Low Frequency Vibrations analysis
as a method for condition based monitoring system
for railway axles

Doctoral Dissertation of:
Paweł Rolek

Supervisor:
Prof. Stefano Bruni

Tutor:
Prof. Stefano Beretta

The Chair of the Doctoral Program:
Prof. Bianca Maria Colosimo

2015 – cycle XXVII
Safety on European railways is relatively high: it is one of the safest modes of transport in Europe. Even so, it is essential to maintain and improve the current level of safety for the benefit of its users. A safe railway is more efficient and also a more attractive transport choice, which meets the needs of society due to increasing mobility.

Railway vehicle is composed of many different elements, from which some are critical from safety point of view. One of such kind of elements are the axles, thus special attention to their health needs to be paid. They are mechanical components working under effects of rotating bending at high number of cycles. To prevent failure of the axles working under this demanding conditions, periodical inspections needs to be performed. Railway axles in modern vehicles are not only transferring load but also providing support for brakes and other auxiliary systems, thus their proper inspection against faults often requires complete disassembling process and use of one of the non-destructive diagnostic techniques. This approach has significant disadvantages: inspection needs to be performed in workshop which is time, effort and money consuming. Due to the lack of information about the state of the axle during its operation the inspections are scheduled periodically on time or kilometre basis, which can cause either money loss for operator in case of unnecessary check (no problems found) or serious accident in case of axle failure due to overcome its critical condition before planned inspection.

In this work analysis of possible application of ale bending vibration measurement as a method for monitoring the structural integrity of railway axles is presented.

After an examination of the State of The Art, results of full scale tests performed on cracked axles are presented, showing the feasibility of this monitoring approach.

Next, a Finite Element model of the rotating cracked axle is developed and used to investigate the size of recognisable defects together with the effect of various sources of disturbance, namely track irregularity and wheel out-of-roundness.

The final results of the research showed that the increasing vibration signal components (mainly 1xRev and 2xRev) are promising indicators of axle fault development and they can be fruitfully measured and monitored by proposed system.

The increase of vibration components amplitudes was reaching up to ten times the values recorded when no crack was existing in the axle. The level of vibration components and knowledge about possible crack growth rate in railway axle allows to assume that LFV method can provide diagnostic information during service of railway vehicle, also when wheels are undergoing degradation process, thus developing greater out of roundness profiles which are introducing additional excitation in measured system.
Introduction ........................................................................................................................................... 7

1. Non-destructive Fault Detection Techniques for railway wheelsets ........................................... 10
   1.1. Railway axles: role, requirements .......................................................................................... 10
   1.2. Fatigue assessment of railway axles ...................................................................................... 11
   1.3. Railway axles failures .......................................................................................................... 12
   1.4. Defect detection methods in condition based monitoring of railway wheelsets .................. 16

2. Crack-related vibrations in rotating shafts ................................................................................... 23
   2.1. Cracks in rotating shafts ....................................................................................................... 24
      2.1.1. Crack propagation in rotating shafts .............................................................................. 24
      2.1.2. Crack breathing mechanism .......................................................................................... 26
      2.1.3. Dynamic behavior of cracked shafts .............................................................................. 28
   2.2. Crack modelling .................................................................................................................. 30
      2.2.1. Types of crack models used in simulations ..................................................................... 30
      2.2.2. Influence of the crack shape on simulation results ......................................................... 34

3. Laboratory tests .......................................................................................................................... 36
   3.1. Laboratory test of cracked axles ........................................................................................... 36
   3.2. Experiment set-up ............................................................................................................... 36
   3.3. Experiment procedure ......................................................................................................... 38
   3.4. Data acquisition .................................................................................................................. 39
      3.4.1. Sensors in laboratory test ................................................................................................ 39
      3.4.2. Signal processing ......................................................................................................... 41
      3.4.2.1. Anti-aliasing .............................................................................................................. 42
      3.4.2.2. Signal synchronization (triggering) .......................................................................... 43
      3.4.2.3. Synchronous averaging ............................................................................................ 43
      3.4.2.4. Order analysis and windowing .................................................................................... 44
      3.4.2.5. Initial vibration level extraction .................................................................................. 45
      3.4.2.6. Fast Fourier Transform (FFT) .................................................................................... 45
      3.4.3. Signal processing summary ............................................................................................ 46
3.5. Experiment results........................................................................................................................................ 47
3.6. Experiment conclusions.............................................................................................................................. 51

4. Finite Element Model .................................................................................................................................... 52
   4.1. 3D Finite Element Method model ............................................................................................................ 52
   4.2. Geometrical model of railway axle ........................................................................................................... 52
   4.3. Finite element model of cracked railway axle .......................................................................................... 53
      4.3.1. Model definition .................................................................................................................................... 55
      4.3.2. Cracked area modelling ....................................................................................................................... 56
      4.3.3. Boundary conditions ........................................................................................................................... 60
      4.3.4. Load definition ..................................................................................................................................... 61
      4.3.5. Damping in model ............................................................................................................................... 62
   4.4. Simulation scenarios .................................................................................................................................... 64
      4.4.1. Laboratory case setup ........................................................................................................................... 64
      4.4.2. ‘Railway’ case setup ............................................................................................................................ 64
      4.4.2.1. No excitation in model ..................................................................................................................... 65
      4.4.2.2. Wheel out of roundness excitations .................................................................................................. 65
      4.4.2.3. Wheel-rail interaction excitations ...................................................................................................... 69
   4.5. Simulation results ........................................................................................................................................ 71
      4.5.1. Laboratory case simulation results ...................................................................................................... 71
      4.5.2. Railway case simulation results ........................................................................................................... 76

CONCLUSIONS.................................................................................................................................................... 89
ACKNOWLEDGEMENTS ..................................................................................................................................... 92
BIBLIOGRAPHY.................................................................................................................................................. 93
List of figures:

<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>Significant accidents per type of accidents (EU-28:2010-2012) [1]</td>
<td>7</td>
</tr>
<tr>
<td>0.2</td>
<td>Breakdown of number of derailments in Europe into major causes categories [2]</td>
<td>8</td>
</tr>
<tr>
<td>1.1</td>
<td>Types of railway axles</td>
<td>11</td>
</tr>
<tr>
<td>1.2</td>
<td>Different kind of damaging of an in-service railway axle</td>
<td>13</td>
</tr>
<tr>
<td>1.3</td>
<td>Crack growth rate due to impact [15]</td>
<td>14</td>
</tr>
<tr>
<td>1.4</td>
<td>Crack growth models due to combined action of corrosion fatigue</td>
<td>14</td>
</tr>
<tr>
<td>1.5</td>
<td>Example of POD curve for hollow axle inspection (10% threshold) [23]</td>
<td>15</td>
</tr>
<tr>
<td>1.6</td>
<td>Simple definition of the inspection interval, from the crack growth simulation [11]</td>
<td>15</td>
</tr>
<tr>
<td>1.7</td>
<td>Railway axle failure [3]</td>
<td>16</td>
</tr>
<tr>
<td>1.8</td>
<td>Schematic representation of Phased Array Probe</td>
<td>18</td>
</tr>
<tr>
<td>1.9</td>
<td>Phased array working principle</td>
<td>19</td>
</tr>
<tr>
<td>1.10</td>
<td>Principle of the conical phased array probe</td>
<td>19</td>
</tr>
<tr>
<td>1.11</td>
<td>AC thermography method [33]</td>
<td>20</td>
</tr>
<tr>
<td>1.12</td>
<td>AE theory principle</td>
<td>21</td>
</tr>
<tr>
<td>1.13</td>
<td>AE application in railway axle crack detection tests</td>
<td>21</td>
</tr>
<tr>
<td>1.14</td>
<td>High Frequency Response of cracked axles</td>
<td>22</td>
</tr>
<tr>
<td>2.1</td>
<td>Cracked axle cross section with visible crack rest lines</td>
<td>25</td>
</tr>
<tr>
<td>2.2</td>
<td>Vibration signal amplitudes of shaft with propagating crack [51]</td>
<td>26</td>
</tr>
<tr>
<td>2.3</td>
<td>Area in contact between crack surfaces as a function of axle rotation</td>
<td>27</td>
</tr>
<tr>
<td>2.4</td>
<td>First and Second harmonic component of cracked shaft vibration signal [66]</td>
<td>29</td>
</tr>
<tr>
<td>2.5</td>
<td>Vertical displacement as function of cracked beam angular position [68]</td>
<td>32</td>
</tr>
<tr>
<td>2.6</td>
<td>Vertical displacement of cracked shaft, comparison between measured and calculated deflections [68]</td>
<td>33</td>
</tr>
<tr>
<td>2.7</td>
<td>Lateral deflection of 30% cracked shaft with different shear loads applied [68]</td>
<td>33</td>
</tr>
<tr>
<td>2.8</td>
<td>Vertical vibration components dependence on crack depth [68]</td>
<td>34</td>
</tr>
<tr>
<td>2.9</td>
<td>Vibration signal harmonic components in case of rectilinear crack shape [68]</td>
<td>35</td>
</tr>
<tr>
<td>2.10</td>
<td>Vibration signal harmonic components in case of elliptical crack shape [68]</td>
<td>35</td>
</tr>
<tr>
<td>3.1</td>
<td>Dynamic Test Bench for Railway Axles</td>
<td>37</td>
</tr>
<tr>
<td>3.2</td>
<td>Drawing of full-scale specimen example</td>
<td>37</td>
</tr>
<tr>
<td>3.3</td>
<td>Scheme of the full-scale axle equipped with transducers</td>
<td>38</td>
</tr>
<tr>
<td>3.4</td>
<td>Laser sensor head and main control unit</td>
<td>40</td>
</tr>
<tr>
<td>3.5</td>
<td>Data acquisition system</td>
<td>41</td>
</tr>
<tr>
<td>3.6</td>
<td>Input signal not periodic in time record</td>
<td>43</td>
</tr>
<tr>
<td>3.7</td>
<td>Input signal periodic in time record</td>
<td>43</td>
</tr>
<tr>
<td>3.8</td>
<td>Axle speed determination using once per revolution tachometer pulse</td>
<td>42</td>
</tr>
<tr>
<td>3.9</td>
<td>Synchronous averaging</td>
<td>44</td>
</tr>
<tr>
<td>3.10</td>
<td>Signal processing scheme</td>
<td>46</td>
</tr>
<tr>
<td>3.11</td>
<td>Polar plot of vibration from one revolution of axle and its corresponding spectra</td>
<td>47</td>
</tr>
<tr>
<td>3.12</td>
<td>Comparison of harmonics trends of vibration signals from axles tests</td>
<td>49</td>
</tr>
<tr>
<td>3.13</td>
<td>1xRev (first order) harmonic comparison</td>
<td>50</td>
</tr>
<tr>
<td>3.14</td>
<td>2xRev (second order) harmonic comparison</td>
<td>50</td>
</tr>
<tr>
<td>3.15</td>
<td>Harmonics of vibration signal from axle with no fault development during test</td>
<td>51</td>
</tr>
<tr>
<td>4.1</td>
<td>Geometrical model of railway axle</td>
<td>53</td>
</tr>
<tr>
<td>4.2</td>
<td>Full and reduced integration method in 8-nodes element [93]</td>
<td>53</td>
</tr>
<tr>
<td>4.3</td>
<td>Hourglass effect in reduced integration element type due to applied moment [94]</td>
<td>54</td>
</tr>
<tr>
<td>4.4</td>
<td>Hourglass control energy verification</td>
<td>55</td>
</tr>
<tr>
<td>4.5</td>
<td>Finite Element model of railway axle</td>
<td>55</td>
</tr>
<tr>
<td>4.6</td>
<td>Partitioning of cracked axle model</td>
<td>56</td>
</tr>
<tr>
<td>4.7</td>
<td>Reduced possibility of master-slave surface penetration in S-to-S contact type [98]</td>
<td>57</td>
</tr>
<tr>
<td>4.8</td>
<td>‘Hard contact’ formulation for pressure-overclosure behaviour [98]</td>
<td>58</td>
</tr>
<tr>
<td>4.9</td>
<td>Friction model in contact area [99]</td>
<td>59</td>
</tr>
<tr>
<td>4.10</td>
<td>Specimen in crack opening situation (100x magnified in crack opening direction)</td>
<td>59</td>
</tr>
<tr>
<td>4.11</td>
<td>Closed crack and resulting contact pressure area</td>
<td>60</td>
</tr>
<tr>
<td>4.12</td>
<td>Additional axle deflection due to influence of crack breathing mechanism</td>
<td>60</td>
</tr>
<tr>
<td>4.13</td>
<td>Boundary conditions defined in model</td>
<td>61</td>
</tr>
<tr>
<td>4.14</td>
<td>Load definition in model</td>
<td>62</td>
</tr>
<tr>
<td>4.15</td>
<td>Rayleigh damping coefficients and resulting damping ratio</td>
<td>63</td>
</tr>
</tbody>
</table>
Figure 4.16 Influence of damping on axle vibration decay..........................................................63
Figure 4.17 'Laboratory case' model definition scheme..............................................................64
Figure 4.18 'Railway' case model definition scheme.................................................................65
Figure 4.19 New wheel OOR profile versus wheel circumferential coordinate..........................67
Figure 4.20 Extended wheel OOR profile and its corresponding spectra ......................................67
Figure 4.21 Worn wheel out of roundness profile ......................................................................68
Figure 4.22 Extended worn wheel OOR profile (upper) and its corresponding spectra (lower) ....68
Figure 4.23 Rail irregularity profile ............................................................................................69
Figure 4.24 Common excitation for wheel and rail used in simulation .........................................70
Figure 4.25 Non-cracked axle deflections and its corresponding harmonic components .............71
Figure 4.26 Deflection and spectra components for defined crack depth ...................................72
Figure 4.27 Harmonic components amplitude trend as function of crack depth .......................75
Figure 4.28 Vertical displacement components (FFT) for 35% crack, no external source of excitation in model ........................................................................................................76
Figure 4.29 Harmonic components (FFT) of axle vertical vibrations influenced by 'new' OOR profiles .77
Figure 4.30 Comparison of harmonic components amplitudes ('new' wheel OOR) .....................78
Figure 4.31 Harmonic components (FFT) of axle vertical vibrations influenced by 'worn' OOR profiles .79
Figure 4.32 Comparison of harmonic components amplitudes ('worn' wheel OOR profile) ........80
Figure 4.33 Axle vertical vibration from 'Mixed case' simulation results (under 'new' wheel OOR and rail profile influence) ......................................................................................81
Figure 4.34 Axle vertical vibration from 'Mixed case' simulation results (under 'new' OOR profile, rail profile and 35% crack depth influence) ...............................................................82
Figure 4.35 Comparison of harmonic components amplitudes ('new' wheel OOR and rail irregularities included) ........................................................................................................83
Figure 4.36 Axle vertical vibration from 'Mixed case' simulation results (under 'worn' OOR profile and rail profile influence) ......................................................................................84
Figure 4.37 Axle vertical vibration from 'Mixed case' simulation results (under 'worn' OOR profile, rail profile and 35% crack depth influence) ...............................................................85
Figure 4.38 Comparison of harmonic components amplitudes ('worn' wheel OOR and rail irregularities included) ........................................................................................................86
Figure 4.39 FFT of axle vertical vibration from 'Mixed case' simulation results including 'worn' OOR profile, rail irregularity profile and 30% deep crack influence .......................................87
Figure 4.40 Change in NxRev components due to crack size (excitations from 'worn' wheel OOR and rail irregularities included) .....................................................................................87
Safety on European railways is relatively high allowing to formulate the statement that it is one of the safest modes of transport in Europe [1].

