Separating Different Vibration Sources in Complex Fault Detection

Sulo Lahdelma and Jouni Laurila

Mechatronics and Machine Diagnostics Laboratory, Department of Mechanical Engineering, P.O. Box 4200, FI-90014 University of Oulu, Finland
E-mail: sulo.lahdelma@oulu.fi

Jens Strackeljan and Robert Hein

Institut für Mechanik, Fakultät für Maschinenbau, Otto-von-Guericke-Universität Magdeburg, Universitätsplatz 2, 30106 Magdeburg
E-mail: jens.strackeljan@ovgu.de

Abstract

Any attempt to detect different types of machine faults reliably at an early stage requires advanced signal processing methods. It is well known that vibration measurements provide a good basis for condition monitoring and fault detection. The complexity of signal processing techniques depends on the type of fault. In many cases root mean square and peak values are useful features for fault detection. Unbalance, misalignment, bent shaft, mechanical looseness and some electrical faults, for example, can be detected using features of displacement and velocity. Advanced filter settings and higher order derivatives provide additional possibilities if faults cause high frequency vibrations or impacts. This paper presents results from investigations about various faults, e.g. unbalance, misalignment, resonance, cage fault, absence of lubrication in a ball bearing and their combinations. To detect the faults, also higher order derivatives, $l_p$ norms and dimensionless measurement index MIT are utilised.

Keywords: Condition monitoring, diagnostics, simultaneous faults, feature extraction, higher order derivatives, $l_p$ norms, MIT index

1 Introduction

Different types of faults in machines can cause expensive production losses, especially in capital-intensive plants. To avoid unplanned downtimes the faults have to be detected reliably at an early stage. Advanced signal processing methods are needed for this purpose. Vibration measurements provide a good basis for condition monitoring. Displacement $x = x^{(0)}$ and velocity $x^{(1)}$ are good indicators for many faults, such as unbalance, resonance or bent shaft [1, 2]. However, the drawback of these signals is that they are not
usually sensitive enough to allow the detection of impact-like faults at a sufficiently early stage. Higher, real and complex order derivatives provide additional methods for vibration analysis. The first time derivative of the acceleration signal, the jerk, was used by Smith [3] for examining slowly rotating bearings. The $x^{(3)}$ signal had already been used earlier for assessing the comfort of travel in the design of lifts. Lahdelma introduces in [4] the $x^{(4)}$ signal (napse) in an investigation on the condition of an electric motor. In [5, 6], it was shown in practice that higher order derivatives are very sensitive to impact-like faults.

Statistical features, moments and norms are well established in condition monitoring. The root mean square (rms) and peak values are the most commonly used features in vibration analysis. The generalised moments were presented in [7] and generalised norms in [8]. These features, moments and norms can be combined in a normalised form in order to obtain measurement indices, which are very suitable for machine monitoring. In this investigation, different faults such as rotor unbalance, coupling misalignment, resonance, cage fault, absence of lubrication in a ball bearing and their combinations are studied using a test rig. This paper also extends the field of single fault detection to more complex fault situations, where different faults occur simultaneously and the rotational frequency changes.

2 Test Rig

A test rig (Figure 1), which consists of an electric motor and a belt transmission between two shafts, has been used here to obtain information on different faults. The test rig was originally built by PIM Bt. and later modified in Mechatronics and Machine Diagnostics Laboratory in Oulu. It is a convertible small device with 0.18 kW AC motor. The drive

![Test rig](image-url)
ratio is about 1.21. The ball bearings on the primary and secondary shafts are of type SKF 6002, and there is a fan with 12 blades at the end of the secondary shaft. The motor and the primary shaft are coupled by means of a claw clutch with a four-tooth elastic element. Two rubber elements stacked on top of each other were used in each corner of the foundation plate for the vibration isolation. The same test rig has been used in earlier studies by Juuso et al. [9].

