Condition monitoring of an epicyclic gearbox at a water power station

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Abstract: Epicyclic gearing or planetary gearing is a gear system that consists of one or more outer gears, or planet gears, revolving around a central, or sun gear. Typically, the planet gears are mounted on a movable arm or carrier, which itself may rotate relative to the sun gear. It is challenging to monitor epicyclic gearboxes, due to their complex structure consisting of many rolling elements. A complex structure and versatile components also result in a long stoppage if a failure occurs. Therefore, it is important to detect incipient faults at an early stage. In this paper, vibration analysis is used for the condition monitoring of an epicyclic gearbox at a water power station. There is a distinct difference between vibration quantities: vibration velocity responds very well to vibrations with frequencies less than 1000 Hz; an even better response is obtained when using acceleration and its higher derivatives, which also provide more information on higher frequencies. Because of the quite high rotational speed of the output, vibration velocity is not good enough for the condition monitoring of the gear in question. Acceleration and its higher order derivatives should be used in order to obtain better responsiveness to changes in the condition of the gearbox. Complex models based on mechanisms are needed in order to calculate the vibration components in the frequency range and to identify the possible faulty components. There was also one vibration component in the gear the source of which could not be discovered with certainty.

Keywords: Condition monitoring; diagnostic; higher order derivatives; epicyclic gearbox.

1. INTRODUCTION

Gear trains are used to transmit motion between shafts. They are critical components in industry and therefore their condition monitoring must be in order, particularly with large gears that can have months of delivery time. The aim of this paper was to develop the current condition monitoring of a two stage epicyclic gearbox at a water power station in order to detect faults at an earlier stage. The previous gearbox suffered a sudden breakdown despite continuous condition monitoring. The breakage was due to the second stage planet gear breaking into half. There were also other faults in the gears, namely impact traces and wearing on the ring gears.

Water power plants are used as an adjusting energy source and thus the load of the gear varies during usage even though the rotation speed stays constant. Depending on energy needs, there are also frequent starts and stops, which stress machines. The condition monitoring of a variable loaded planetary gear, such as that used in a bucket wheel excavator and wind turbines, has been under active investigation (Villa et al., 2012; Bartelmus and Zimroz, 2009a, 2009b; Vicuña, 2010; Hameed et al., 2009; Barszcz and Randall, 2009; Combet and Zimroz, 2009). There are also studies of the modelling of epicyclic gears (Ambarisha and Parker, 2007; Bartelmus et al., 2012). The models based on mechanisms are very detailed.

Different methods for fault detection in epicyclic gears have been investigated. Feng and Zuo (2012) used a common spectral analysis to detect faulty gears in a two stage planetary gearbox. Eltabach et al. (2012) investigated the condition monitoring of a planetary gear in a lifting crane. They suggested using parameters extracted from spectrum, demodulation or cyclostationary analysis to diagnose a fault more accurately. Blunt and Keller used new methods to detect a fatigue crack in the planet gear carrier of a helicopter transmission. The methods worked well under test-cell conditions but failed in low-torque on-aircraft conditions (Blunt and Keller, 2006). Rzeszucinski et al. (2012) presented a new condition indicator based on the amplitude of a probability density function to monitor the health of epicyclic transmissions in helicopters. Wu et al. (2012) studied the characterisation of gear faults in a variable rotating speed using Hilbert-Huang Transform (HHT), and

Heyns et al. (2012) used HHT to compute an envelope of a residual signal er to obtain a discrepancy signal. McFadden (1991) has developed a technique for calculating the time domain averages of the individual planet gears and sun gear.

2. EPICYCLIC GEARBOX

The gearbox under study was a two stage epicyclic gearbox that transforms the water turbine's 87.2 rpm to 750 rpm for the generator. The first stage, i.e. the low speed side, is in the planetary mode and the second stage, i.e. the high speed side, is in the star mode. The structure of the gearbox is presented in Figure 1: the middle section, consisting of a sun gear in the 1st stage and a ring gear in the 2nd stage ring gear, is a floating installation. The gear toothing is double helical.

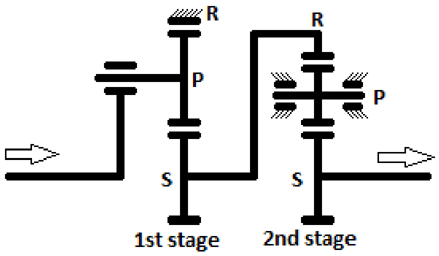

Figure 1. Structure of the two stage epicyclic gearbox.