Even so, it is essential to maintain and even improve the current level of safety for the benefit of its users and to meet the European rail network strategy of targeting considerable expansion of passenger and freight traffic by year 2020.

According to the common safety indicators data, railway safety continued to improve across the EU, but despite a general improvement, there has been no progress in reducing the number of several types of accidents, from which 5% of total accidents is tied up to train collisions and derailments [1]. As European Railway Agency reports, the number of collisions and derailsments is the highest in the three-year period for which fully comparable data are available, giving as example almost 100 cases of each in 2012 year (Figure 0.1). It means that on average a derailment or a collision is reported at least every second day in the EU, causing significant disruptions to railway operations.

![Figure 0.1 Significant accidents per type of accidents (EU-28:2010-2012)[1]](image)

As far as derailments are under consideration, interesting results can be seen after analysing reports from accidents and breaking down derailment causes into their major categories. As analysis of harmonised data from D-RAIL project database done by [2] shows, within Europe, rolling stock causes are responsible for 38% of major freight train derailments, followed by infrastructure (34%), and operations (22%) (Figure 0.2).

As a result of European Union regulations, also events which could lead to accident have to be reported by EU members. These events are called 'precursors to accidents' and are a valuable source of information which serves as input for proactive safety management system. By looking at analysed data from reported precursors considering broken axles and wheels just in 2012 total of 104 reports indicating this accident near-miss has been registered showing significant increase compared to previous years (61 in
2011 and 95 in 2010)[1]. Broken wheels and axles are not only a threat in EU, i.e. in Canada itself there have been at least 29 axle fatigue-related failures between 2000 and 2010 [3].

Figure 0.2 Breakdown of number of derailments in Europe into major causes categories [2]

As can be seen, the wheelset is a crucial component which failure can lead to train derailment causing losses in transported goods or in worst case even fatalities, thus the axle life itself plays key role both the safety and economic performance of the vehicles.

Due the fact that axle deteriorates through its lifetime by means of fatigue and corrosion mechanisms, periodic inspection is used to ensure that these mechanisms have not compromised the axle safety. Inspection of railway axles is performed in periodic manner, based on distance travelled by vehicle in which the axle is operating. The most popular inspection methods are magnetic particle tests for surface defects investigation and ultrasonic tests for internal defects. These types of inspections in most of cases require dismounting of auxiliary components attached to axle to gain access to its critical areas. As consequence, this inspection takes a vehicle out of service and has a significant impact on the economic aspects of train operation.

Apart from economic losses due to service operations there is a safety uncertainty coming from the fact that axle inspections are planned in periodical manner. According to research and accidents investigation results, some factors like corrosion pits and track ballast impacts, can cause crack initiation and propagation process leading to axle failure in reduced time. As an example of the importance of proper estimation of axle condition can be the one coming from derailments cause analysis conducted within D-Rail project [4], in which authors are putting axles failure (including bearing, axle and journal failures, and hot axle boxes) at the top of the ranking of major derailments causes in Europe (i.e. wheel failure was on the third position in this ranking).

Taking into consideration all abovementioned facts, need for development of online system capable of monitoring crucial safety elements in railway vehicle is evident.

Considering actual state of the art technologies in railway axles inspections, as well as developments in other fields of industry new approach of monitoring health parameters of
railway axle during its operation is proposed. The methodology for estimating the condition of axle is based on so called breathing mechanism of eventual crack, which is affecting the bending stiffness and thus generating additional vibration due to rotation of the axle during operation. Specific patterns of this vibration, namely NxRev components (harmonic components at frequencies that are integer multiples of the frequency of rotation) can be used for the fault detection purposes, leading to the continuous structural health monitoring of the axle.

More detailed theoretical considerations, as well as feasibility study for applying LFV methodology in railway axle monitoring system will be closer described in following chapters of this work:

**Chapter 1**: In this chapter role of railway wheelsets and basic concepts of design requirements are briefly introduced. Next, railway wheelsets monitoring systems are listed and briefly described to give a reader insight of currently leading edge technology of non-destructive testing.

**Chapter 2**: Here vibrations of rotating shafts are introduced, focusing on crack occurrence and its influence on specific vibration spectra components. The so called crack breathing mechanism is explained and different attempts to model it numerically to gain proper diagnostic information about shaft health are presented.

**Chapter 3**: This chapter covers detailed description of preparation and conducting laboratory tests of different types of cracked and non-cracked railway axles for assessment of low frequency vibration analysis technique for fault detection system in railway axles. Laboratory test results are also presented and discussion on their relevance is included.

**Chapter 4**: In this chapter detailed description of Finite element Model of cracked railway axle is included, different simulation scenarios are explained and simulation results are presented.

Concluding discussion and comparison between experimental results and simulation results are presented in **Conclusions** section at the end of work.
Non-destructive Fault Detection Techniques for railway wheelsets

1.1. Railway axles: role, requirements

Railway axle is a mechanical component which in combination with wheels is creating substructure named wheelset. Wheelset is providing guidance for railway vehicle, but also needs to sustain loads coming from engines torque (in case of powered axle), braking forces, train-track interaction and vehicle weight. All abovementioned load spectra are causing cyclic loading of railway axle which can lead to its failure due to fatigue.

Railway axles, as they are integral part of railway wheelset, are safety critical components. Their failure may result in derailments, causing serious damage for the rolling stock and the infrastructure, injuries of passengers and in the most serious cases can even lead to fatalities. Therefore, axle resistance to failure is a key issue in designing and correctly maintaining railway vehicles, to ensure high safety standards and, at the same time, to optimize life-cycle costs from a system point of view.

Even if high safety level is provided by the present practices of railway operation, a continuous improvement of the safety is always recommended. Due to this fact the design of axles was since decades approved and always adjusted according the newest State of the Art.

Modern axles are produced mostly from two types of material, namely A1N low strength steel grade (normalized C40 carbon steel) and medium strength A4T steel grade (quenched 25CrMo4 alloy), with strictly described parameters [5]. Nevertheless, due to increased demand of weight reduction and improved durability new materials for axle production are developed and introduced in operation (ex. 30NiCrMoV12 alloyed steel used in some Italian high speed train axles production [6]).

Two main different types of axles can be found on modern trains, axles with bore, named hollow axles and axles with continuous cross section, named solid axles (Figure 1.1) Railway axles are designed, manufactured and maintained so that they should not fail in service, usually targeting up to 40 years of service, or $10^7$ km.
1.2. Fatigue assessment of railway axles

Recent reports on development on railway safety are showing that the safety of railway system has continuously improved [1]. However it is necessary that safety critical subsystems or components, such as wheelsets and axles, are properly addressed in area of safety requirements [7]. Therefore, axle resistance to failure is a key factor in process of designing and maintaining railway vehicles, which ensures high safety along with keeping life-cycle costs on optimized level. High reliability requirements and increasing railway safety led to definition of railway standards which statements have to be fulfilled in different stages of life cycle of railway components.

The current regulations for the design of railway axles in the European Union, are EN13103 [8], EN13104 [9], relative to trailer and motor railway axles respectively, and EN13261 [5] / EN 13260 [10] regarding product requirements for axles and wheel sets.

Due to the fact that railway axles are mechanical components which are working under effects of rotating bending at high number of cycles (lifetime of railway axle is considered for $10^7$ kilometres) bending fatigue is the critical factor which has to be addressed in both axle material selection process and design guidelines.

Investigation on material fatigue resistance was established by Wohler studies (extensive tests were conducted on railway axles) and was continued by many researchers up to nowadays allowing increase material properties knowledge and safety level of products made of them. As outcome of material properties studies many types of steel has been used for production of railway axles. Nowadays for railway axles production in Europe mostly two types of steel grades are used, namely, the low strength A1N (normalized C40 carbon steel) and the medium strength A4T (quenched 25CrMo4) but research and development of new steel grades successfully introduced in railway applications is still ongoing [6].

Despite ensuring proper material for axle production, in order to provide safety and reliability of axle during its entire life cycle a procedure for proper dimensioning based on calculated stress spectra acting on axle sections needs to be applied. In European norms [8, 9] infinite fatigue life assessment for calculation methodology is considered. For this type of assessment calculations are carried out evaluating the stress state, along all the critical sections of the axle, due to the bending moments given a defined distribution of loads acting at the bearing journals, at the contact points between wheels and rails and on the brake disks; moreover, regarding the powered axles, the effect of traction is
considered. Due to fact that the service loads acting on a wheelset are not well known, [8, 9] adopts a combination of loads onto the axle, chosen as representative of the most critical loading condition. The design is then performed by a simple comparison of the stress state acting on the critical sections of the axle against the Fatigue Limit of the involved material reduced by a generous safety factor in order to take into account all the uncertainties about the working conditions of the wheelset during its life [11].

Another approach for the railway axle design, according to the FKM guidelines [12], is the fatigue assessment adopting damage sum criteria derived from the concepts of damage accumulation originally due to Miner [13]. In particular, two approaches are proposed by the guidelines: the simplest is the Haibach model [14], identical to the original Miner’s rule except that two regions of the fatigue curve are defined, the ‘finite life’ region and the ‘infinite life’ one. The more complex approach, on the other hand, is represented by the consequent Miner’s rule: when the component starts its service under variable amplitude loadings, and in the case the load spectrum contains some cycles above the (initial) endurance limit, these cycles will reduce it for a certain amount, and this process of reduction progressively increase while the damage rises; as a consequence of the phenomenon, continuing the variable amplitude loading, also the cycles of the spectrum initially under the endurance limit begin to contribute to the damage, till the damage sum reaches a critical value and the endurance limit gets to zero.

1.3. Railway axles failures

Railway axles, even though designed for infinite life, and severely checked before their re-admission to operation are subject to failure due to various types of surface damage (i.e. ballast hits, corrosion pits) that may occur during their very long service life, thus occasional incidents have been observed in service. Due to this fact regular axle examinations in the form of non-destructive testing inspections are performed. Period of operation between tests is calculated on the basis of the propagation lifetime of a given initial defect. The key point is therefore a reliable estimation of propagation lifetime under service loads. Reported failures were caused by fatigue fracture which typically occurs at the press-fits for wheels, gears and brakes or at the axle body mid-span and close to transitions. Although this type of failure always occurs due to fatigue crack propagation, its actual initiation, during the long lasting service life of axles (up to 30 years or $10^7$ km), can be due to various factors such as paint detachment, pitting from corrosion, damage from ballast impacts, fretting fatigue in areas subject to interference fits, and others. Some of this crack initiation factors are presented in Figure 1.2. This kind of damage can potentially act as an initiation of a fatigue crack, causing final fracture of the component.
Figure 1.2 Different kind of damaging of an in-service railway axle
upper: paint detachment  middle: corrosion pits  lower: damage from ballast impact
credit: Lucchini RS, Italy

Mechanical damage (like damage from ballast impact) can occur at any time during operation of railway axle and even overhaul process. According to estimations probability of damage during service cycle is assumed to be in a level of 30% for high speed trains and around 5% for other types of trains [15]. As can be seen from Figure 1.3 sudden impact causing 3 [mm] dent in railway axle is sufficient to initiate crack growth which dimension is progressing rapidly during vehicle operation.
Another dangerous and frequent cause of crack initiation is corrosion of railway axle. Axle life assessment due to corrosion is still under investigation and more accurate models are being developed but it is also worth noting that the synergetic action involved in the mechanics of corrosion fatigue make the threshold stress intensity factor of short cracks initiated at corner pits lower and the crack growth faster when compared to the laboratory air conditions [16]. According to [15] crack growth mechanisms initiated by corrosion “change over” at 2-3 [mm] depth for stress levels typically seen in axles and cause rapid growth of crack dimension (Figure 1.4).

