Advanced condition monitoring of epicyclic gearboxes

Sulo Lahdelma¹, Esko Juuso² and Jussi Immonen¹
¹Mechatronics and Machine Diagnostics Laboratory, Department of Mechanical Engineering, P.O.Box 4200, FI-90014 University of Oulu, Finland
Phone: +358-294-482083
E-mail: sulo.lahdelma@oulu.fi

²Control Engineering Laboratory, Department of Process and Environmental Engineering, P.O.Box 4300, FI-90014 University of Oulu, Finland
E-mail: esko.juuso@oulu.fi

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. The complex structure consisting of many rolling elements and versatile components results 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. Because of the quite high rotational speed of the output, acceleration and its higher order derivatives should be used in order to obtain good responsiveness to changes in the conditions. 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. A combined signal processing and feature extraction approach based on generalised moments and norms provides efficient indicators for condition monitoring. The features defined by the order of derivation, the order of the norm and the sample time are used in dimensionless indices, which are calculated by dividing the feature value with a reference value corresponding to the normal operation. The indices are calculated for different real order derivatives and weighted $l_p$ norms to form $S$-surfaces, which are used in selecting suitable orders.

Keywords: Condition monitoring, diagnostics, higher, real and fractional 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 (1-7). There are also studies of the modelling of epicyclic gears (8,9). The models based on mechanisms are very detailed.

Different methods for fault detection in epicyclic gears have been investigated. Feng and Zuo (10) used a common spectral analysis to detect faulty gears in a two stage planetary gearbox. Eltabach et al. (11) 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 (12). Rzeszucinski et al. (13) 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. (14) studied the characterisation of gear faults in a variable rotating speed using Hilbert-Huang Transform (HHT), and Heyns et al. (15) used HHT to compute an envelope of a residual signal or to obtain a discrepancy signal. McFadden (16) has developed a technique for calculating the time domain averages of the individual planet gears and sun gear.

Vibration analysis have been used for the condition monitoring of an epicyclic gearbox at a water power station (17). Later the studies were extended to real order derivatives and weighted lp norms, which were used to obtain S-surfaces for selecting suitable orders for the features (18). This paper summarises these two studies.

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 (17)
Epicyclic gears are complex structures. Therefore, it is difficult to calculate their vibration frequencies, and some mistakes in calculations are present in literature (19,20). In this study, the gear frequencies were calculated with the help of papers (4,21,22).

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.

<table>
<thead>
<tr>
<th>different types of epicyclic gears</th>
<th>fixed component</th>
<th>driving/driven shaft in reduction gear</th>
</tr>
</thead>
<tbody>
<tr>
<td>planetary (pl)</td>
<td>ring gear</td>
<td>sun / carrier</td>
</tr>
<tr>
<td>solar (so)</td>
<td>sun gear</td>
<td>ring / carrier</td>
</tr>
<tr>
<td>star (st)</td>
<td>carrier</td>
<td>sun / ring</td>
</tr>
</tbody>
</table>

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, $f_P$ the rotating frequency of a planet gear, $f_R$ the rotating frequency of a ring gear, $f_S$ 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

$$f_m = \left| f_R - f_C \right| z_R.$$

A cracked or broken sun gear tooth causes an impact every time a planet gear passes over it. Therefore, the broken sun gear tooth frequency is:

$$f_{BS} = y \left| f_S - f_C \right|,$$

where $y$ is the number of planet gears. (21, 22)

<table>
<thead>
<tr>
<th>rotating frequencies of three types of epicyclic gears</th>
<th>fixed component</th>
<th>driving shaft</th>
<th>$f_{output}$</th>
<th>$f_P$</th>
</tr>
</thead>
<tbody>
<tr>
<td>planetary (pl)</td>
<td>$f_R = 0$</td>
<td>$f_S$</td>
<td>$f_C = \frac{z_S}{z_S + z_R} f_S - f_S$</td>
<td>$f_R = -\frac{z_S (z_R - z_P)}{z_P (z_S + z_R)} f_S$</td>
</tr>
<tr>
<td>solar (so)</td>
<td>$f_S = 0$</td>
<td>$f_R$</td>
<td>$f_C = \frac{z_R}{z_S + z_R} f_S$</td>
<td>$f_R = \frac{z_R (z_S + z_P)}{z_P (z_S + z_R)} f_R$</td>
</tr>
<tr>
<td>star (st)</td>
<td>$f_C = 0$</td>
<td>$f_S$</td>
<td>$f_R = -\frac{z_S}{z_R} f_S$</td>
<td>$f_R = \frac{z_S f_S}{z_P}$</td>
</tr>
</tbody>
</table>