Epicyclic gears are complex structures. Therefore, it is difficult to calculate their vibration frequencies, and some mistakes in calculations are present in literature (Klein, 2000; Taylor, 2000). In this study, the gear frequencies were calculated with the help of papers (Vicuña, 2010; Mäkikyrö, 2006).

Epicyclic gears can have three different modes on the basis of which the component is fixed, with the exception of a differential gear, in which all the parts are in motion and which therefore needs two drives. The three modes of epicyclic gears are *planetary (pl)*, *star (st)* and *solar (so)*. Table 1 shows these modes in a reduction gear. To calculate failure frequencies, one must first calculate the rotating frequencies of the shafts.

Rotating frequencies for the gear types shown in Table 1 are indicated in Table 2, where f_C is the rotating frequency of a carrier, f_P is the rotating frequency of a planet gear, f_R is the rotating frequency of a ring gear, f_S is the rotating frequency of a sun gear, z_P the number of teeth in a planet gear, z_R the number of teeth in the ring gear and z_S the number of teeth in the sun gear.

The general solution for gear mesh frequency in epicyclic gears is

$$f_m = |f_R - f_C| z_R \,. \qquad (1)$$

A cracked or broken sun gear tooth causes an impact every time a planet gear passes over it.

Table 1. Different types of epicyclic gears.

	fixed component	**driving / driven shaft in reduction gear**
planetary (pl)	*ring gear*	*sun / carrier*
solar (so)	*sun gear*	*ring / carrier*
star (st)	*carrier*	*sun / ring*

Table 2. Rotating frequencies of three types of epicyclic gears.

	fixed component	**driving shaft**	f_{output}	f_P
planetary (pl)	$f_R = 0$	f_S	$f_C = \frac{z_S}{z_S + z_R} f_S$	$\frac{-z_S(z_R - z_P)}{z_P(z_S + z_R)} f_S$
solar(so)	$f_S = 0$	f_R	$f_C = \frac{z_R}{z_S + z_R} f_R$	$\frac{z_R(z_S + z_P)}{z_P(z_S + z_R)} f_R$
star (st)	$f_C = 0$	f_S	$f_R = \frac{-z_S}{z_R} f_S$	$\frac{-z_S}{z_P} f_S$

Therefore, the broken sun gear tooth frequency is:

$$f_{fS} = y|f_S - f_C|, \qquad (2)$$

where *y* is the number of planet gears.

By inserting values from Table 2 to the above formulas, frequencies are obtained for the three modes of epicyclic gears. A cracked or broken planet gear tooth meshes with both the sun and the ring gear and therefore a broken tooth is in contact with other teeth twice per revolution. The frequencies for broken planet gear are

$$f_{fP}^{pl} = 2\frac{z_S z_R}{z_P(z_S + z_R)}|f_S|, \qquad (3)$$

$$f_{fP}^{so} = 2\frac{z_S z_R}{z_P(z_S + z_R)}|f_R|, \qquad (4)$$

$$f_{fP}^{st} = 2\frac{z_S}{z_P}|f_S|. \qquad (5)$$

For ring gear, a cracked or broken tooth gives the following frequencies:

$$f_{fR}^{pl} = y|f_C| = y\frac{z_S}{z_S + z_R}|f_S|, \qquad (6)$$

$$f_{fR}^{so} = y\frac{z_R}{z_S + z_R}|f_R|, \qquad (7)$$

$$f_{fR}^{st} = y|f_R| = y\frac{z_S}{z_R}|f_S|. \qquad (8)$$

The frequencies for the gearbox under study are shown in Table 3.

The impacts caused by a cracked or broken tooth are usually detected with the help of time synchronous averaging or an envelope spectrum. For a single planet gear, however, these methods may not be sufficient, because the vibrations caused by a broken tooth may be hidden under other meshing vibrations. For this, a short signal sample is taken every time a planet gear passes by the vibration sensor. The waveform signal samples are then put in order by taking into account the rotation sequences and averaged. (McFadden, 1991)

Table 3. Gear's rotating frequencies and number of teeth.