The type of abovementioned failures can be managed by adopting the “Damage Tolerance” methodology, originally developed for aeronautic applications in the seventies [17, 18] and more recently introduced in the railway industry [19-21]. Philosophy of this methodology is based on accepting that a crack/flaw could exist and propagate in a component. The main aim of axle/component integrity assessment is to calculate propagation lifetime under service conditions from a given initial defect. A common assumption is to consider an initial crack depth equal to 2 [mm] which corresponds to a conservative assumption for damage induced by ballast hits [22]. Once propagation lifetime has been estimated, the inspection interval is then chosen by calculating axle
failure probability from the “Probability of Detection” (POD) curve (as in Figure 1.5) of the type of NDT technique adopted (an alternative method is to define the NDT specifications for a given inspection interval [8]).

![Example of POD curve for hollow axle inspection (10% threshold)](image)

Figure 1.5 Example of POD curve for hollow axle inspection (10% threshold) [23]

A simple definition of the inspection interval, by which at least one possibility of detecting the crack during the axle’s service is given, is shown in Figure 1.6.

![Simple definition of the inspection interval, from the crack growth simulation](image)

Figure 1.6 Simple definition of the inspection interval, from the crack growth simulation [11]

Other methodologies for the definition of the inspection intervals are based onto the probabilistic approach, for the definition of the probability of failure.

Concerning the current regulations, only operative indications for the freight axles, regarding the distance between inspections, are present in the active guidelines, drafted by VPI Germany, Austria and VAP Swiss [24]. These guidelines recommend, as general indication that an inspection has to be carried out every 600000 [km], controlling the axle by ultrasonic inspections. Relatively to the railway axles for freight wagons, a full inspection of the axle, including magnetic particles, is usually carried out each 1.2 millions km, when the wheels are re-profiled; intermediate inspections are carried out only adopting
visual and ultrasonic inspections. It’s worth to remark, that there is no Standard regarding the inspections of railway axle, but only guidelines about the freight wagons.

Keeping in mind that even when the axles are operated to the operating loads axle failure (Figure 1.7) can happen due to the synergetic effect of both corrosion (or others factors causing crack initiation) and cyclic loads. Although new models are being developed for estimation of reduced life of railway axle due to structure degradation effects [25], there is strong need both from safety point of view and maintenance costs to develop system capable of monitoring axle during its normal service operation, allowing to estimate its condition and warn operator in case any failure is predicted.

![Figure 1.7 Railway axle failure](image)

**Figure 1.7 Railway axle failure [3]**

left: wheel set with broken axle  right: axle fracture surface with three distinct zones

### 1.4. Defect detection methods in condition based monitoring of railway wheelsets

As could be concluded, based on previous sections of this work, proper maintenance as well as monitoring is crucial to ensure that wheelset of railway vehicle is operating in good condition. Possible damage of axles can include cracks initiated from the inaccessible (invisible) areas (like wheel seat, break seat), for which mostly ultrasonic test are being used, as well as cracks appearing on accessible parts of axle (mostly surface cracks, like ones initiated by corrosion) which are detected by means of methods like magnetic particle inspection, ultrasound techniques or eddy currents. Although these abovementioned methods are mostly used during wheelsets maintenance process continuous effort is paid to improve detection methods so they are becoming quicker and more reliable. To avoid unplanned interruptions and thus meet the growing demands on cost efficiency, reliability and safety for railway vehicles, the implementation of intelligent condition monitoring systems is highly desirable. A number of techniques have been utilised to perform fault detection in railway vehicles. They include, advanced filtering, system identification and signal analysis methods. The practical application of condition monitoring of the train dynamics are done either through the employment of track-based sensors or vehicle-based sensors.

Despite the fact that vehicle based sensors are mostly used to monitor track parameters (like rail irregularities, track cross levels, vertical alignment, etc. ) there are some applications which are utilising information gathered by this sensors to estimate
overall vehicle performance generally focusing on wheel rail interaction [26], estimating forces at wheel–rail interface [27] and analysing acceleration signals of vehicle bogie [28].

As far as wheelsets and specifically axle is under consideration, most of tests for fault detection are conducted in-house during maintenance process, causing extended interruption in vehicle operating time. Modern railway is facing the necessity to be available all time, thus disruptions caused by activities such as inspection, removal and reactive maintenance need to be minimised. In this way, it is necessary to conduct more effective inspection and maintenance in less time by optimising and introducing automation where possible.

Some existing methods and automated systems of fault detection in wheelsets were recently reviewed by [29] and are briefly presented here, specifically:

**Wheel profile monitoring systems**
These systems are mostly using a laser line or high-intensity strobe light which illuminates the wheels allowing images of wheel profile to be taken using high-speed digital cameras. Afterwards images are analyzed by specialized software providing information on the wheel parameters like flange height and width, tread hollow and rim thickness, etc. This detection method provides results when train is passing detector at low speed, mostly below 20 km/h.

**Tread condition detectors**
Technology employed in these systems utilises the effect of ultrasonic waves attenuation caused by discontinuity in investigated thread surface of wheel. Ultrasonic waves are generated by means of lasers and after passing through wheel tread are detected by non-contact ultrasonic transducers. Defects as surface breaking and cracks are fruitfully detected due to high sensitivity of signal measuring transducers. This application allows defects to be detected during train passage at approximate 10 km/h.

**Wheel impact load detectors**
These systems are capable of detecting wheel out-of-round, shelling, spalling and flat spots, which are creating additional loads (impacts) when wheel is rotating. The rail deflection caused by the vertical forces exerted by the wheels is measured and analyzed to determine the wheel tread irregularities. Very often additional optical sensors, strain gauges and load cells are used to increase precision of measurements and provide comprehensive wheel condition data.

**Hot axle bearing detectors**
In these systems mostly infrared cameras are being used to measure temperature of axle bearings while train is passing detection station. Any temperature above predefined threshold is detected and registered, thus allowing to identify which bearing is undergoing defect situation. This system is very robust and allows detection of hot bearings when train speed reaches up to 500 km/h. This technology allows to detect also defects which are causing thermal effects at wheel discs and breaks, as well as so called cold and hot wheels (breaks failed to apply or failed to release breaking respectively)
**Acoustic bearing defect detectors**

For defect detection in bearing array of microphones located close to the track is used. If noise generated by passing bearing is recorded and its level is beyond defined threshold corresponding bearing is investigated for possible faults. This system is capable of detecting defects before they cause complete failure of bearing.

**Brake pad inspection systems**

Inspection method in these systems is based on image processing techniques, which are analyzing digital picture of brake pad taken when train is passing diagnostic station, thus providing information about brake pad wear, uneven wear and in worst case missing pad.

Although many automated systems exists providing detection of possible faults in wheelsets, for axle fault detection specifically, still mostly stationary methods are used, mostly requiring to remove whole axle or some of its auxiliary components from vehicle prior conducting test. Most popular NDT techniques are eddy current testing, magnetic particles and ultrasonic tests. In general surface NDT methods (i.e. magnetic particle inspection, eddy currents) can be used when the axle body is inspected except areas where access is restricted for example where there is no visual and/or hand access. For cracks under the wheel seat (for example) surface inspection methods can be used only when the wheels are removed. Internal defects as well as those located at inaccessible areas are mostly investigated by means of ultrasonic tests. Nevertheless of stationary nature of abovementioned tests improvements are introduced to shorten the testing time and thus reduce interruption in vehicle service, as well as new methods are introduced to allow inspection on moving train. Few chosen methods already in use in stationary testing as well as most promising methods for axle tests on moving train are briefly introduced here.

**Phased array detection**

Phased array detection method is an evolution of the standard ultrasonic inspection. The difference is in a type of probe used to generate ultrasonic waves. Phased array probe is capable of generating a vast number of different ultrasonic beam profiles from a single probe assembly, thus covering a greater area of specimen under investigation. Moreover, thanks to electronic control system, ultrasonic beam steering and focusing can be performed decreasing testing time and increasing detection precision. Typical phased array probe compared with conventional one is shown in Figure 1.8.

![Figure 1.8 Schematic representation of Phased Array Probe](image-url)
Phased array probe system for both types of axles, solid and hollow has been developed. In case of solid axle inspection, most transverse defects located along the axle can be detected by opening and inspecting from the end side of axle. After removing end cover of axle phased array probe is attached and automatic beam conditioning (steering and focusing) is utilised along with automatic probe rotation allowing to perform complete axle inspection within less than 5 minutes [30]. To achieve this short inspection time measuring system is mostly equipped with additional devices, namely automatic electromagnetic fixing and scanning mechanism and electronic control device. Phased array working principle is schematically presented in Figure 1.9.

![Phased array working principle](image)

**Figure 1.9 Phased array working principle**
(a) phased array beam scanning (b) corner beam covering (c) axle ultrasonic transmission

Phased array bore inspection method for hollow axles is utilising its biggest advantage of electronically rotated ultrasonic field, replacing rotating probe methods.

Mechanical part of system is necessary only to provide longitudinal translation of probe along axle bore. In this approach complete hollow axle is scanned with a fixed beam angle in the axial direction and an electronic rotation of the ultra sound field by sweeping the active element groups around the circumference.

![Principle of the conical phased array probe](image)

**Figure 1.10 Principle of the conical phased array probe**

Inspection system for hollow axle consist mostly a phased array device with a probe, a motor driven linear axis for moving the probe axially inside the bore and a PC for device control. This configuration allows reduction of inspection time four times in comparison with conventional ultrasonic method (20 minutes versus 5 minutes respectively) [31].
Microwave sensor testing

This method is based on a physical phenomenon - the interaction of electron gas and ultrasonic waves in metals. Emission of ultrasonic waves on the metal surface is determined by microwave sensor. Theoretical works showed that there is possible fact to determine remotely dangerous-or-non-dangerous inner metal stress, measuring the surface conduction (density of surface charges)[32]. Research and testing were conducted, including remote indicators of active defects on steel samples and railway wheels in static mode, as well as, dynamic testing in moving mounted wheels. Experiments showed that this new method could be used to determine the initial formation of active defects (cracks), even under elastic tension.

One of the most important advantages of microwave method is the fact that it can be used for detecting surface cracks under dielectric coatings. Because microwave signals easily penetrate inside dielectric materials, this methodology is expected to detect cracks under dielectric coatings of various thicknesses. It must be noted that dielectric coatings such as paint, corrosion preventative substances, etc., may have varied thicknesses although they are generally not very thick and are commonly known as the family of low-loss dielectric materials.

AC thermography

AC thermography detection method is based on high frequency electric current passing through axle on inspection station. Current is deployed through the wheels using a direct application. Due to current flow a slight heating effect is generated on the surface of inspected specimen. Heat generation is increased at crack corners, creating hot spots at crack ends, while lower temperature can be observed at crack centre [33]. The disadvantage of this method is the fact that no equipment can be located in the way of the camera recording temperature changes. Field tests showed that AC thermography allows to shorten inspection time, but due to possible interference in temperature reading caused by additional sources (sunlight interference, unexpected sources of heat reflected) and the fact that only exposed areas of axle can be inspected this method needs engineering improvements prior to its introduction in larger scale application [31, 34].

AC thermography working principal as well as crack detection result on tested specimen is presented in Figure 1.11.

Figure 1.11 AC thermography method [33]
left: installation scheme middle: inspected specimen, right: detection result
Acoustic Emission (AE)

Acoustic Emission Method (AE) is based on the observation that damage developing in a material releases energy in form of ultrasonic elastic waves (Figure 1.12). These waves are typically short and transient with bandwidth in the 100-1000 kHz range, meaning that they can be successfully detected [35] and making this detection method quite robust against audible noise and structural vibrations. Traditionally, the so-called “parametric AE” is used, in which the transient waveforms are stored for the calculation of some well-established parameters (like peak amplitude, duration, energy). These parameters are then stored and used for further processing.

More advanced techniques allow the analysis of the full waveforms (continuous AE).

![Figure 1.12 AE theory principle](image)

AE is traditionally used and standardized as a non-destructive technique to assess the structural integrity of metallic components (e.g. pressure vessels and pipelines) under static and fatigue loading [36], but with only few applications in the railway field [37].

In railway field AE tests are being introduced mostly for investigating the state of a bogie bolster [38], monitoring the state of rail defects [39] and for acceptance tests of bearing races [40] but research effort is put on possible application in railway axles monitoring. Tests conducted on railway axle showed that sensors are capable of sensing fracture of the axle, given that appropriate signal processing is used (achieved with wavelet denoising techniques) [37, 41].

Laboratory tests of axles equipped with AE sensors proved that crack detection is possible (in acceptance / rejection tests) [42] but for crack location more tests needs to be performed and more advanced de-noising techniques applied by means of signal processing. This method is still under development but showing promising laboratory results [43].

![Figure 1.13 AE application in railway axle crack detection tests](image)
Laser-based ultrasonic cracked axle detection

This type of technology for crack axle detection utilises a laser in conjunction with standard ultrasonic transducer to detect defects on the axle of train passing through a testing station. A high-energy, pulsed laser is used to generate ultrasonic modes in a axle and a non-contact, air-coupled transducer to receive the ultrasonic signal emitted by the specimen which is then sent to a signal processing unit for analysis to determine the presence of cracks across the axle circumference [44]. Each inspection is capable of detecting circumferentially oriented cracks across the body of the axle. By repeating the inspection multiple times around the circumference of the axle, it is possible to detect cracks around the entire axle body. The automated cracked axle detection system consists of 10 inspection stations, which inspect the axle for cracks greater than 7 [mm] long [29]. The system works at speeds up to 32 [km/h].

High Frequency Vibrations

System based on High Frequency Vibrations analysis detects cracked axles through the use of compensated resonance, by exciting the axle with an impact and measuring the resulting vibration with an accelerometer in contact with the axle. Changes in the high-frequency components of the vibration indicate the presence of a crack.