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\(^{(21, 22)}\):

\[
f_{pl}^{P} = 2 \frac{z_S z_R}{z_P(z_S + z_R)} |f_S| \tag{3}
\]

\[
f_{oo}^{P} = 2 \frac{z_S z_R}{z_P(z_S + z_R)} |f_R| \tag{4}
\]

\[
f_{uu}^{P} = 2 \frac{z_S}{z_P} |f_S| \tag{5}
\]

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

\[
f_{pl}^{R} = y|f_C| = y \frac{z_S}{z_S + z_R} |f_S| \tag{6}
\]

\[
f_{oo}^{R} = y \frac{z_R}{z_S + z_R} |f_R| \tag{7}
\]

\[
f_{uu}^{R} = y|f_R| = y \frac{z_S}{z_R} |f_S| \tag{8}
\]

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.\(^{(16)}\)

Table 3. Gear’s rotating frequencies and number of teeth\(^{(17)}\)

<table>
<thead>
<tr>
<th>Stage</th>
<th>Teeth</th>
<th>Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>1st stage (planetary)</td>
<td></td>
<td>1.45</td>
</tr>
<tr>
<td>Gear mesh frequency</td>
<td></td>
<td>132.24</td>
</tr>
<tr>
<td>Planet rotation</td>
<td></td>
<td>4.9</td>
</tr>
<tr>
<td>Sun gear</td>
<td>35</td>
<td>22.67</td>
</tr>
<tr>
<td>Planet gears</td>
<td>27 x 6</td>
<td>9.8</td>
</tr>
<tr>
<td>Ring gear</td>
<td>91</td>
<td>8.72</td>
</tr>
<tr>
<td>2nd stage (star)</td>
<td></td>
<td>12.5</td>
</tr>
<tr>
<td>Gear mesh frequency</td>
<td></td>
<td>450</td>
</tr>
<tr>
<td>Planet rotation</td>
<td></td>
<td>18</td>
</tr>
<tr>
<td>Sun gear</td>
<td>36</td>
<td>75</td>
</tr>
<tr>
<td>Planet gears</td>
<td>25 x 6</td>
<td>36</td>
</tr>
<tr>
<td>Ring gear</td>
<td>86</td>
<td>31.4</td>
</tr>
</tbody>
</table>
3. Signal Processing

A combined signal processing and feature extraction approach based on generalised moments and norms provides efficient indicators for condition monitoring \( (23,24,25) \). The weighted \( L_p \) norm is

\[
\left\| x^{(a)} \right\|_{p, \frac{1}{N}} = \left( \frac{1}{N} \sum_{i=1}^{N} \left| x^{(a)} \right|^{p} \right)^{\frac{1}{p}} = \left( \frac{1}{N} \right)^{\frac{1}{p}} \left\| x^{(a)} \right\|_{p}, \]

where \( \left\| x^{(a)} \right\|_{p} \) is the classical \( L_p \) norm and all the weight factors are equal to \( \frac{1}{N} \) \( (26) \). The real number \( \alpha \) is the order of derivation, and the real number \( p \) is the order of the norm. Duration of each sample is called sample time, denoted by \( T \). The number of signal values \( N = \tau N_s \), where \( N_s \) is the number of signal values taken in a second.