	Teeth	**Hz**
1st stage (planetary)		1.45
Gear mesh frequency		132.24
Planet rotation		4.9
Sun gear	35	22.67
Planet gears	27 x 6	9.8
Ring gear	91	8.72
2nd stage (star)		12.5
Gear mesh frequency		450
Planet rotation		18
Sun gear	36	75
Planet gears	25 x 6	36
Ring gear	86	31.4

3. SIGNAL PROCESSING

The weighted l_p norm,

$$\left\|x^{(\alpha)}\right\|_{p,\frac{1}{N}} = \left(\frac{1}{N}\sum_{i=1}^{N}\left|x_i^{(\alpha)}\right|^p\right)^{\frac{1}{p}} = \left(\frac{1}{N}\right)^{\frac{1}{p}}\left\|x^{(\alpha)}\right\|_p, \qquad (9)$$

where all the weight factors are equal to $\frac{1}{N}$, has been proved to be an efficient indicator in condition monitoring (Lahdelma and Juuso, 2011a, 2011b; Lahdelma et al., 2010; Lahdelma and Laurila, 2012).

The measurements were performed with the SKF Microlog Consultant CMXA 48 data collector with SKF CMSS 2111 accelerometers. Sensors were attached to the gearbox with magnets. The upper cutoff frequency of the measurements was 20 kHz for one sensor, 15 kHz for two sensors and 10 kHz for four sensors. At the time of the measurements, generator power was 7.8 – 8.0 MW and water flow in the turbine 143 - 146 m^3/s. Each measurement lasted about 15 seconds. Both the ends of the gearbox were measured in the horizontal and vertical direction. The measurements were analysed by means of the SKF Analysis and Reporting Module program. The turbine rotated at a constant speed of 87.2 rpm.

4. RESULTS

It should be noted that the gearbox under study was a new one, and no measurements from the broken gearbox are presented. Vertical acceleration signals from both the ends of the gear are shown in Figure 2. The levels of the vibrations in the 1st stage are low and no signs of impacts are present. The vibration levels in the 2nd stage are three times higher than in the 1st stage but still very low considering the size of the gear

Figures 3 and 4 show measured amplitude or peak spectra from both the stages of the gearbox in the vertical direction. In both the spectra, the dominating frequency components are at 450 Hz and its multiples and sidebands. This is the gear mesh frequency of the 2nd stage. A common time-domain feature is a peak value obtained as the absolute maximum values of the signal in a chosen sample. The calculated peak values differ in some cases slightly from the actual peaks seen in Figures 3 and 4 because of the frequency resolution of the visualisation program.

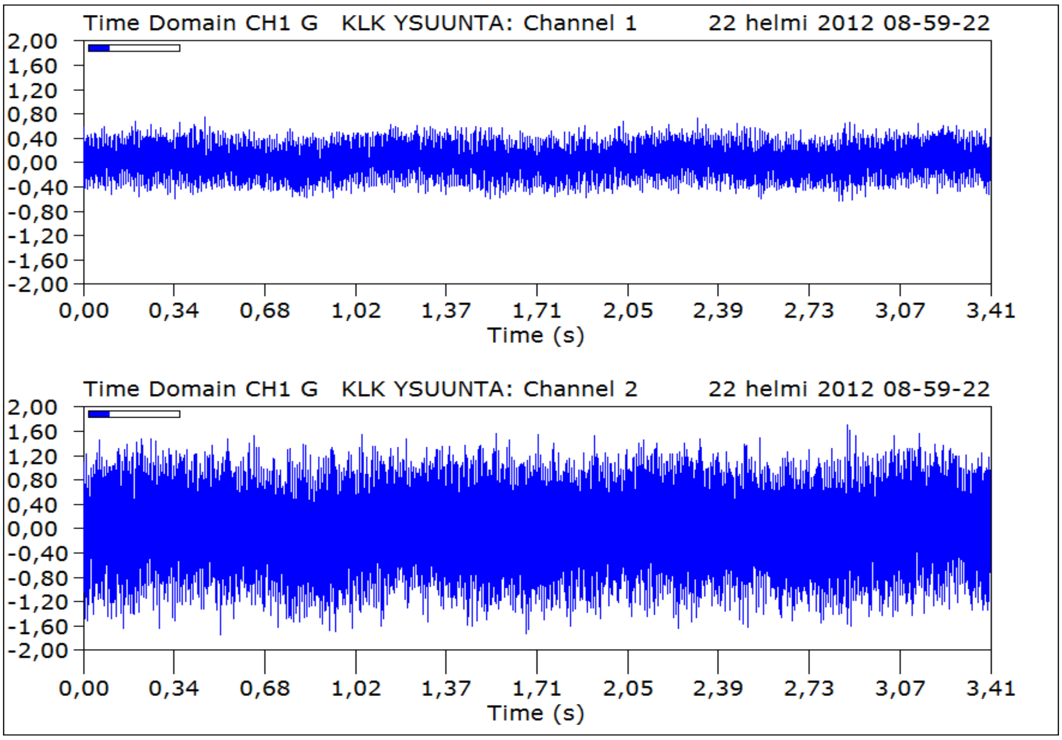

Figure 2. Acceleration signals (g) of the gearbox measured in the vertical direction: channel 1 is the 1st stage and channel 2 the 2nd stage, g=9.80665 m/s^2.