This system is currently being tested and developed as a part of WIDEM, a partly EU funded wheelset improvement research consortium. Initial testing indicates that the system is capable of identifying cracked axles, but the product has not been commercially released at this time.

Due to the fact that most of presented methods either require gaining access to the axle by removing some components, or building wayside inspection stations, through which trains have to pass during their operation there is still gap to be filled for online system gathering real-time data form railway axles thus allowing for continuous monitoring of this component. To achieve this goal present work is proposing an approach which is new to the railway field by introducing low frequency vibration analysis of the axle. Theoretical background and examples of application will be given in the following chapter.
Crack-related vibrations in rotating shafts

Due to the fact that railway axle is a mechanical component operating under variable loads and angular speeds, its behaviour falls into the wider field of dynamics of rotating systems called rotodynamics. Condition monitoring of rotating systems has been an active field of research during the past four decades. During this time rich database of theoretical as well as experimental research has been created, as a result of demand for reliable procedures that allows to detect and characterize fault condition of an operating system without interrupting its normal working regime. Since generally any mechanical component during operation passes through some degradation states before failure, if such a condition can be detected in advance then proactive and corrective maintenance can be applied before a complete failure occurs. This approach therefore, offers cost savings compared to classical preventive activities which are performed periodically without knowing if the component is really defective [45].

Among different strategies of estimating health condition of rotating elements, vibration analysis is one which has many advantages. Vibrational signals can often be measured by relatively inexpensive sensors (displacement proximity probes, accelerometers, etc.) and what is most important, vibration signal of working system gives immediate evidence of an eventual fault developing in its structure.

For characterising and analysing vibrational signal obtained from rotating components different techniques of signal processing are used. Vibration analysis has been used for a long time and already sophisticated techniques were developed over the last twenty years, i.e. [46]

In general, the vibration signal contains the information of the oscillatory motion of the machine and its auxiliary components. Each of the sub-components of rotating machinery, like gears, bearings, shafts, couplings, engines, electric generators, pumps, fans, etc., can be modelled so that the vibration from various modes of failure can be predicted [47]. The goal of vibration signal processing is to analyse the vibration signal measured at particular locations on the machine, and extract enough information to determine the condition of each of the sub-elements of the machine.

Due to the fact that developing faults are giving response in specific range of angular frequency of rotating component, very useful from diagnostic point of view is to observe measured signal in the frequency domain. This can be achieved by applying an algorithm well-defined in literature called Fourier Transform. The reference situation, where vibration signal was recorded under normal (no fault) working condition of system is necessary in any case, enabling comparison and trending of eventual changes in vibration signal components. Trending of the Fourier transform frequency vectors itself, was
acknowledged to be useful for monitoring the condition of rotor shafts for the presence of cracks [48].

Having proper, reliable, information coming from signal analysis, combined with diagnostic experience or guidelines for system under consideration gives the possibility for decision maker to correctly judge whether the system is acceptable for service or maintenance should be planned.

2.1. Cracks in rotating shafts

Crack occurrence and its development are one of the most common causes of failure in mechanical structures. Specifically, systems working under cyclic loading basis are prone to this type of phenomena, which if not monitored can lead to complete failure of elements in which crack was advancing.

As research done by many authors shows (some examples specifically for railway axles were given in Chapter 1) crack may be initiated in areas with high stress concentration (i.e. sharp changes of diameter of part geometry), surface imperfections (including corrosion) and also due to material inclusions from which parts are being made. Frequent source of crack initiation is fretting corrosion in case of shrink/press fitted connections specifically when operating in wet and corrosive environments (i.e. railway wheels).

Crack existence and its dimensions are investigated commonly by means of non-destructive tests, i.e. dye penetrant, ultrasonic, MPI (brief description given in Section 1.4). In case of shafts working in field of turbomachinery (i.e. power plants) dynamic tests, based on vibration measurements are suggested for identifying presence and location of cracks as the change in the overall dynamic behaviour might be caused by the local reduction of the stiffness due to a crack that has developed somewhere in the structure [49].

In rotating components like shafts, the most frequent type of crack appearance is so called transverse crack, although its possible to find also slant, helicoidal, and multiple cracks. Transverse cracks, are the type in which the crack surface is orthogonal to the rotation axis of the shaft.

In general, crack existence is much harder to detect in case of rotating element during its operation, especially when compared with stationary structures, thus vibrations analysis have been found to be a suitable method to extract symptoms justifying developing fault in a component under investigation. If these symptoms are found and they are giving the right to suspect fault existence which is putting threat to system safe operation, then maintenance activities might be planned with minimised disturbance in machine working schedule.

2.1.1. Crack propagation in rotating shafts

Research on crack initiation and propagation phenomena generally falls into the field of fracture-mechanics. More accurate models are continuously developed to investigate crack propagation paths and speeds [25, 50] which are giving possibility to better design or predict components life prone to crack occurrences.
Since the most successful approach to the study of fatigue crack propagation is based on fracture-mechanics concepts thus in case of crack investigation in rotating elements fracture mechanics along with rotordynamics play key role in modelling, analyzing, and interpreting results of analysis.

During extensive studies of cracking phenomena done by many authors it has been shown that high stress intensity factors are developing at the crack tip and thus allow the crack to propagate deeper inside element structure, even if the external loads are not changing. The crack propagation path is generally determined by the direction of maximum stress or by the minimum material strength. The crack can propagate slowly at the beginning, often with periods of rest alternating with periods of propagation, but as soon as the crack will reach a relevant depth, then the propagation speed increases dramatically and the shaft may fail in short time of operation. At the initial stage of crack propagation the propagation direction is depending on local conditions thus it can follow virtually any direction. The situation changes as soon as the crack reaches the deeper regions of the structure when the direction of propagation is happening mainly in radial direction with small vanishing curvatures. When a cracked element is loaded with constant amplitude its very often visible that crack propagates in steps, creating specific cracked surface pattern (Figure 2.1) where rest lines are clearly visible [49].

Example of crack propagation and its influence on the vibration levels of a rotating shaft (from power plant turbomachinery) is presented in Figure 2.2. In this case 1x and 2x harmonic components of vibration signal were analyzed, and analysis results showed dramatically increasing amplitudes. After machine shut-down a transverse crack was found that had the circumferential extension of about 25% [51]. As can be seen in Figure 2.2 the crack, after reaching its critical size, has propagated in just 4 days up to a level when further machine operation was dangerous.

Figure 2.1 Cracked axle cross section with visible crack rest lines
2.1.2. Crack breathing mechanism

As previous section of this work shows, propagating crack in rotating element is causing change in its vibration signal. This change is coming from specific phenomenon called the breathing mechanism of crack.

The breathing mechanism in general is a phenomenon of opening and closing crack lips (surfaces created by propagating crack tip) caused by the application of bending moment and rotation to a cracked element. This behavior comes from the fact that due to the presence of gravity and/or load, the upper portion of the cracked rotor at the beginning of a rotation cycle is under compression which is closing the gap in structure caused by crack and prevents it to open. This situation holds until the rotor starts to rotate. If it is assumed that the gravity/load direction is constant, the upper part of the shaft consisting crack along with rotation gradually moves towards the bottom, where compression is changing into tension (due to shaft bow) thus causing the crack to open gradually until the gap in between crack walls is fully opened. The process repeats and a periodic crack opening and closing phenomenon occur, thus giving rise to the actual breathing movement. As have been proven by researchers the breathing mechanism of the crack comes from the fact that the static deflection of the shaft is much greater than the deflection due to the dynamic response of the cracked rotor [52]

The change in cross sectional area of shaft at crack plane which is resisting bending is causing variation of shaft stiffness during its full rotation, which is a direct result of the crack breathing mechanism. These stiffness variations are then causing additional deflections of the shaft even if load amplitude remains constant.

Recently, tests investigating in details crack breathing mechanism were conducted by [49]. In these tests, small scale horizontal cracked specimen have been loaded with different stationary loads and has been rotated in different angular positions in order to excite the breathing of the crack. During the test two different effects have been observed, firstly, that the crack closure effect generates an internal bending moment that holds the crack closed, which stays closed until the external bending moment overcomes the
internal bending moment, and secondly that when the crack is closed the measured compressive strain is much higher than the theoretical strain calculated assuming a linear compressive stress distribution over the cracked section. This was explained by assuming that the contact is not spread over all the cracked area when the crack is closed, but it occurs only on a smaller portion of the cracked surface, or on the crack lips only, determining higher strains associated also to stress intensity factors. This aspect is also related to the crack closure effect.

As can be understood the breathing mechanism is the result of the stress and strain distribution around the cracked area, due to static loads (weight, reaction forces, etc.) and dynamical loads (i.e. unbalance). As experiment results are showing [49] as long as static loads are overcoming the dynamical ones, the breathing is governed by the angular position of the shaft with respect to the stationary load direction. In this configuration crack opens and closes completely once per revolution of shaft.

Important conclusion after observing the crack breathing mechanism and its influence on structure was that despite the highly non-linear stress and strain distribution in the cracked area and the non-linear breathing behaviour, the overall load-strain behaviour of rotating shaft resulted to be quite linear.

Although some authors [53] were assuming an abrupt change of crack condition from completely open state to completely close state, detailed investigation of crack breathing mechanism done by further research proved that this situation is unusual in real applications. Effect of gradually closing and opening crack lips is more closely representing crack breathing phenomena of real shafts. This fact was confirmed also by the use of finite element models to investigate crack influence on train axle, which is the topic of present work. Rather fluent decrease in contact area between two crack surfaces can be seen when analyzing contact characteristics from simulation case. Decrease and following increase of contact area versus axle rotation angle can be observed as shown in Figure 2.3.

![Figure 2.3 Area in contact between crack surfaces as a function of axle rotation](image-url)

(35% cracked shaft cross section)
2.1.3. Dynamic behavior of cracked shafts

Proper assessment of whether or not a rotating shaft is operating in healthy condition requires knowledge of its dynamic behavior. To describe and understand dynamics of a cracked shaft both rotordynamics as well as fracture mechanics concepts are playing key role. Due to the fact that both of these fields of science are wide, but also well defined in literature, this section will focus on the review of the influence of crack on the vibrational signal, thus giving insight to understanding results of vibrational signal analysis.

A crack in the rotor causes local change in stiffness, which than affects the dynamics of the system: frequency of the natural vibrations and the amplitudes of forced vibrations are changed. The influence of a transverse crack on the vibration of a rotating shaft has been studied by researchers since three decades. Wide review of crack modeling, understanding the behavior of cracked shafts as well as crack localization techniques was widely contributed mainly by [48, 54-57]. More recent studies have been published by [58].

As was already stated, apart from the overall level of vibration, vibration signal harmonics are offering great diagnostic information. Due to the fact that every sub-element of a rotating system generates a vibration specific pattern (i.e. gears, bearings, etc.) in specific range of frequency, analysis in frequency domain is very useful from diagnose point of view. In cracked rotating shafts, crack presence has peculiar influence on dynamic behavior of the shaft due to the breathing mechanism occurring during each revolution of shaft. By analyzing the harmonic components of vibration signal recorded from a cracked shaft some observations can be made. According to [49] when deflections are measured along the shaft loaded by its weight during a complete revolution of the shaft, the Fourier expansion allows the definition of 1X, 2X and 3X deflection shapes. All are due to the local flexibility of the crack only and have therefore a bi-linear shape with a pronounced curvature in correspondence of the crack. Therefore some typical symptoms of crack influence on shaft dynamic behaviour can be observed, namely a change in 1X, 2X and 3X vibration signal harmonic [59-62].

A brief summary of common changes and its causes can be made based on literature [49, 50] specifically:

- the 1X component is increasing mostly due to the development of the bow caused by the crack, however also can be caused by many other faults (e.g. unbalance, coupling misalignments, etc.), so attention needs to be paid when taking diagnostic decision;

- the 2X component is mostly changing amplitude because of axial asymmetry in rotating shaft due to fault, but also can be caused by surface geometry errors (journal ovalisation) and intrinsic stiffness asymmetry in case of asymmetric shaft that could cause 2X vibration component also during normal operating, which has to be treated as machine native vibrations;

- the 3X harmonic component is mostly caused by a breathing crack mechanism, but this excitation is rather small and can be recognized only when resonance amplifies the response. It's necessary to note that 3X components can also be caused by surface geometry errors (journal ovalisation). In some cases it was
observed that 3X component didn't change its amplitude even if deep crack were present in shaft, which was coming from the fact that crack was permanently opened, thus no breathing mechanism occurred [63].

Examples are mostly showing that the presence of a crack caused a remarkable change in the normal vibration level of the machines [51, 64, 65].

It is worth emphasizing, that not only an increase of the harmonic components can indicate the presence of a crack. Examples of decreasing spectra amplitudes due to crack propagation can be found in literature (Figure 2.4). Interesting case of deeply cracked generator rotor (more than 50%) [66] was reviewed by [49] who concluded as follows:

- the reduction of the synchronous (1X) vibration amplitude is not at all a symptom of a healthy machine, but indicates in this case that something dangerous is happening. In this particular case the 1X vibration component generated by the propagating crack was in phase opposition with respect to the vibration component due to unbalance and bow, producing a reduction in the total 1X component;
- the 2X component is increasing continuously;
- the phase of both 1X and 2X vibration component is changing;
- the crack can propagate very rapidly in its final stage.

On the other hand there are plenty of examples (i.e. [67]) where both 1X and 2X components are undergoing increase in amplitude due to crack propagation (see example in Figure 2.2)

In fact it is obvious that decision about planned maintenance of shaft due to crack presence suspicion cannot be made only based on the observation of increased
amplitudes of specific harmonics (mostly 1X to 3X) but rather based on the rate of change of their amplitudes.