Measurements were carried out using eight accelerometers, a tachometer and a sound level meter simultaneously. The accelerometers types are Wilcoxon Research 726 and IMI 621B51. The first of them has a frequency range (± 3 dB) up to 15 kHz and the second one up to 20 kHz. The accelerometers were screwed directly into the bearing housing in the radial direction, while a glued mounting pad was used in the axial direction. Each of the four bearing housings was measured using a horizontal sensor, and vertical and axial sensors were used on two bearings. The tachometer (Monark Instruments ROS-5) is used to measure the rotational speed of the primary shaft.

Measurements were performed in the LabVIEW environment by means of three analog input modules (NI 9233) with 24 bit A/D converter and anti-alias filtering. Separate modules were connected using a four-slot USB chassis (NI cDAQ-9174), which confirm simultaneous measuring from all the channels. Each combination of a sensor, cable and measurement channel was calibrated using a vibration calibrator (B&K 4294). The Sample rate was 50 kHz and the time period for continuous data collection was about 80 seconds in each fault case. The tests were carried out with 16 different rotational frequencies between 4 - 23.5 Hz, and 12 fault states were executed. In this paper, only 6 different frequencies and signals from two accelerometers were investigated in the following states:

- initial state
- rotor unbalance \( u = mr = 11 \text{ g} \cdot 50.2 \text{ mm} = 552 \text{ gmm} \)
- coupling misalignment 0.35 mm
- rotor unbalance and coupling misalignment
- no lubrication in bearing
- no lubrication in bearing and no cage
- rotor unbalance, coupling misalignment and no lubrication in bearing and no cage.

### 3 Advanced Signal Processing Methods

The increasing requirements for more efficient condition monitoring call for advanced signal processing methods. Several methods have been developed in this field. In this paper, feature extraction is based on the order of derivative, order of moments and dimensionless vibration indices.
3.1 Order of Derivative

Fault detection depends essentially on the order of derivative. Faults such as unbalance, bent shaft, resonance and mechanical looseness can be detected successfully by means of displacement and velocity. Bearing faults, such as outer race, inner race or rolling element faults as well as faults in gears or cavitation can be detected more efficiently with the acceleration signal. The use of higher order derivatives offers more sensitive solutions, especially in the case of impact-like faults. Often, the most sensitive solution cannot be found with an integer order of derivation. Additional flexibility can be achieved with the real order of derivation. For sinusoidal signal \( x = X \sin \omega t \) appears:

\[
\frac{d^\alpha x}{dt^{\alpha}} = \omega^\alpha X \sin \left( \omega t + \alpha \frac{\pi}{2} \right) = X_\alpha \sin \left( \omega t + \varphi_\alpha \right),
\]

where \( \alpha \in \mathbb{R} \) is the order of derivation, the amplitude \( X_\alpha = \omega^\alpha X \), and the change of the phase angle \( \varphi_\alpha = \alpha \frac{\pi}{2} \).

3.2 Moments and Norms

The features are calculated by means of a generalised moment about the origin:

\[
\tau M_p^\alpha = \frac{1}{N} \sum_{i=1}^{N} \left| x^{(\alpha)}_i \right|^p,
\]

where \( \alpha \in \mathbb{R} \) is the order of derivation, \( p \in \mathbb{R} \) is the order of the moment, \( \tau \) is the sample time, i.e. the moment is obtained from the absolute values of signals \( x^{(\alpha)} \). The number of signal values \( N = \tau N_s \) where \( N_s \) is the number of samples per second. Alternatively, the signal values \( x^{(\alpha)}_i \) can be compared to the mean \( \bar{x}^{(\alpha)} \):

\[
\tau M^p_\alpha = \frac{1}{N} \sum_{i=1}^{N} \left| x^{(\alpha)}_i - \bar{x}^{(\alpha)} \right|^p.
\]

A norm can be defined e.g. by

\[
\|\tau M^p_\alpha\|_p = \left( \tau M^p_\alpha \right)^{1/p} = \left( \frac{1}{N} \sum_{i=1}^{N} \left| x^{(\alpha)}_i \right|^p \right)^{1/p},
\]

which is the \( l_p \) norm

\[
\|\tau M^p_\alpha\|_p = \| x^{(\alpha)} \|_p.
\]