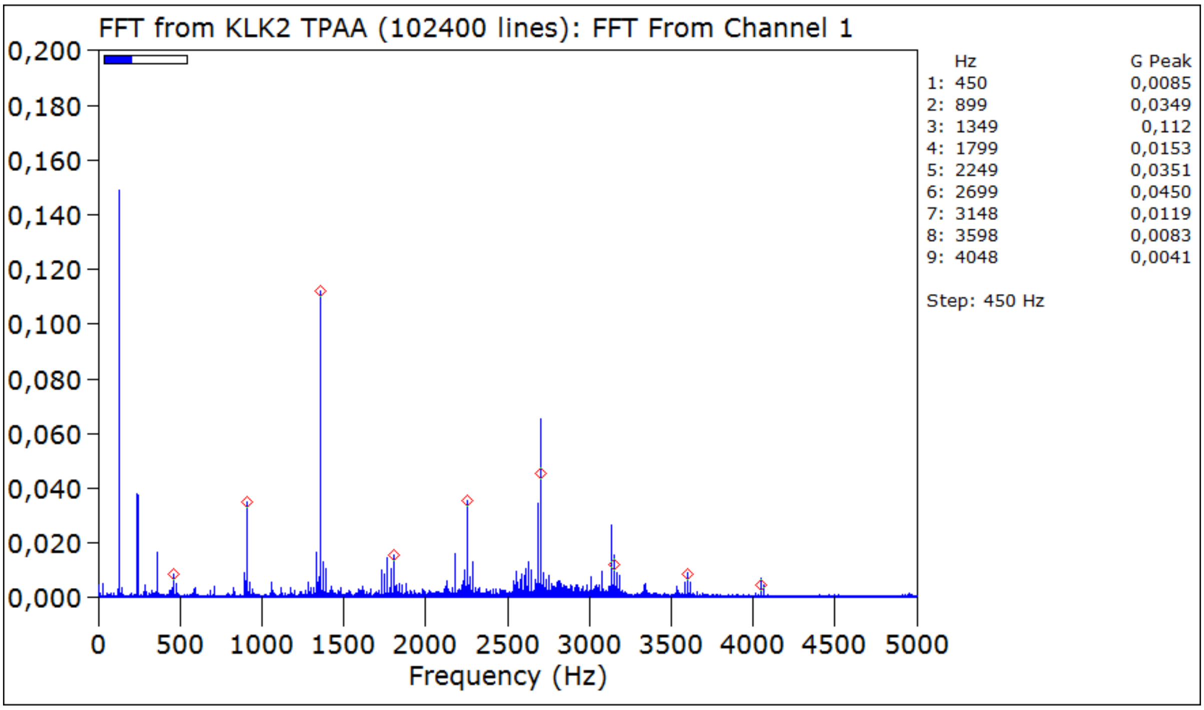

Figure 3. a_{peak} **(g) spectrum from the 1st stage, g=9.80665 m/s^2: the peak value of the gear mesh frequency (450 Hz) and its multiples are denoted by ◊ .**