Generally, if comparison with stationary structures could be made, rotating shafts have some special characteristics. First one is coming directly from the crack breathing phenomena, which is independent from the vibration. In rotating components crack is opening and closing gradually, while in stationary structures this change is rather abrupt, due to character of applied external forces. Second important characteristic is the fact that vibrations originated from crack in rotating components are excited naturally by the rotation of shaft (breathing phenomena), thus periodicity is much more expected, while in stationary structures these vibrations are excited by external forces.

As [68] concluded, after extensive literature review and own experiments, in case of rotating shafts 2X component trend analysis is the best tool for discovering a crack. It comes in line with other works, i.e. [60], where nonlinear dynamic behaviour of the cracked rotor was investigated experimentally and authors also observed higher harmonics (especially 2X frequency component) in the response of the cracked rotor, when the rotor is running in the subcritical speed range (below the angular frequency that causes a resonance).

Due to the fact that obviously vibration measurements are giving sufficient information to estimate crack occurrence, and that specific crack-related vibrations characteristics depend directly on crack breathing phenomena (thus crack size and location) the importance of adequate modeling of crack zone is essential in case of numerical simulations of cracked shaft dynamical behavior.

### 2.2. Crack modelling

Accurate modelling of the cracked shaft allows to simulate its dynamic behavior, including influence of crack different location, size and shape. The comparison of calculated vibrations and measured vibrations allows the identification of not only the presence of a crack in a rotating shaft, but also of its location and depth, using a diagnostic procedure [59], thus having available a correct model of crack zone is essential for a research process. Some popular crack models will be briefly introduced in following subsection.

#### 2.2.1. Types of crack models used in simulations

Nonlinear dynamics of the cracked rotor is mostly investigated using two well-known crack models, i.e. switching crack model and response-dependent breathing crack model.

Switching crack model was developed following the fact that the transition from closed crack (full) stiffness to the open crack (weak) stiffness has been generally considered abrupt [53]. This model was easier to implement mathematically but further investigations revealed its disadvantages. In real cracked rotors, the stiffness variation is likely to be gradual, the assumption of abrupt stiffness switching is not appropriate. Hence, the switching crack model might adequately represent only very shallow cracks. [52] proved that switching crack modelling reveals chaotic, quasi-periodic and subharmonic vibration
response for deeper cracks. Chaos and quasi-period nature of the response is due to the
unstable stiffness switching due to crack breathing in the rotor system. Rotor system
enters into chaos or quasi-periodic motion directly from periodic motion, and leaves the
chaos through route of quasi-period. Transient chaos behaviour was also observed for the
first time in the case of cracked rotor. Unbalance eccentricity level and its phase, crack
depth and damping were found to have considerable influence on the bifurcation
phenomena of the cracked rotor modelled using switching crack model. The most
significant finding of [52] is that the chaotic and quasi-periodic vibrations reported in the
past for cracked rotor could be due to the mathematical approximation model of the crack
breathing. It was also emphasized that the kind of chaotic, quasi-periodic and subharmonic
response found using switching crack model, is not observed experimentally by
researchers [55, 56, 69] thus better models have to be used for dynamic behaviour
analysis of cracked rotors.

More realistic, breathing crack model was presented by researchers, which accounts
for the forces acting on the crack cross-section. It evaluates the open-close area of crack
and in turn estimate the shaft stiffness, allowing closer representation of the crack
breathing in real rotors. This model revealed no evidence of chaotic, quasi-periodic and
subynchronous vibrations in the response, which was also confirmed by paper [52]
stating that unbalance phase, unbalance level, depth of crack and damping in the system
have significant influence on the nonlinear vibration features of the cracked rotor with
switching crack but have no influence on nonlinear vibration features of the rotor with
breathing crack.

Different mathematical models of breathing crack are reported in literature. Breathing
mechanism described by sinusoidal stiffness variation was introduced by [70], which was
more practical for deep cracks than a hinged model (switch model) in case when crack
opens and closes gradually due to gravity. Crack breathing steered by a square pulse
function was presented by [52, 61] for small cracks, whereas crack behavior described by
mains of harmonic variation [71] was proposed for deeper cracks. For heavy shafts cosine
steering function for crack breathing mechanism was used by [62] which accounted for
partially opening and closing of crack area, thus representing reality more closely. In that
model though, no direct relationship between the shaft stiffness and the material properties
of the shaft was existing. Further development of models resulted in proposing stress
intensity factor approach [61, 69, 72, 73] based on linear elastic fracture mechanics
(LEFM) assumptions. However also this method has its limitations coming directly from
LEFM theory (initial crack has to be assumed in part, fracture area has to be small
compared with shaft, errors coming from the fact that crack passes from stress state
cased by vertical moment to one caused by lateral moment, etc.), and also there is a
need of geometrical simplifications (mostly assuming rectilinear crack front). Moreover
when the crack reaches depth passing the radius of shaft than elements of compliance
matrix present a divergence, which is not reflecting reality [71]. Interesting method was
proposed by [74] which was based on harmonic balance, validated using simulation and
experimental results.

Despite the large number of mathematical model existing in literature, authors agree
[54, 56-58, 75] that the breathing mechanism of cracks in rotating shafts is accurately
investigated by means of 3D non-linear finite element models.

Finite element model representing rotating shaft was introduced in [76] where by using
fracture mechanics approach full 6x6 stiffness matrix of opened crack surface was
obtained. In [77] the rigid finite element model to investigate a crack in a rotating shaft was
used, where spring-damping elements with variable stiffness were representing crack breathing zone and were applied in between two sections of the shaft. Recently more advanced model has been presented by [78]. In this approach the cohesive zone model (CZM) is used to represent crack area, which describes material failure on a phenomenological basis (i.e. without considering the material microstructure). The general advantage of this model when compared to LEFM is that the parameters of the respective models depend only on the material and not on the geometry. This concept guarantees transferability from specimen to structure over a wide range of geometries. The CZM, a model can deal with the nonlinear zone ahead of the crack tip due to plasticity or micro-cracking. The constitutive behaviour which causes the cohesive elements to open and eventually to fail is described by the so-called traction–separation law. This model could predict also crack propagation due to possibility of separating neighbouring finite elements when stress caused by bending moment was overcoming values specified in traction separation law. Despite many advantages raising from the use of the finite element approach, there is one major drawback. 3D FEM models mostly requires high computational effort, thus simulation cost is pretty high. Because of this reason precise simplified models are still required. Recently, simplified but precise model of breathing crack in rotating shaft was presented by [68]. In this approach, called FLEX method, the shaft is described according to the Timoshenko Beam theory, assuming that beam element defining crack area have different parameters than neighboring elements. Crack area beam element has predefined periodically varying reduced stiffness to take into account bending stiffness reduction coming from crack breathing mechanism. The length of this special element is chosen based on crack depth function, provided by Author. As a result, equivalent beam element representing crack is changing its parameters while changing its angular position. This approach proved to give simulation results in a very good agreement with measurements (Figure 2.6). Reported results of comparison between simulation results using FLEX method (simplified), 3D FEM method and SERR method are presented in Figure 2.5, where vertical displacements for a crack with rectilinear tip and relative depth of 50% of the diameter are plotted (displacements of corresponding non-cracked beam were subtracted). The considered loads causing deflection were in that case not only bending moments, but bending plus torsion loads or bending plus shear loads, as it occurs in machines where torsion and shear loads are generally combined to bending.

![Figure 2.5 Vertical displacement as function of cracked beam angular position [68](bending and torsion, 50% crack depth)](imageURL)
General insight of cracked shaft behavior can be gained by looking at the vertical and lateral deflection as a function of the shaft angular position. In Figure 2.6 vertical displacement of the shaft loaded by two 20000 [N] loads (in order to have in correspondence of the crack only bending moment and no shear force) with elliptical shape crack is presented. It can be clearly seen that stiffness reduction due to crack opening situation is causing additional deflection of shaft. This additional deflection is disappearing as soon as the position of the crack is moving to the upper parts of shaft symmetry plane, thus causing crack surfaces to close and in result allowing the shaft to re-gain its initial stiffness. Change in deflection is smooth which is in agreement with the theory of gradually closing and opening crack lips.

![Figure 2.6 Vertical displacement of cracked shaft, comparison between measured and calculated deflections [68]](image)

In Figure 2.7 the lateral displacement of a 30% cracked shaft is presented. In this case the specimen was loaded with constant bending moment (M=600 [Nm]) and different shear forces (T = 4285:12855 [N]). Typical lateral displacement curves can be observed as a function of rotation angle, as well as very minor shear force influence can be noticed (this is also true for axial and vertical displacements, although not reported here).

![Figure 2.7 Lateral deflection of 30% cracked shaft with different shear loads applied [68]](image)

C1: M=600[Nm] / T1: M=600[Nm] T= 4258[N] / T2: M=600[Nm] T= 8570[N] / T3: M=600[Nm], T=12855[N]
Proven accuracy of FLEX model allowed to conduct the simulation of a simple symmetrical shaft affected by a crack with rectilinear tip located at mid-span between the two bearings [68]. In this way, analysis of the dependence of vibration excitation on crack depth was done. The shaft was loaded by its own weight only. The vibration amplitudes for different crack depths were calculated at mid-span of the shaft at very low rotating speed, in order to avoid any dynamic effect. After obtaining vibrational signal the three first harmonic were obtained and plotted as a function of crack depth. To obtain dimensionless quantities these harmonics were divided by the mean static deflection, thus providing dimensionless ratios that are independent of the used shaft model and give an idea on how the vibration components increase as the crack propagates deeper (Figure 2.8).

![Vertical vibration components dependence on crack depth](image)

**Figure 2.8 Vertical vibration components dependence on crack depth [68]**

### 2.2.2. Influence of the crack shape on simulation results

Due to the fact that different models reported in literature assume different crack tip shape (mostly rectilinear, elliptical) it is worth to report here the comparison made by [68] concerning different crack tip shapes and their influence on vibrational answer of cracked rotating shaft. The comparison was made for equal cracked area, i.e. the elliptical crack was slightly deeper than the rectilinear crack. The breathing behaviour was calculated for both cracks and finally the vibrations were calculated during a run-down transient of the machine, considering the two cracks in the same position. Comparison results are presented on Figure 2.9 and Figure 2.10 for rectilinear and elliptical crack tip respectively.
It can be seen that the 1X component is somewhat greater for the rectilinear crack, the 3X component is slightly greater for the elliptical crack, but the 2X component is equal for both shapes. As [68] highlighted these simulations suggest that the effects caused by elliptical cracks on shaft vibrations can be considered equivalent to those caused by rectilinear cracks, provided that the same cracked area is assumed. The most significant symptom, the 2X components, is equal for both crack shapes and the differences in the other harmonic components are small.

Nevertheless, based on the use of finite element simulation and some reported experimental results and in order to model more realistically the breathing crack mechanism during rotation of rotor, the breathing crack shape is suggested to be modelled by a parabolic shape that opens and closes due to bending stresses. This approach was also used by other researchers [78-80].
CHAPTER 3

Laboratory tests

3.1. Laboratory test of cracked axles

The experimental tests have been carried out by means of the “Dynamic Test Bench for Railway Axles” (BDA) available at the labs of Politecnico di Milano – Department of Mechanical Engineering (Figure 3.1). The BDA is fully compliant with the relevant standards about the qualification of axles (EN13261 [5], EN13103 [8] and EN13104 [9]). Configuration of test bench allowed conducting tests on real, full size, railway axles.

Totally during test campaign 14 different cases (axles) have been tested. Mentioned tests were conducted in a framework of a project investigating crack growth rates and NDT inspection intervals, but for means of LFV analysis additional measuring equipment was chosen and installed, as well as signal acquisition, conditioning and analysing procedures prepared. Vibration signal was measured during each test, but because of different loading conditions applied not all specimens have encountered crack initiation and propagation phenomena. Due to this fact only chosen results will be reported in following subsections.

3.2. Experiment set-up

A three point rotating bending was applied to the full-scale specimens through an actuator group and an electric motor (Figure 3.1). In this way, both constant amplitude and block loading fatigue or crack propagation tests could be carried out as explained in [81]. As a test specimens both hollow as well as solid real size railway axles were used. In test campaign both crack propagation tests as well as fatigue test were conducted. During crack propagation test axle with introduced notches as crack initiation factors was investigated (Figure 3.2), whereas for fatigue tests axles without any artificially introduced imperfections were used. In this way various cases of crack initiation as well as crack propagation scenarios were investigated.
According to theoretical considerations occurrence of the crack (with sufficient size) due to rotating bending should generate increased amplitude of low frequency vibrations. To achieve accurate measurements of this phenomenon and neglect the influence of vibration signal transmission path direct vibration measurement method was applied. For this reason laser transducers were introduced as vibration signal measuring sensors. Because of difficulties to monitor axial vibration in real application (during operation railway axle is covered at both ends by axleboxes) only radial direction vibration was measured. Two laser transducers were mounted in vertical direction and one in horizontal direction pointing to the central region of the axle (Figure 3.3). In this configuration the highest displacements produced by the applied loads could be measured and highest sensitivity for vibration changes was achieved. To avoid possibility of taking into account vibrations superimposed by different sources both test bench frame (where lasers have been mounted) and actuator providing load have been equipped with accelerometers. Details on used measuring equipment are given in further subsections of this work.

For rotating speed measurement one per rev signal (further called the tachometer pulse) was obtained from the electric motor used to rotate the axle.

Measuring equipment configuration and data acquisition path are presented in Figure 3.3.
Signal acquisition was done using National Instruments DAQ board and LabView routine designed and created for automatic signal conditioning and acquisition triggering based on tachometer pulse at user specified time intervals. Signal analysis was done using routine written in MATLAB environment.