This norm has the same dimensions as the corresponding signals \( x^{(\alpha)} \). The \( l_p \) norms are defined in such away that \( 1 \leq p < \infty \). In this study, the order \( p \) is four. The absolute mean,

\[
\|\tau M^4_\alpha\|_1 = x^{(\alpha)}_{\text{av}} = \frac{1}{N} \sum_{i=1}^{N} \left| x^{(\alpha)}_i \right|,
\]
and the rms value,
\[
\| \tau M^p_\alpha \|_2 = x_{\text{rms}}^{(\alpha)} = \left( \frac{1}{N} \sum_{i=1}^{N} |x_i^{(\alpha)}|^2 \right)^{1/2},
\]
are special cases of (4).

### 3.3 Dimensionless Vibration Indices

In [4], Lahdelma introduced the measurement index MIT for estimating machine condition. MIT is an abbreviation of the Finnish word mittaus and means “measurement”. The calculated features are normalised e.g. by division through the initial state in which the machine was in good conditions. Dimensionless vibration indices can be combined in a measurement index
\[
\tau MIT_{\alpha_1,\alpha_2,...,\alpha_n}^{p_1,p_2,...,p_n} = \frac{1}{n} \sum_{i=1}^{n} b_{\alpha_i} \frac{\|x_i^{(\alpha_i)}\|_{p_i}}{\|x_i^{(\alpha_i)}\|_{p_i}^0},
\]
where the norms \(\|x_i^{(\alpha_i)}\|_{p_i}\) are obtained from the signals \(x_i^{(\alpha_i)}\), \(i = 1,...,n\). The index zero denote the machine a in good condition and \(b_{\alpha_i}\) represents a weight factor. This factor allows the rating of individual faults. The sum \(\sum_{i=1}^{n} b_{\alpha_i} = n\). Further investigation can be found in [8, 10].

### 4 Fault Types

Different faults cannot usually be detected reliably by means of one feature only. Based on this, some typical machine faults are simulated and different features are studied in view of the identification of specific faults.

#### 4.1 Initial State

At the beginning of the test series, the bearings of the second shaft were replaced by brand new ball bearings of the type SKF 6002. There were used bearings on the first shaft. For the purpose of the investigation, the signal was filtered by means of a bandpass Butterworth filter of the order 10, and the frequency range was from 3 Hz to 10 kHz. Furthermore, the time signal with a sample length of 4 s and the rms amplitude spectra were used.

Figure 2 shows the \(x^{(0)}\), \(x^{(1)}\), \(x^{(2)}\), \(x^{(3)}\) and \(x^{(4)}\) signals and their frequency spectra for bearing 1 in the resonance case (f=11.6 Hz). In the initial state, the test rig shows small unbalance, which can be seen in the \(x^{(0)}\) and \(x^{(1)}\) signals. Their peak values were 23.3 \(\mu\)m and 2.6 mm/s, respectively. The spectrum of the displacement signal shows only one frequency component corresponding to the rotational frequency and it is only displayed up to 200 Hz, due to higher frequency components were not excited. The peak value of the \(x^{(2)}\) signal is 3.4 m/s\(^2\). In the \(x^{(4)}\) signal there are small impacts, which could not be
seen in the \( x^{(0)} \), \( x^{(1)} \) and \( x^{(2)} \) signals. The higher frequency components can be found at 1.5 kHz, 4.5 kHz and over 8 kHz. The component at 1.5 kHz is 5.5 Mm/s\(^4\).

\[ \text{Figure 2. The } x^{(0)}, x^{(1)}, x^{(2)}, x^{(3)} \text{ and } x^{(4)} \text{ signals and their frequency spectra for bearing 1 in the initial state.} \]
4.2 Unbalance

Rotor unbalance was generated by a mass of 11 g in the disk close to the coupling. The unbalance was 0.552 kgmm. The peak value of the displacement increases from 23.3 \( \mu m \) in the initial state to 140.9 \( \mu m \) as a result of the unbalance (Figure 3). The \( x^{(4)} \) signal and its spectrum show no detectable changes, as compared with the initial state.