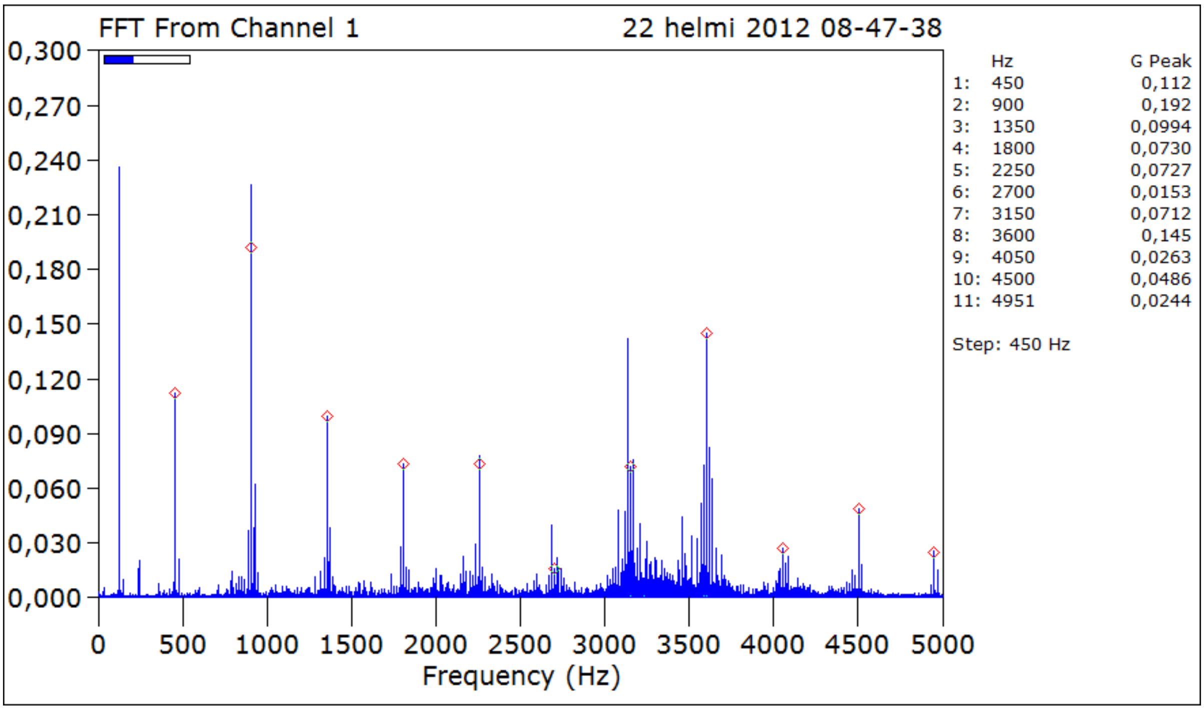

Figure 4. a_{peak} **(g) spectrum from the 2nd stage, g=9.80665 m/s^2: the peak value of the gear mesh frequency (450 Hz) and its multiples are denoted by ◊.**

The sidebands of the 2nd stage gear mesh frequency are 18 Hz apart, which is the same as the rotating frequency of the planets in that stage. There is a strong component at the frequency of 119 Hz, whose second multiple also is visible. This requires further studies since the source is not easy to discover unambiguously.

The 1st stage gear mesh frequency 132 Hz is not visible in the spectrum shown in Figure 3, which is due to the gearbox structure where all the planet gears are in different phases of the mesh. The phase difference is

$$\frac{z_R}{y} = \frac{91}{6} = 15.1667\,, \qquad (10)$$

where z_R is the number of teeth in ring gear and y is the number of planet gears.

The phase difference between planet gears is $\frac{1}{6}2\pi$ and it is evenly distributed in the range [0, 2π]. Therefore, opposite planet gears are in opposite phases and annul each other’s vibrations.

The condition monitoring of the gearbox was conducted earlier by monitoring the root-mean-square velocity (v_{rms}) trend in the frequency range 0 - 1000 Hz and by analysing regularly recorded vibration signals. Figure 5 shows the v_{rms} spectrum and Figure 6 the $\ddot{x}_{peak}$ spectrum from the 2nd stage in the frequency range 0 - 10 kHz in the horizontal direction. They clearly show that v_{rms} alone is not sufficient to monitor this type of gear.

The gear faults can be indicated early with acceleration and its higher derivatives. The velocity emphasises lower frequencies. The higher multiples of the gear mesh frequency are not seen, since all the velocity components in the higher frequencies are negligible (Fig. 5). Acceleration and its higher derivatives within the range 0 - 10 kHz provide substantially more information particularly from the multiples of the gear mesh frequency and their sidebands, see Figures 3, 4 and 6. The higher frequency range also shows possible friction-induced vibration.