### 3.3. Experiment procedure

During the test campaign multiple axles were investigated. The preliminary analysis of the feasibility of LFV for SHM of axles was carried out considering the special case of cold-rolled solid axles made of A4T steel grade (according to EN13261[5]). The drawing of the full-scale specimen used in abovementioned test is shown in Figure 3.2. In the tested axle, two starting notches, located at 180° one from the other, useful for initiating two independent fatigue cracks in the section of interest of the axle, were introduced by electro-discharge machining (EDM) in the cold-rolled cylindrical part of the axle and had a semi-circular shape with radius R = 4 mm. The full-scale specimen was than subjected to a block loading sequence experimentally derived from the typical service spectra of a tilting train on European lines. As results showed, increasing spectra amplitudes could be observed during propagation of crack [42], which proved initial feasibility of LFV method and allowed for further investigation.

Further tests on different axles were done according to fatigue life assessment procedure, where axles under investigation were not subjected to any crack initiating factors like notches or pre-cracks. Tests were conducted under either constant load and speed or variable load/speed parameters. Due to lack of initial cracks/notches not in all of the axles faults have developed. During these tests an axle was stopped only in case of crack development up to critical size causing great reduction of axle bending stiffness and thus causing excessive deflection under constant load, which was used as alarm trigger in the test bench. Axles after fatigue life assessment were always checked by means of magnetic particle method to investigate crack occurrence and its superficial size.
For proper assessment of eventual crack development initial level of vibration of rotating axle without any defects were always recorded and stored as reference situation. All of measured vibrations signals were conditioned by means of aliasing filters, low pass filters, and synchronous averaged after which Fast Fourier Transform was applied allowing to investigate signal parameters in frequency domain. Spectra orders of vibration signals were plotted and trends of eventual changes in magnitudes saved. Results of this experiment procedure are collected and analysed in following sections of this work.

3.4. Data acquisition

Low Frequency Vibration measuring equipment included three laser transducers to measure absolute axle vibrations in vertical (two) and lateral (one) direction, two accelerometers to measure vibration of test bench frame and hydraulic actuator applying load to the axle causing its deflection. Complete signal bundle was collected in Data Acquisition modules (NI_9234, NI_9239) gathered in DAQ-9172 chassis. In next step LabView software package was used to develop automatic acquisition routine taking into account acquisition triggering (using tachometer pulse) allowing to collect vibration signal for user specified time duration. At this step signal was also filtered (low pass filter) and afterwards saved in convenient file format for further analysis. Because of the high flexibility and capability of MATLAB software further signal analysis was done in this environment.

3.4.1. Sensors in laboratory test

During conducted measurements of low frequency vibrations of rotating railway axle following equipment was used:

a) Laser distance sensor for absolute axle vibrations

Axle vibrations were measured by laser distance meter Mel Electronic M7L/20 (Figure 3.4). This type of sensor allows to measure displacements at maximum 0.009 [mm] resolution, defining the smallest change in distance sensor can detect. Linearity, as the largest deviation between an ideal straight-line characteristic and the real characteristic, is kept in a level of +/- 0.04 [mm]. Maximum supported sampling frequency is 54 [kHz] but for actual test application sampling frequency at level of 10 [kHz] was chosen. The distance output voltage is in a range +/- 5 [V], which gives 0.5 [V] over 1[mm] of displacement for measurement range of this specific sensor.

Due to very minor temperature changes in measurement environment laser temperature drift was negligible (0.02% of range/K).
b) Accelerometers for external source vibration measurements

Due to the fact that additional testing devices are used in the same laboratory where the vibrations measurements of axles were conducted it was necessary to ensure that no other source of vibration is influencing measured signals. For this reason accelerometer was mounted at the test bench frame, measuring total vibration of test bench rigidly connected to the foundation. In this way possible transient vibrations originating from different sources than rotating axle could be detected. Bending load, simulating loads coming from real train, was applied to the axle by means of hydraulic actuator at centre of axle. Because of that any deviation of force value generated by this unit had direct effect on deflection of axle. To be able to measure possible vibrations generated by hydraulic actuator accelerometer was mounted at the actuator head, and additionally force value of actuator was stored along with vibration data.

Accelerometers used during the test were of type 4508 DeltaTron, with measurement frequency range 0.3 [Hz] to 8 [kHz] (Amplitude), and measuring range −/+700 ms² peak (−/+/71g peak).

Signals measured by accelerometers were analysed and during whole mentioned axle tests no additional sources of vibration have been found which could possibly influence low frequency vibration generated by rotating axle.

c) Laser transducers acquisition module

Signal measured by laser transducers was collected by means of acquisition module NI 9234 (Figure 3.5a). This is an AC/DC Analog Input dynamic signal acquisition module for making high-accuracy measurements. It was specifically designed for high-channel-count sound and vibration applications [82]. The four input channels of this module are capable of simultaneously digitize signals at rates up to 51.2 [kHz] per channel with built-in anti-aliasing filters that automatically adjust to specified sampling rate. This module offers 24-bit resolution and 102 [dB] dynamic range.

Sampling rate value chosen for measuring axle vibrations (10 kHz) was applied to each of module channel. It was possible due to the fact that analog-digital converters (ADC’s) used in this acquisition module are channel-dedicated and user specified sampling rate value is not divided by number of channels (as it happens when device...
without simultaneous sampling capabilities use multiplexing to connect each channel to its ADC.

Connectivity between acquisition module and laser transducer was provided by BNC cables.

d) Accelerometer signal acquisition module

For accelerometers signal acquisition the NI 9239 data acquisition module was used (Figure 3.5b). This device is a 4-channel, 24-bit C Series analog input module. It is designed to provide high-accuracy measurements for advanced data acquisition systems. Module contains measurement-specific signal conditioning (including antialiasing filter), bank and channel-to-channel isolation options, allowing ±10 [V] measurement range, and 24-bit resolution. Maximum allowable sample rate is at level of 50 [kS/s] per channel simultaneous sampling.

e) Data acquisition chassis

To ensure possibility of recording and then analysing measured signals connectivity with PC was established by means the NI cDAQ-9172 chassis. It is an eight-slot USB chassis designed for use with C Series I/O modules, capable of measuring a broad range of analog and digital I/O signals and sensors using a Hi-Speed USB 2.0 interface. This chassis supports different trigger modes, such as start trigger, reference trigger, and pause trigger with analog, digital, or software sources. It supports four independent high-speed data streams allowing for up to four simultaneous hardware timed tasks.

Figure 3.5 Data acquisition system:
   a) laser transducers module   b) accelerometer module   c) modules rack

credit: National Instruments Corp.

3.4.2. Signal processing

Rotating or reciprocating mechanical devices are generating vibrations during their work. Vibration signal, when properly measured and analysed, gives valuable information for estimation if device is operating in its designed condition or maintenance/troubleshooting needs to be performed. Vibration signal analysis always deals with dynamic events like forces and displacements which are functions of time. To
be able to properly understand the influence of vibration on the machine it's necessary to understand the type of vibration signal, namely perform signal classification. Determining to which class a signal belongs (i.e. periodic, random or transient) is even more important when damaging effects of vibrations are of interest, such as in fatigue analysis.

Signals of interest for low frequency vibration are periodic vibrations signals which occur whenever repeating phenomena appears (such as crack breathing mechanism in rotating component).

Periodic signals and most transient signals are the deterministic type of signals. In real life signals occurs as mixed combinations of the signal classes, for example periodic signals with background noise, and analysis methods applied in this case are more complicated.

To understand vibrations, and to obtain correct diagnostic information about condition of machine or device with vibrations are measured, proper parameters of signal measurement and analyse methodology needs to be ensured.

Vibration analysis starts with a time-varying signal from a transducer. From the output of transducer to the result of vibration analysis process, a variety of options are possible to apply to process signal. Signal processing applied for laboratory experiment for low frequency vibration of railway axles is explained in following sections.

3.4.2.1. Anti-aliasing

The sampling theorem [83] states that the analog signal x(t) can be uniquely represented by its discrete samples (can be digitised), if it is sampled using a sampling frequency larger than twice the bandwidth. Half of the sampling frequency, fs/2, is generally called the Nyquist frequency, being named so by Shannon [84]. Important outcome of sampling theorem is that, if sampling frequency provided by measuring equipment is fulfilling sampling theorem, than sampled signal, x(n), is an exact representation of the analog signal x(t), which means that it contains all information in the analog signal. By fulfilling the sampling theorem its guaranteed that the spectrum of the sampled signal, within |B| < fs/2, is the same as the original spectrum of x(t) [85]. Proofs of the sampling theorem can be found in many textbooks on digital signal processing (i.e. [86, 87]).

Nyquist criteria, coming directly from sampling theorem implies that ideally the sampling frequency should be no lower than 2.56 times the highest frequency in the data to be processed [88]. It's important to keep in mind that, as [85] clarifies, sampling theorem implies that the signal has to be sampled at more than twice the bandwidth of the signal and not twice the highest frequency of it. A direct implication of the sampling theorem is to make sure before sampling a signal, that it has no frequency content above half the sampling frequency. If sampling theorem (Nyquist criteria) is not fulfilled, a phenomenon called aliasing will occur. Aliasing occurs if a sine signal is sampled with a sampling frequency less than twice the frequency of the sine, which results in a sine signal of a different frequency. This undesirable phenomenon is avoided by removing the frequency content from above half of sampling frequency, which is done by an analog anti-alias filters installed before the A/D converter (ADC). If anti-aliasing filters are not applied false low frequency components are appearing in the spectra that can give wrong information content. Aliasing is a potential problem in any sampled data system.
3.4.2.2. Signal synchronization (triggering)

Due to the fact that most of the analysis process in this work is done in the frequency domain, measured signal needs to be properly treated, to provide optimal transition from time domain to frequency domain. Transition process is done using discrete Fourier Transform (section 3.4.2.6). There are many properties of the Fast Fourier Transform which affects its use in frequency domain analysis. They are coming from the fact that FFT computes the frequency spectrum from a block of samples of the input signal (time record). Moreover the FFT algorithm is based on the assumption that input time record is repeated throughout time. Thus, it is optimal from correct analysis results point of view to provide input signal with integer number of measured cycles. In that case the input waveform will be periodic in the time record and matching the actual input waveform (Figure 3.6). Figure 3.6 [88] demonstrates the difficulty when the input is not periodic in the time record. The FFT algorithm is computed on the basis of the highly distorted waveform (Figure 3.6c) and as result produces results with inaccurate signal spectra.

Non integer number of signal cycles does not cause a problem with analysis of the transient events, but it’s becoming important when it’s necessary to deal with time repeatable signal (periodic) as in case of breathing mechanism of crack in rotating element which is the topic of this work.

To minimise the possibility of introducing into FFT truncated waveform of signal so called triggering was used in data acquisition process. A trigger is a signal that causes starting or stopping the acquisition of data. To achieve reliable trigger signal both axle and electrical engine of test bench was equipped with transducer capable of measuring 1xRev signal. In case of engine proximity probe was installed to measure revolutions of engine drive belt pulley, whereas axle was equipped with additional laser transducer measuring passage of axle rib introduced specially for this purpose. Both signals were compared and due to their coherence signal from engine pulley was used as triggering indicator. Due to the fact that triggering signal is excited once per axle revolution it is called tachometer pulse.

Increasing slope of tachometer pulse was used for triggering data acquisition process from all of transducers. Procedure was done automatically by setting value of signal that
has to be passed (threshold level) to trigger acquisition process. Due to the fact that continuous trigger method was used and acquisition time was based on internal clock of measuring system (time based) and not on number of axle revolutions, additional routine was written in Matlab environment to truncate signals at the end of acquisition process. By following this procedure integer number of axle revolutions was introduced in further analysis.

By repeatedly computing the time of arrival of tachometer signal over a period of measurement time, the actual speed of the axle could also be determined. Knowing the arrival times of tachometer pulse it was possible to relate them to the rotation speed, $v_r(t)$, expressed in rotations per minute (RPM) as $v_r(t) = \frac{60}{N_p}(t_2 - t_1)$ [85].

The axle speed (RPM) versus time from one of the tests is illustrated in Figure 3.8 and shows that the nominal shaft speed was approximately 9.86 [Hz] (after converting to frequency domain), and at times a deviation of less 0.001 [Hz] occurred in the actual shaft speed between single shaft revolutions. This illustrates stability of speed provided by test bench engine.

Due to high sampling frequency of tachometer channel high accuracy of the RPM–time profile was ensured, thus reducing the uncertainty in the trigger occasions.

![Tachometer pulse](image)

![Speed profile](image)

Figure 3.8 Axle speed determination using once per revolution tachometer pulse
upper: portion of digitised tachometer signal   lower: resulting axle speed
Tachometer pulse (trigger) was also necessary in a process of Time Synchronous Averaging (used to enhance the time-domain data by cancelling noise) [47]. This approach is presented in following subsection (3.4.2.3).

3.4.2.3. Synchronous averaging

Synchronous averaging is a signal processing technique that significantly increase the possibility of detecting and diagnosing faults in rotating machinery by ‘extracting’ the vibration for a particular shaft from the overall vibration [89] and used to improve the signal-to-noise ratio of deterministic signals [85].

The process is based on the consideration that the forced vibration of a shaft is periodic with the shaft rotation. As a result all the data coming from the vibration measurements of rotating element can be averaged across cycles, thus eliminating the noise components due to disturbances not related to the rotation.