The signals \( x^{(a)} \) are calculated from the measured acceleration signals \( x^{(2)} \). The rapid changes in acceleration are emphasised upon the derivation of the signal \( x^{(2)} \). The velocity \( x^{(1)} \) and displacement \( x^{(0)} \) are special cases corresponding to \( \alpha = 1 \) and \( \alpha = 0 \). The norm (9) include the norms from the minimum to the maximum, which correspond the orders \( p = -\infty \) and \( p = \infty \), respectively. It includes the absolute mean \( (p = 1) \), and the root mean square (rms) value \( (p = 2) \) as special cases, respectively. Impacts are detected if the orders are sufficiently high, e.g. \( \alpha \in [2,6] \) and \( p \in [4,8] \). On the other hand, unbalance is efficiently detected if the order \( \alpha \in [0,1] \).

The sensitivity of the feature \( \left\| x^{(a)} \right\|_{p, \frac{1}{N}} \) is evaluated as a dimensionless index calculated by dividing the feature value with a reference value corresponding to the normal operation. The indices obtained for different orders \( \alpha \) and \( p \) form a S-surface, which is used in select suitable orders. The surface is flat if the condition is not changed \( (18) \).

The measurements were performed with the SKF Microlog Consultant CMXA 48 data collector with SKF CMSS 2111 accelerometers at Kelukoski power station of Kemijoki Oy. 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.

The measurements have been continued with two remote sensors: Webrosensor WBS CM301 acceleration sensors is connected to internet via WLAN or 3G/GPRS. The sampling frequency is 5 kHz. The \( L_p \) norms were calculated by using Webrosensor DCS software and LabVIEW.
4. Results

It should be noted that the gearbox under study was a new one, and no measurements from the broken gearbox are presented. 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 2 and 3 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. The strongest component is only about 0.24g (2.35 m/s²). 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 2 and 3 because of the frequency resolution of the visualisation program.

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 132.24 Hz is not visible in the spectrum shown in Figure 2, which is due to the gearbox structure where all the planet gears are in different phases of the mesh. The phase difference is

\[
\frac{z_i}{y} = \frac{91}{6} = 15.1667 , \quad \text{.................................................................(10)}
\]
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 $[0, 2\pi]$. 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 4 shows the $v_{rms}$ spectrum and Figure 5 the \( \bar{x}_{peak} \) 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. 4). 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 2, 3 and 5. The higher frequency range also shows possible friction-induced vibration.

![Figure 3. Acceleration (g) amplitude spectrum from the 2nd stage, $g=9.80665$ m/s²: the peak value of the gear mesh frequency (450 Hz) and its multiples are denoted by $\hat{\phi}$](image)
The $l_p$ norms were calculated from the Webrosensor measurements obtained in frequency range 1-1953 Hz. The results are shown in Figures 6 and 7: the order $\alpha$ is in the range from zero to ten and the order $p$ from 0.2 to 8. The step size is 0.2 for the orders $\alpha$ and $p$. During the first period shown in Figure 6, the values of the S-surface are close to one since the power was all the time the same 9.1 MW and the condition of the gearbox had not changed.
The power has a strong effect on the norm values. This was studied by comparing the features calculated from the axial measurements at the power 9.0 MW to the features obtained for the reference power 3.7 MW. The changes are bigger when the order is the range from three to five (Fig. 7). The order $p$ does not have any considerable effect on the S-surface since there are no faults which cause impacts.

Figure 6. S-surface of the slow speed side of the planetary gear at 9.1 MW: measurement interval 15 minutes (18)

Figure 7. S-surface of the slow speed side of the planetary gear: features at 9.0 MW compared with the reference case at 3.7 MW (18)
Partly this may be a consequence of the relatively low frequency range of the measurements. However, the cavitation has also been detected with features based on measurements in rather low frequency range (27,28). S-surfaces can also be constructed to follow operating conditions to provide information about the stress of the machines. Stress indices are efficient tools in improving (29).

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. Generalised moments and norms provide efficient indicators for condition monitoring. The features are defined by the order of derivation, the order of the norm and the sample time. Dimensionless indices are calculated by dividing the feature value with a reference value corresponding to the normal operation. The indices are calculated for different real order derivatives and weighted $l_p$ norms to form S-surfaces, which are used in selecting suitable orders.

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