synchronous with the rotating speed R_1.

$$\tilde{x} = x - z \tag{3}$$

The basic dephase algorithm removes the STA of the entire signal and is therefore appropriate for stationary signals. Its performance depends on the accuracy of the measurement and the accuracy of the rotating speed analysis.

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