![Figure 3](image)

**Figure 3.** The \( x^{(0)} \) and \( x^{(4)} \) signals and their frequency spectra for bearing 1 in the case of rotor unbalance.

4.3 Coupling Misalignment

Coupling misalignment was induced by moving the motor 0.35 mm in the horizontal direction. The measurement was performed by means of two dial gauges (Figure 1). Many vibration diagnostic charts show that displacement and velocity are usually good signals for detecting misalignment. In this case, however, misalignment could not be detected reliably with the help of \( x^{(0)} \) or \( x^{(1)} \) signals (Figure 4), but this could be done with the \( x^{(2)} \), \( x^{(3)} \) and \( x^{(4)} \) signals. Displacement as well as its spectrum do not show any changes as compared with the initial state. Therefore, the signal to be used for detecting misalignment depends on the construction of the coupling, which in this case was a claw clutch. The interval of peaks in the \( x^{(4)} \) signal corresponds the frequency which is equal to twice the rotational frequency. Additionally, major changes could be observed in the frequency spectrum of the \( x^{(4)} \) signal. The maximum component between 4-5 kHz was 15.7 Mm/s\(^4\) and 4.0 Mm/s\(^4\) between 8-9 kHz.
4.4 Coupling Misalignment and Rotor Unbalance

When coupling misalignment and rotor unbalance occur simultaneously, the $x^{(0)}$ signal and its frequency spectrum are dominated by unbalance (Figure 5). This corresponds to the case where only unbalance occurs. On the other hand, the $x^{(4)}$ signal and its frequency spectrum correlate with the case, when only misalignment appears. This means that at this test rig unbalance has no significant influences on misalignment and vice versa.

**Figure 4.** The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra for bearing 1 in the case of coupling misalignment.

**Figure 5.** The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra in the case of coupling misalignment and rotor unbalance for bearing 1.
4.5 No Lubrication in Bearing

All the grease in bearing 3 was washed away using solvent-based liquid (CC Solvent). After start-up, the motor was unable to keep rotational frequency constant, due to the high friction forces in bearing 3. Based on this, the $x(0)$ and $x(4)$ signals in Figure 6 are measured at a rotational frequency of 11.75 Hz instead of 11.6 Hz. The displacement signal and its spectrum show slightly increased values, as compared with the initial state. The component at the rotational frequency grows from 11.7 $\mu$m to 14.3 $\mu$m. However, there are significant changes in the $x(4)$ signal, and its peak value is 38 times higher than in the initial state due to the high friction. Furthermore, major changes can be observed in the high frequency range between 7.5 kHz and 9.5 kHz. The maximum component increases from 1.9 Mm/s$^4$ to 120.7 Mm/s$^4$.

![Figure 6.](image1.png)

Figure 6. The $x(0)$ and $x(4)$ signals and their frequency spectra from bearing 1 in case of no lubrication in bearing 3.

The signals shown in Figure 7 are for bearing 3. The peak value of the displacement signal of bearing 3 is slightly higher than that of bearing 1, but the maximum component in the spectrum remains constant. Moreover, there are some frequency components up to 40 Hz. The peak value of the $x(4)$ signal is 5 times higher than that of bearing 1 and also the maximum component has changed from 120.7 Mm/s$^4$ to 505.5 Mm/s$^4$. Furthermore, two components, at 8 kHz and 9 kHz, dominate the spectrum.
4.6 No Lubrication in Bearing and no Cage

In this case, the cage of bearing 3 was removed, so that two faults, i.e. no lubrication in bearing 3 and no cage occur simultaneously. It could be seen during the measurements that the motor ran more smoothly, owing to the absence of the cage and to the associated increased bearing clearance and as a result of smaller friction forces. The Figure 8 shows, that the displacement signal corresponds to the signal in the initial state. The peak value of the $x^{(4)}$ signal, though, is about 21 times smaller than in Section 4.5, and the spectrum shows that the maximum component is about 9 times smaller.