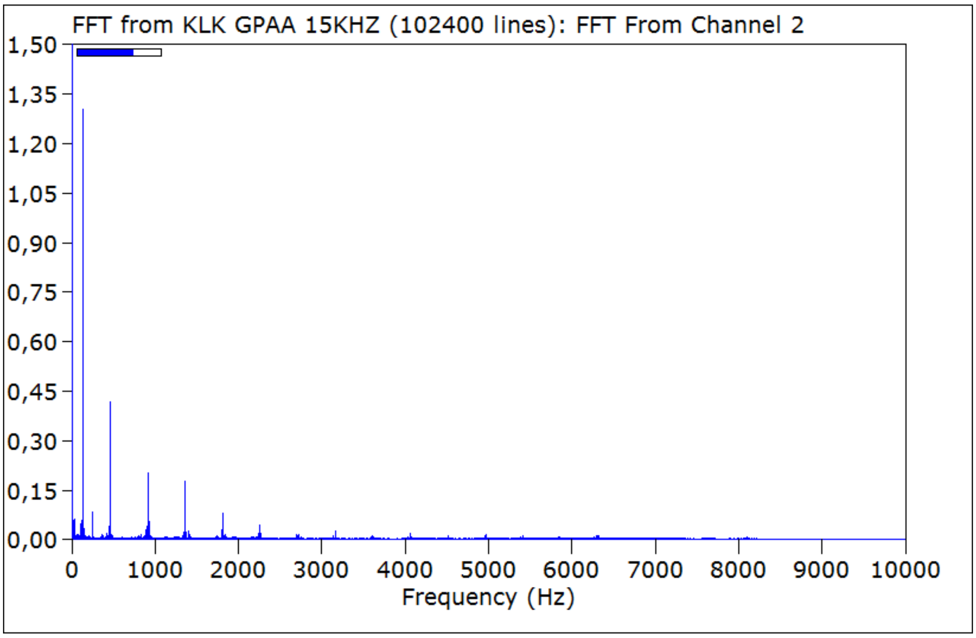

Figure 5. v_{rms} **(mm/s) spectrum from the 2nd stage in the horizontal direction.**

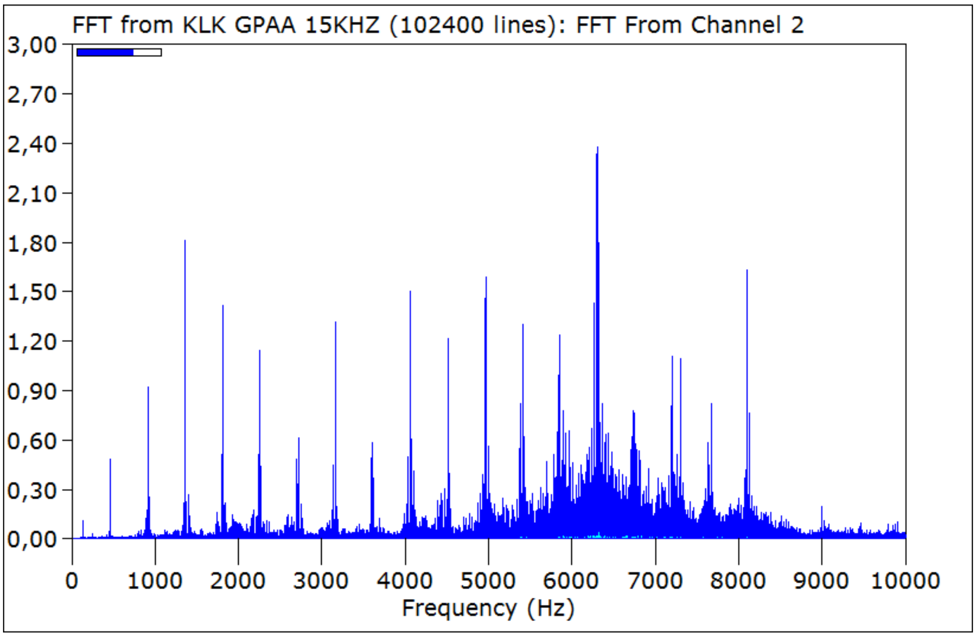

Figure 6. $\dddot{x}_{peak}$ **(kg/s) spectrum from the 2nd stage in the horizontal direction, g=9.80665 m/s^2.**

5. CONCLUSIONS

Vibration velocity responded very well to the vibrations with frequencies less than 1000 Hz. However, an even better response was obtained by acceleration and its higher derivatives, which also produced more information on higher frequencies. Considering the quite high rotational speed of the 2nd stage, it can be concluded that vibration velocity is not sufficient for monitoring the condition of the gearbox in question. Acceleration and its higher order derivatives should also be used in order to obtain better responsiveness for changes in the condition of the gearbox. Future work will include studies of the usability of the weighted l_p norm and real order derivatives $x^{(\alpha)}$ in the condition monitoring of an epicyclic gearbox.

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