To achieve correct results out of data recorded during axle vibration test signal processing technique had to be performed before averaging process was started. First of all along with vibration signal measurement proper (giving a reference phase angle) tachometer signal was recorded (section 3.4.2.2). This signal was used to synchronize the sampling of the vibration signal with the rotational frequency of the axle, which means digitally resampling the data and converting it to the angle domain. Tachometer signal was used assuming that the RPM does not change substantially during one cycle, so in approximation the speed was constant during each cycle. This assumption could be stated also due to fact of stable rotational speed of test bench engine (Figure 3.8) used to rotate axle and also that there is considerable inertia of railway axle thus preventing rapid changes of the rotational speed.

Time domain data was resampled to provide signal values at positions indicated by points in angle domain. Resampling process could be done arbitrarily accurately by using an upsampled vibration signal, and then using a linear interpolation algorithm. This technique was developed by Hewlett-Packard in the 1980s and has become the industry standard for synchronous sampling of rotating machinery signals [85].

After resampling operation each data point was representing equally spaced positions around a rotation cycle and not equally space points in time. In this way any perturbations in axle speed were not influencing number of samples across axle circumference, that is, sample rate in the angle-domain was at level of N/2π, where N is number of samples (it is necessary to keep in mind that sample rate in angle domain also has to fulfil the requirements for avoiding aliasing (section 3.4.2.1)).

The synchronous samples were then divided into segments of one rotational period, and ensemble-averaged across the angle domain to attenuate the non-synchronous frequencies. As a result averaged signal was treated as time-domain signal for one revolution of axle.

Precision of this process is highly dependent on the accuracy of the tachometer signal. Any error in this signal will make the synchronous vibration appear slightly non-synchronous, and cause it to be attenuated during the averaging process [89].

43
The scheme of synchronous averaging process applied in axle vibration analysis is presented in Figure 3.9.

![Synchronous Averaging Diagram](image)

Figure 3.9 Synchronous averaging

3.4.2.4. Order analysis and windowing

When dealing with vibration analysis of rotating components it’s desirable to perform order tracking analysis, meaning to extract information about vibration signal level in relation with specific rotational speed of analysed component. Order in that case means the number of occurrences of specific event in one full revolution of examined component. As result it’s possible to obtain integer number of event occurrences (like 1x Rev, 2xRev etc.) or in case of gearboxes or bearings non integer number of event occurrences.

Due to the fact that the spectrum of signal is not constant in length of measuring block (due to slight deviation of rotational speed) phenomena called smearing can arise. The smearing effect causes the peak of signal spectrum to decrease, whereas frequency bins on both sides of the peak are increased, resulting in transferring of power from one band to another. Leakage creates peaks at wrong frequencies, nonexisting peaks, or change in amplitudes of existing peaks [90]. Smearing and leakage reduces the ability to correctly observe peaks in the signal spectrum, due to creation of ‘false’ peaks and reducing spectral resolution.

To avoid leakage phenomena so called windows are used to precondition the signal before entering FFT process. Windows in this case are special functions by which analysed signal is multiplicated in time domain (or convoluted in frequency domain). This operation compress signal at the ends of window function thus allowing FFT processor to work on non-truncated sine waves (both ends of wave entering FFT process is compressed to zero). This operation has several implications, depending on the type of window function used. For example very popular, general application, Hanning window
gives very good frequency resolution but relatively poor amplitude accuracy, flat-top window gives poor frequency resolution but highly accurate amplitude values, etc. Usage of window functions is not intuitive, requires experience and frequently rise necessity of trial and error approach to choose window allowing best analysis results.

In order to avoid the leakage effects discussed above, different approach was used, namely synchronous sampling was applied instead of sampling the data with fixed sampling frequency. According to principles of discrete Fourier Transform each frequency line corresponds to sine wave with specific periods in the time window. This implicates that correct measurement of desired order requires sampling synchronously sampled signal in matter to ensure proper order resolution. Its also necessary to ensure that highest order which is about to be measured is below $N/2$ frequency line (where $N$ is the number of samples). To estimate required number of samples following equation can be used [85] $N = 2O_{max}n_o$, where $O_{max}$ is the highest order under consideration, and $n_o$ is the order resolution.

Conversion from time based sampling to angle based sampling was done using resampling method with interpolation as described in section 3.4.2.3. According to [85] if the resampling was perfect no time window should be necessary prior to computing the FFT. However a flattop window was used to get the accurate amplitude level accuracy, in case the resampling process has failed at some instance.

### 3.4.2.5. Initial vibration level extraction

During analysis process attempt to remove axle initial vibration signal generated by axle shape imperfections was performed. At first laser transducer signal recorded from one full revolution of axle rotating at slow speed was recorded. This signal was stored as reference value for base level vibration removing process. In next steps, signal from measurements of axle vibrations coming from normal operating speeds was segmented based on tachometer pulse and resampled to obtain the same samples in same positions as reference signal. Then reference signal was subtracted from remaining samples, which after this process were reassembled back in full length signal. In next step FFT analysis was done to investigate possible increase of vibrations caused only by the breathing mechanism of propagating crack. After performing analysis of signals before base vibration level extraction and after this process, this approach was dropped. Reason for this was coming from the fact that more errors due to post processing could be introduced in analysis process, mostly due to fact of possible thermal expansion of axle and slight movement inside side bearer seats, thus bringing possibility of measuring different imperfection of axle surface and introducing high errors during reference vibration signal extraction.

### 3.4.2.6. Fast Fourier Transform (FFT)

Signals coming from transducers used in experiment are measurements of complex vibration waveform. This complex waveform is a summation of all the vibrations present at measurement location. The time waveform can be converted to a frequency spectrum in order to indicate from where the vibration energy is coming from, by representing its original components on a frequency versus amplitude diagram.

This is known as a vibration spectrum and is extremely valuable for fault diagnosis, thus frequency analysis is the essence of vibration analysis and enables the satisfactory resolution of most machine problems [88].
The conversion from time domain to frequency domain is achieved by use of an algorithm called the Fast Fourier Transform (FFT). FFT is a nonparametric method of investigating the spectral content of a signal. Nonparametric method of spectral investigation of signal was chosen due to its simplicity and reliability, over parametric method (i.e. maximum entropy, autoregressive moving average, etc.) which in many cases deliver unreliable results and except very rare special cases, it should be avoided, as the risk of misinterpreting the results is very high [85]. Due to fact that vibration signal from experiments was resampled and converted from time base into angle base, the signal was assumed to be periodic in samples blocks which is one of the fundamental statements of FFT.

3.4.3. Signal processing summary

Signal processing techniques were applied allowing proper analysis of vibration signal coming from rotating railway axle. Sequence and type of methods used in signal processing stage are presented schematically in Figure 3.10.

Vibration signals were acquired by means of equipment described in section 3.4. During acquisition process basic signal treatment was performed, namely antialias filtering and signal triggering by means of tachometer pulse. After this process acquisition data was saved for further analysis. In next steps signals were filtered by means of lowpass filter, and resampled due to conversion into angle domain (N samples along axle circumference). By ensuring equal spacing between samples, and thus equal and rotation speed independent number of samples for each axle revolution time based synchronous averaging technique was applied to attenuate nonperiodic components of signal. Averaged vibration signal was then passed to FFT algorithm and changed from time domain to frequency domain allowing observation of spectral components and their amplitudes. Variation of amplitudes of specified components (namely orders 1 to 7) were saved and compared allowing trending of vibration signal along duration time of experiment.
Code for signal processing and analysing was created in Matlab and validated using LabView toolboxes (Signal Express, Sound and Vibration) proving to provide reliable results.

3.5. Experiment results

During test campaign multiple tests were conducted. As previously stated, that not all of the axles under investigation faced a crack development, thus not all the results will be described here. This section is focusing mostly on discussing results from cracked axles but for reference also analysis result from non-cracked axle is presented.

Due to the fact that vibrations under investigation were located at low frequency range (up to 100 Hz) results are presented in displacement units \([\text{mm}]\), thus overall vibration levels are presented in peak values.

Example of vibration level increase due to propagating crack is presented in Figure 3.11. In this figure time domain vibration signal is plotted in polar coordinate system and its corresponding spectral components (up to 7th order) for initial (Figure 3.11a) and final (Figure 3.11b) stage of test is presented. Difference in total vibration amplitude is caused by reduced bending stiffness of railway axle due to reduced resistant cross section caused by propagating transversal crack. Constant load applied at canter of axle is causing greater deflections when axle is at angular position at which crack is opened (breathing mechanism), thus decreasing distance to transducer.

![Figure 3.11 Polar plot of vibration from one revolution of axle and its corresponding spectra](image)

a) initial stage of test  b) final stage of test (crack developed)
From the point of view of vibrational analysis it is very common to trend the overall values of component displacements, thus allowing to observe eventual increase or instability in overall vibration values. In fact, this is the most basic form of vibration monitoring. Also for this test abovementioned method of presenting data was used.

During experiment a number of nxRev harmonics appear in the measured signal and after trending process the increase in the level of vibration became evident for some of them. As the result of test, it was possible to observe that the most sensitive for crack occurrence are the 1xRev to 3xRev harmonics.

Figure 3.12 compares the trend of the amplitude of the first seven harmonics (1xRev to 7xRev) with the number of axle rotations (cycles) in the fatigue experiment for 4 selected tests which ended with axle failure (critical crack size development). Vibration signal presented on those figures has been processed according to description in previous subsections of this work (section 3.4).

Photos of related cracks at final stage of the tests are also included in the graphs.
Analysis result of vibrations signal coming from cracked axle confirms, as expected, that a rotating cracked axle produces an increased level of bending vibration at some integer multiples of the rotation frequency. The most sensitive for crack occurrence and propagation are first three harmonics. First and second harmonic were showing almost constant increase, while third one was exhibiting some significant instability. Both cases contains possible diagnose information, due to the fact that increase of vibration amplitude as well as decrease can be crack occurrence indicator [49].
In Figure 3.13 and Figure 3.14 trend of first and second vibration signal harmonic is compared among five different axle tests. As can be easily seen noticeable increase of vibration signal occurred at the final stages of tests. In all tests except test with axle no.5 crack was found after removal of axle from test bench followed by magnetic particle test.

Result of this analysis proves that vibration signal is very sensitive for crack occurrence and its applicability for online crack monitoring of railway axles is worth further investigation.
3.6. Experiment conclusions

Based on the analysis of experiment results, it can be seen that the first, second, and third harmonic component is important for the proper real time condition monitoring of railway axles.

The 1xRev component shows the largest increase in amplitude at the final stage of the tests (and hence with the abrupt increase of the crack size). The amplitude of the 2xRev component is also sensitive to the number of cumulated loading blocks. The 3xRev component shows a lower sensitivity to the number of block loading repetitions, still a nearly monotonically increasing trend, but the trend is less clear, probably on account of disturbances such as thermal effects that may produce a bow of the rotating axle. Finally the amplitudes of the remaining 4xRev to 7xRev components remained very low during the whole tests with a slight increase in the final stage of the test. In conclusion, the 1xRev, 2xRev and in some cases also 3xRev harmonic components of vibration signal appear to be the best suited to be set in relationship with the presence (and possibly with the size) of a propagating crack in the axle. It is necessary to keep in mind that not only increase of amplitude of vibration signal can indicate presence of fault. Some experimental results are showing decrease in amplitudes during initial stage of crack development, which could be explained by influence of opposing phases of generated signals [91].

In Figure 3.13 and Figure 3.14 trend of 1xRev and 2xRev harmonic from final stage of tests has been plotted, including one signal from axle without crack (for reference). As can be seen increasing amplitude can be observed at each case except non-cracked one, which was true for all the comparisons with non-cracked axles experiment results.

Figure 3.15 has been included as a reference, and it presents analysis result in case where axle was running without any crack development for whole test (up to 10 million cycles). No significant increase in trend of vibration signal harmonics can be observed.
CHAPTER 4

Finite Element Model

4.1. 3D Finite Element Method model

3D analysis by finite element method was used to investigate vibrations inducted by breathing mechanism of crack existing in railway axle under working conditions (applied loads and rotations). Due to the fact that created model was representing the exact size and shape (with some simplifications typical for 3D analysis) of railway axle, thus its analysis in a sense of calculation time was more expensive than in case of simplified models [49]. Nevertheless this type of model was used, to provide results, which could serve as benchmark for simplified models created in future representing the same analysis scenarios. For the analysis purpose Abaqus software was used.

4.2. Geometrical model of railway axle

Geometrical model of railway axle was created according to technical documentation delivered by axle manufacturer. For the analysis purpose hollow axle with outer diameter of 160 [mm] and bore diameter of 55 [mm] was used. This is typical axle used in passenger trains. Due to the fact that railway axle consist many geometrical details in transition zones these elements were simplified in order to reduce size and number of finite elements applied in FEM analysis which in other case would be necessary to correctly represent them.

Additional sections were defined on axle surface representing locations of auxiliary equipment used during tests campaign, namely load application section, rotational velocity application section, supports section and deflections measurement section (Figure 4.6).

During modelling stage crack shape and size was defined, based on requirements of simulated scenario. Both straight line cracks as well as elliptical shape cracks were modelled and applied in axle cross section representing simplified transversal crack.

The complete 3D model shown in Figure 4.1 was exported to pre-processor of software in which numerical analyses were conducted.
4.3. Finite element model of cracked railway axle

Finite element model of axle was discretised by means of second-order interpolation, reduced integration C3D8R elements. These are the isoperimetric hexahedral elements from solid element library, capable of representing three-dimensional entities. Due to the fact that analyses were conducted by means of explicit method, the lumped mass formulation for all elements was applied by explicit solver code. With explicit methods, the performance bottleneck tends to be the element computations, thus computationally inexpensive lower-order integration elements were chosen. Reduced integration elements are using one Gauss integration order less than required for exact integration of strain energy density over element volume to obtain element stiffness matrix. Fully integrated linear hexahedral elements (C3D8) use two integration points in each coordinate direction. Therefore, fully integrated linear hexahedral elements have eight (2x2x2) integration points. Reduced integration C3D8R element employed in analysis is using only one integration point which is computationally more efficient. Due to reduced number of integration points except savings on computational time, also flexibility of such element is increased, thus decreasing shear locking in certain cases of problems [92].