Figure 7. The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra from bearing 3 in the case of no lubrication in bearing 3.

Figure 8. The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra from bearing 1 in the case of no lubrication and no cage in bearing 3.
Figure 9 shows the $x^{(0)}$ and $x^{(4)}$ signals for bearing 3. The displacement signal shows major variations as compared with the corresponding signal for bearing 1. The spectrum exhibits that vibrations are also induced below rotational frequency. The maximum component is 25.3 $\mu$m at 4.3 Hz. The basic level of the $x^{(4)}$ signal is about 10 times higher than in bearing 1. The frequency components around 9 kHz for bearing 3 and bearing 1 are 81.0 $\text{Mm/s}^4$ and 14.1 $\text{Mm/s}^4$, respectively.

Figure 9. The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra from bearing 3 in the case of no lubrication and no cage in bearing 3.

### 4.7 Rotor Unbalance, Coupling Misalignment, no Lubrication in Bearing and no Cage

Here, all the simulated faults occur simultaneously, except for no lubrication in bearing with cage. The displacement signal in Figure 10 is dominated by unbalance. Compared with the case of unbalance, a slight increase from 88.9 $\mu$m to 98.0 $\mu$m can be recognised in the spectrum. The peaks in the $x^{(4)}$ signal result from coupling misalignment, and the magnitude of the peaks is almost the same as in Figure 4. The peaks are masked by a high basic level, which is mainly caused by the absence of lubrication. The magnitude of the basic level corresponds to the values in Figure 8. The components around 4.5 kHz (Figure 10) are caused by coupling misalignment and between 8-9 kHz by the absence of lubrication.

Figure 11 shows the measurement results for bearing 3. The component of 92.1 $\mu$m in the displacement spectrum is slightly smaller than the corresponding component from bearing 1. Nevertheless, unbalance may be also detected clearly from bearing 3. Moreover, some low frequency components can be observed, due to the absence of lubrication and cage. The basic level in the $x^{(4)}$ signal of bearing 3 is about 10 times higher than that of bearing 1. This can also be observed in the spectra (Figures 10 and 11). Around 9 kHz, the components are 98.3 $\text{Mm/s}^4$ and 11.5 $\text{Mm/s}^4$, respectively.
Figure 10. The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra from bearing 1 in case of rotor unbalance, coupling misalignment, no lubrication and no cage in bearing 3.

Figure 11. The $x^{(0)}$ and $x^{(4)}$ signals and their frequency spectra from bearing 3 in the case of rotor unbalance, coupling misalignment, no lubrication and no cage in bearing 3.

5 Feature Calculation

Many features have been developed for signal analysis. Peak value, rms value, crest factor, kurtosis, moments and $l_p$ norms are only a few examples for feature calculation. Table 1 shows values on some features concerning the sensitivity of fault detection. The $l_p$ norm (4) was studied for $p=1,2,\ldots,8$. The case of $p=4$ denoted as $l_4$ norm offered the
Table 1. Most commonly used features and $l_4$ norms calculated from $x^{(1)}$ and $x^{(2)}$ signals (above) and $x^{(0)}$ and $x^{(4)}$ signals (below) in different fault situations at rotational frequency 11.6 Hz for bearing 1.

<table>
<thead>
<tr>
<th></th>
<th>Peak</th>
<th>RMS</th>
<th>CF</th>
<th>Kurtosis</th>
<th>$l_4$-norm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$x^{(1)}$</td>
<td>$x^{(2)}$</td>
<td>$x^{(1)}$</td>
<td>$x^{(2)}$</td>
<td>$x^{(1)}$</td>
</tr>
<tr>
<td>Initial State</td>
<td>[mm/s]</td>
<td>[m/s²]</td>
<td>[mm/s]</td>
<td>[m/s²]</td>
<td>[ ]</td>
</tr>
<tr>
<td>Unbalance</td>
<td>2.58</td>
<td>3.36</td>
<td>0.97</td>
<td>0.65</td>
<td>2.65</td>
</tr>
<tr>
<td>Misalignment</td>
<td>10.74</td>
<td>3.87</td>
<td>6.86</td>
<td>0.83</td>
<td>1.56</td>
</tr>
<tr>
<td>Unb &amp; Misal</td>
<td>2.70</td>
<td>7.68</td>
<td>1.02</td>
<td>0.90</td>
<td>2.64</td>
</tr>
<tr>
<td>No lubrication</td>
<td>10.23</td>
<td>7.62</td>
<td>6.62</td>
<td>1.01</td>
<td>1.55</td>
</tr>
<tr>
<td>No lubr &amp; no cage</td>
<td>5.08</td>
<td>29.05</td>
<td>1.45</td>
<td>3.21</td>
<td>3.50</td>
</tr>
<tr>
<td>All faults</td>
<td>13.25</td>
<td>12.60</td>
<td>7.87</td>
<td>1.95</td>
<td>1.68</td>
</tr>
</tbody>
</table>

|                  | $x^{(0)}$ | $x^{(4)}$ | $x^{(0)}$ | $x^{(4)}$ | $x^{(0)}$ | $x^{(4)}$ | $x^{(0)}$ | $x^{(4)}$ |
| Initial State    | [µm] | [Mm/s⁴] | [µm] | [Mm/s⁴] | [ ] | [ ] | [µm] | [Mm/s⁴] |
| Unbalance        | 25.31 | 1324.4 | 12.80 | 55.91  | 1.82 | 23.60 | 1.57  | 38.41  | 14.32 | 139.91 |
| Misalignment     | 140.86| 1024.0 | 94.32 | 58.99  | 1.49 | 17.36 | 1.50  | 25.54  | 14.39 | 132.61 |
| Unb & Misal      | 23.23 | 3715.4 | 13.23 | 272.65 | 1.76 | 13.63 | 1.59  | 50.38  | 14.85 | 726.38 |
| No lubrication   | 133.35| 3659.0 | 90.71 | 284.86 | 1.47 | 12.84 | 1.50  | 40.14  | 100.38| 717.04 |
| No lubr & no cage| 27.44 | 50235.7| 15.99 | 2383.03| 1.72 | 21.08 | 1.55  | 48.90  | 17.84 | 6301.86|
| All faults       | 160.20| 3171.8 | 106.94| 296.84 | 1.50 | 10.69 | 1.51  | 9.24   | 118.54| 517.58 |

best compromise in this work regarding the detection of all the different faults. The crest factors (CF) of the $x^{(0)}$ signals are slightly over 1.41, which corresponds to the crest factor of a pure sine wave. The kurtosis values of the $x^{(0)}$ signals are close to 1.5 which is the kurtosis for a pure sine wave. Furthermore, it can be seen that the $x^{(4)}$ signal offers much higher sensitivity than the $x^{(2)}$ signal in the cases of misalignment or the absence of lubrication. In the case of no lubrication, very high values for rms and $l_4$ norm occur, when the $x^{(4)}$ signal is used.

5.1 $l_1$ Norm

The $l_1$ norm was calculated for all the fault types for bearing 1 as well as bearing 3, using (4). Figure 12 a) shows the $l_1$-norm values for all rotational frequencies based on the displacement signal. It can be extracted from the graph that unbalance as well as faults involving unbalance could be detected reliably in the frequency range from 8 Hz to 16 Hz. In resonance, these values are at least 5 times higher than for the other fault types. It is very difficult to detect unbalance in the low rotational frequency range with this method, due to small dynamic forces. Furthermore, it is also difficult to detect unbalance in the high rotational frequency range. At the maximum speed, the ratio between the $l_1$ norm of unbalance and initial state was only 1.50. The other faults could not be detected with sufficient accuracy using $x^{(0)}$.  

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Figure 12. Variation of $l_4$ norm for $x^{(0)}$ and $x^{(4)}$ signals from bearing 1 in different fault states and rotational frequencies.

In Figure 12 b), the detection of different faults is based on the $x^{(4)}$ signal. The graph confirms the results of Section 4.3, i.e. that coupling misalignment can be identified here reliably by means of the $x^{(4)}$ signal. Furthermore, the graph also shows that unbalance does not react to the $x^{(4)}$ signal. This can be seen clearly when comparing the case of unbalance with initial state or misalignment with unbalance and misalignment. The graphs are almost the same. In the case of no lubrication and no cage, the values grow with increased rotational frequency, and a fault can clearly be detected. The highest value is $1880.1 \text{ Mm/s}^4$ occurring in the case of all the faults at maximum rotational frequency.

Figure 12 c) shows that in the case of no lubrication, it makes a major difference weather or not there is a cage in the bearing. In the case with a cage, the maximum value is $6.3 \text{ Gm/s}^4$ in resonance, which is over 12 times higher than in cases without a bearing cage. The bearing fault resulting from the absence of lubrication can be detected very well in the frequency range from 8 Hz to 23.5 Hz.
Figure 13 a) illustrates the $l_4$ norm for bearing 3, based on the displacement signal. In the case of no lubrication and no cage, as well as all the faults, a high value can be observed at the lowest rotational frequency. For the case of no lubrication and no cage, the maximum value of 141.1 $\mu$m for the $l_4$ norm was recorded at 4 Hz, and the values decrease when the rotational frequency grows up. The increased bearing clearance in bearing 3 could be one reason for the high values at low frequencies. Except the case of no lubrication, the bearing faults in Figure 13 a) can be detected reliably with $x^{(0)}$ up to a rotational frequency of 20 Hz, owing to the increased bearing clearance. At higher rotational frequencies the graphs are close to the graph of the initial state.

![Figure 13. Variation of $l_4$ norm $x^{(0)}$ and $x^{(4)}$ signals from bearing 3 in different bearing faults and rotational frequencies.](image)

The results for the $x^{(4)}$ signal are shown in Figure 13 b). The absence of lubrication can be detected reliably at all rotational frequencies. The value in resonance is over 9 times higher in the case of no lubrication than in the case of the other faults. For no lubrication, the values decrease after resonance and increase later after 20 Hz. In the other cases the values increase when the rotational frequency grows.

### 5.2 MIT Index

In the industry, a large number of indicators are usually calculated for the advanced condition monitoring and they have to be analysed, which is very time-consuming and requires highly skilled personnel. The purpose of MIT indices is to obtain one index, which can be used to determine if a machine is in good or bad condition. In [11, 12] were investigated relative norms, which means the same as that the MIT index ((8) in Section 3.3) has only
one term, i.e. \( n=1 \), and besides the rotational frequency was constant. Here, the MIT indices are used for a wide frequency range from 3 Hz to 10 kHz to detect different types of mechanical faults. Therefore, (8) were used with the parameters \( n=2 \), \( \tau=4 \) s, \( \alpha_1=0 \), \( \alpha_2=4 \), \( p_1=p_2=4 \) and \( b_{\alpha_1}=b_{\alpha_2}=1 \). The two parameters \( \alpha_1 \) and \( \alpha_2 \) allow the detection of the most typical faults (cf. Section 3.1). The weight factors \( b_{\alpha_1} \) and \( b_{\alpha_2} \) can be adjusted according to application. The initial state was used as reference for good conditions.

The graph in Figure 14 a) shows the calculated MIT indices for bearing 1 for different fault cases. In the case of unbalance, reliable detection is limited to a rotational frequency range of 8-16 Hz. Misalignment, however, as well as the case where misalignment and unbalance occur simultaneously, can be detected successfully at all rotational frequencies. The case of no lubrication and no cage decrease to a minimum at the resonance frequency, and increase after this point when rotational frequency increases. The highest MIT index is 7.1 and it is obtained in the case of all the faults at 4 Hz.

![MIT indices for bearing 1 and bearing 3](image)

**Figure 14.** MIT indices for bearing 1 (above) and bearing 3 (below) at different fault states and rotational frequencies.

The graph in Figure 14 b) shows the fault states of bearing 3. The maximum MIT index is 71.0 in the case of all the faults. This case and the case of no lubrication and no cage (orange line) are almost the same. This means that the case of all the faults is dominated by the case of no lubrication and no cage. Furthermore, the domination shows how a dry bearing strongly influences the MIT indices. Therefore, fault detection is clearly possible.
The graph in Figure 15 a) shows the calculated MIT indices for bearing 1 in the case of bearing faults in bearing 3. The maximum value is 23.3 occurring in the case of no lubrication at the resonance frequency. The same faults are shown for bearing 3 in the graph below (15 b)). In this case, the maximum value is 173.7. The high values at 4 Hz are possibly caused by impacts of the balls as a consequence of small centrifugal forces and the absence of the cage.

![Graph showing MIT indices for bearing 1 and bearing 3.](image)

**Figure 15.** MIT indices for bearing 1 (above) and bearing 3 (below) in different bearing fault states and rotational frequencies.

The MIT indices may also be used in the industry. Usually the alarm level is adjusted to be 2-3 times than the values of the normal operating state when using $v_{rms}$ measurement, for example. In the Figure 14 a) the chosen lowest limit for the MIT index is two, which is quite low in view of detecting faults at an early stage. When the index is higher than two, the machine operating system issues a warning message. Afterwards, specific parts of the MIT index are to be examined. If the part with $\alpha_1=0$ exceeds a certain limit, the machine should be checked if there is one fault such as unbalance, bent shaft, resonance or mechanical looseness. Otherwise, if the part $\alpha_2=4$ exceeds a certain limit, bearing faults, misalignment or lubrication problems might be the problem and should be checked more carefully. The limits must be adjusted according to the operating conditions. The advantage of the MIT index is that only one number is needed to decide if the machine is in good or bad condition. This is very important, especially when a very high number of sensors have to be observed. The drawback, however, is the need of a reference value. The state of the machine in good condition must be known.
6 Conclusion

Different mechanical faults were investigated in this paper using a test rig. The aim of the study was to prove that there is an effective way of detecting different fault types at an early stage. Higher order derivatives, \(l_p\) norms and dimensionless MIT indices were used for this purpose. The study showed that the most common fault types can be detected using the \(l_p\) norm of the \(x^{(0)}\) and \(x^{(4)}\) signals. Here, the \(x^{(0)}\) signal proved to be a suitable indicator for detecting unbalance, resonance and faults involving unbalance. Misalignment, the absence of lubrication or their combination could be detected reliably with the \(x^{(4)}\) signal. It offered much higher sensitivity regarding absence of lubrication and misalignment than did the \(x^{(2)}\) signal. Therefore, these faults can be detected much earlier by means of the \(x^{(4)}\) signal.

The calculated \(l_p\) norms were normalised using the values of the initial state, and summarised to obtain the dimensionless MIT index. The measurement was performed over a large frequency range from 3 Hz to 10 kHz. All the fault types could be detected reliably with only one index, almost independent from the rotational frequency and the bandwidth, except the case of unbalance, where a reliable detection is limited to the rotational frequency range of 8-16 Hz. The bandwidth used is wide, which offers a simplification of the measurements. The influence of the absence of lubrication was so strong that this fault type could be detected clearly from all bearings. Furthermore, the measurements also indicated, that misalignment cannot always be detected with the \(x^{(0)}\) or \(x^{(1)}\) signals. The required signal depends on the construction of the coupling.

References