Comparison of 8-nodes element using full and reduced integration method is presented in Figure 4.2.

Except advantage of requiring less calculations and reducing shear locking possibility reduced integration elements have major drawback. Their stiffness calculation scheme can cause mesh instability resulting in so called hourglass effect. Due to fact that single order C3D8R element has only one integration point it has specific bending behaviour. During bending its changing size but strain is not detected due to the fact that there is no change in length of element centroids passing through the integration point, as presented
schematically in Figure 4.3. This means that normal and shear stresses are equal zero at integration point and there is no strain energy generated by deformation. This state is called zero energy mode, which is nonphysical response and produces meaningless results usually in form of excessive flexibility of structure [94].

![Figure 4.3 Hourglass effect in reduced integration element type due to applied moment [94]](image)

One of the common causes of excessive hourglassing when dealing with contact problems is time when contact between two surfaces is realized at single node [93]. This effect can be reduced by distributing contact constraint among several nodes, but due to variable contact location depending on breathing mechanism of analysed cracked axle and thus its nonlinear behaviour this remedy for hourglassing cannot be imposed.

Effective method to control this phenomenon is the enhanced hourglass control formulation, which was also used in the simulations presented here. It is a robust formulation based on enhanced strain methods and not requiring user-set parameters. The enhanced hourglass control formulation is also usable in providing highly accurate results for elastic bending in the presence of coarse mesh [93], and this advantage was used to reduce mesh density in regions located in a distance from crack, allowing further reduction in total calculation time. Artificial energy used in hourglass control was verified and showed that its in a level of less than 5% of internal energy (Figure 4.4). Although recommended level of 1% contribution is violated this situation is acceptable due to coarse mesh used in others parts of axle body [93].

The default first order element formulation in ABAQUS/Explicit is not appropriate for a structure subjected to several revolutions [95], thus the second-order accurate formulation elements were used in case of cracked axle analysis. These elements are capable of representing all possible linear strain fields. Thus, in the case of many problems (elasticity, heat conduction, acoustics) much higher solution accuracy per degree of freedom is usually available with the higher-order elements [96]. Second-order accuracy is usually required for analyses with components undergoing a large number of revolutions (>5) [93].

Stability of numerical solution with parameters defined in abovementioned way can be seen in Figure 4.12, where axle deflection due to breathing mechanism of cracked section is compared with axle without crack. In both cases no deflection drift caused by excessive flexibility is observed, while it was evident for first order reduced integration elements without hourglass control algorithm (not reported in this work).

Cracked axle model was meshed using sweep technique with advancing front algorithm and in its final form (Figure 4.5) was used to perform analyses.
Analysis was conducted in double precision which according to [93] is more suitable for type of analysis which requires high number of revolutions.

4.3.1. Model definition

Aim of model created for simulation purpose was to predict low frequency vibration caused by breathing mechanism of crack inducted in axle under rotating bending load. Thus, no crack propagation path was studied, and modelling effort was focused on crack breathing phenomena as its one of the most expressive symptoms of a crack occurrence[49]. For simulation case hollow axle was used as described in section 4.2. For simulation purpose material parameters were introduced according to parameters of A1N grade steel, from which axle was produced, and also one of the most widely used axle steels[5].

Following values are describing mechanical properties of this type of material:

- Ultimate tensile strength: 550 [N/mm$^2$]
- Elastic modulus: 210000 [N/m]
- Poisson ratio: 0.26 [-]
- Density: 7300 [km/m$^3$]

Cyclic properties, in terms of a Ramberg–Osgood relationship:

- Cyclic elastic modulus: 200,000 [MPa]
- Hardening exponent: 0.168,472
- Hardening coefficient: 1036.24 [MPa]
- Cyclic yield stress: 364 [MPa]
Complete model was divided into 5 or 6 main partitions depending on simulated case, namely partition of the crack, where mesh density was increased, two partitions for left and right side of axle where supports were located and rotational velocity applied, one partition for load application (two load partitions in the ‘railway track’ simulation scenario described further) and one partition where measurements of deflections were performed. Schematic representation of model partitioning is presented in Figure 4.6.

Individual partitions are representing areas where load and supports were defined in laboratory tests, as well as measuring section where deflections were evaluated and crack section where crack was found during tests.

![Partitioning of cracked axle model](image)

**Figure 4.6 Partitioning of cracked axle model**

It’s worth noting that model defined in abovementioned way allowed to conduct simulation of fully rotating part, under user defined loading and excitations affecting either axle supports or introduced by actuator through which loads were applied. In this way both static loading as well as dynamic tests could be conducted, allowing to investigate crack breathing mechanism influence on overall component vibration in many types of simulation scenarios, defined by the user.

4.3.2. Cracked area modelling

Crack breathing mechanism as described in State of the Art section 2.1.2 is a phenomenon of opening and closing crack lips due to bending and rotation of a component. Reduction of cross section area caused by opened crack, thus reduction of bending stiffness of component is causing additional deflection without change in loading parameters. Due to rotation, crack is changing its position in relation to force direction causing bending, and as results starts to close (due to increasing compressive stress). As soon as crack walls are starting to be in contact stiffness of component is gradually coming back to initial value (defined by material parameters) and as consequence deflection is reduced. Described breathing mechanism is in fact the result of the stress and strain distribution around crack area. In general case, when static loads overcome dynamic ones, crack breathing mechanism is governed by angular position of shaft with respect to stationary load direction [49].

Crack in the axle, as already mentioned, was introduced in model as fixed size discontinuity in finite element mesh. Since crack breathing mechanism was most important phenomena to investigate during simulation it was necessary to ensure proper behaviour of model at this specific area. For this reason contact parameters were defined at both sides of cracked specimen walls (crack lips).
The use of contact elements in Finite Element analysis is highly demanding in terms of computational effort required, due to fact that stress between elements is transmitted only when they touch each other. From numerical point of view it is severely discontinuous form of phenomenon and thus it’s undergoing highly nonlinear behaviour. Moreover, due to the fact that axle model was defined as completely flexible body, crack was generating time-dependent boundary condition which had to be calculated at every calculation step. Important parameters in the definition of contact elements are: allowable penetration (if any) of surfaces in contact (especially in normal to surface direction), and eventual friction which can influence results when tangential movement of contact surfaces occur. Contact elements definition also has to take into account possible separation of crack surfaces as soon as tension value will allow for crack to open (Figure 4.10).

To take into account all the abovementioned considerations 'surface to surface' type of contact was prescribed at the cracked area. This contact definition has many advantages which can be used in application of crack breathing mechanism analysis. Important characteristic of surface-to-surface contact is fact that surfaces assigned to act as contact pairs are not simply represented by set of discrete points but each contact constraint is formulated based on integral over the region surrounding a slave node. In this way algorithm used to calculate contact parameters involves coupling among slave nodes as well as tends to involve more master nodes per constraint [93].

Moreover, algorithm used in this type of contact is less sensitive for mesh grid applied in both surfaces, meaning that even if meshes are not matching each other more accurate contact stresses are calculated in comparison with different contact types available, which has considerable effect on modelling time and precision of solution. Surface-to-surface contact shows reduced sensitivity on results if roles of surfaces are swapped (master surface becoming slave and opposite) [97] but to enhance accuracy and account for better performance mesh on crack surface defined as ‘slave’ in contact pair was modelled as more refined in comparison with ‘master’ surface, thus role of this surface was imposed as ‘slave’ in all simulation scenarios.

Except all abovementioned advantages, the most important is the one coming from discretization method used in surface-to-surface contact type, in particular offering better distribution of contact forces among master nodes, and computing average penetrations and slips over regions in contact, thus having smoothing effect that avoids snagging and reducing likelihood of master nodes penetrating slave surface due to numerical errors (Figure 4.7). As [98] states, some penetration may be still observed but large, undetected penetrations of master nodes into slave surface do not occur.

![Figure 4.7 Reduced possibility of master-slave surface penetration in S-to-S contact type [98]](image)

57
Due to the fact that the railway axle is a bulky component many nodes in cracked cross section are in contact at the same time which is computationally challenging, but as long as both crack surfaces are having nearly opposite normal (which is true for deflection values observed during simulations) still surface-to-surface contact type yields best results [98].

Similar to physical part, when two crack lips in numerical model establish a contact local stresses give rise to inter-penetration of contact surfaces. To prevent mutual penetration of opposite surfaces contact constraint reinforcement takes place, which is numerical method applied to resist penetrations. In the calculations presented here ‘hard contact’ pressure-overclosure behaviour was chosen. This type of formulation belongs to the family of direct enforcements methods which in general applies strict enforcement pressure-penetration relationship using Lagrange multipliers. In this approach when no contact is detected at surfaces where contact pairs are defined no multiplier is used, and thus no contact pressure is required. As soon as condition is changed and surfaces start to be in contact, constraint enforcement is applied and contact pressure is increased. For this case contact virtual work contribution can be written as \( \delta W_c = \delta p h + \delta h \) [93], where ‘p’ is the Langrangian multiplier (increasing contact pressure) and ‘h’ is the overclosure. In hard contact pressure-overclosure formulation two cases are possible: \( p=0 \) for \( h<0 \) when crack is opened and \( h=0 \) for \( p>0 \) when crack is closed. Hard contact behaviour and corresponding penetration-pressure graph is presented in Figure 4.8.

![Figure 4.8 'Hard contact' formulation for pressure-overclosure behaviour [98]](image)

Due to torque applied to axle torsional vibrations may occur, thus introducing relative tangential motion of crack surfaces while in contact. To take into account this behaviour finite sliding formulation was applied in contact definition. In this formulation point of interaction on master surface is updated using true representation of master surface, not approximated as planar representation what takes place in other methods. Contact area in this case is always based on the current configuration of contact surfaces. This approach allows to obtain physical results even if relative tangential motion does not remain small, but also introduces more nonlinearity, thus increasing computational cost. Finite sliding allows to use robust contact search algorithm to avoid missing contacts in between contact surfaces, avoiding additional errors in final results.

For tangential relative movement of crack surfaces isotropic friction was defined with friction coefficient at level of 0.2 [49]. Penalty method for friction estimation was chosen. This method is based on approximation of stick with stiff elastic behaviour. By default the same penalty stiffness used in hard contact is used for frictional constraints.
Penalty method allows transition from stick to slip friction (Figure 4.9), where the elastic slip $\gamma_{\text{crit}}$ is calculated as: $\gamma_{\text{crit}} = F_f l_i$, where $F_f$ is the slip tolerance, and $l_i$ is the "characteristic contact surface length" [99].

![Figure 4.9 Friction model in contact area [99]](image)

For the simulated cases different crack size and shape was applied. Crack breathing mechanism occurred in each simulated case, due to applied load and rotation. Example of opened crack shape is presented in Figure 4.10 while contact area (presented by pressure generated at crack surfaces due to contact) for two different cases (50% and 25% cracked cross section of axle) of closed crack is presented in Figure 4.11. Reduction in axle bending stiffness and, as result, additional bending caused by gradually opening crack is presented in Figure 4.12, where influence of crack breathing mechanism (50% cracked cross section) on axle overall vibration at the speed of 590 RPM is plotted (initial deflection caused by applied load has been removed).

![Figure 4.10 Specimen in crack opening situation (100x magnified in crack opening direction)](image)
4.3.3. Boundary conditions

In order to conduct simulation of desired scenario proper boundary conditions were defined. In the presented axle model two types of boundary conditions were applied, namely displacements and rotations.

Model supports were localized at locations where the wheel seats are defined. This boundary condition was applied as displacement type. At both ends of axle lateral and vertical translational degrees of freedom (DOF’s) were fixed (namely U2 for vertical translations and U3 for lateral ones). Different condition was applied for axial translations (U1). In this case one end of axle had fixed U1 DOF and at second end U1 DOF was released. In this configuration axle deflections were not restricted, allowing axial movement of axle during bending.
For rotational velocity application kinematic coupling was defined between wheel-seat area and additionally created point of reference. Thanks to kinematic coupling rotational velocity applied at reference point was transferred to desired surface of axle, thus producing rotation controlled by parameters specified by user (constant or variable speed). Boundary condition configuration is presented in Figure 4.13.

Due to the type of simulation (explicit) loading and rotation were applied in a way to avoid sudden amplitude changes causing convergence problems. This was done by introducing tabular values of gradually rising amplitudes of force and rotation speed at specified model degrees of freedom.

4.3.4. Load definition

For the load definition two separate cases were considered. First one where the load was defined at the central part of axle was representing the scenario defined during laboratory experiment. In this case loading was transferred to the outer surface of cylindrical surface located at centre of axle. Load was defined only in vertical (U2) direction, keeping to zero the force applied along the other degrees of freedom. In this way no disturbance from eventual axial and lateral vibration was interfering with loading force. Load value was defined at the same level as in case of laboratory experiment, meaning approximately 170 [kN] (varying in case of different simulation scenarios). In this way a uniform load history was applied.

Similar approach but with different load sections was used in case of simulation representing railway track case. To recreate close to reality situation loads were moved to both ends of railway axle, where in reality axle boxes and journal bearings are located, transferring loads coming from vehicle weight as well as dynamic loads from train-track interaction.

Both loading conditions are schematically presented in Figure 4.14.
4.3.5. Damping in model

Due to the fact that real components are losing energy during vibration, damping was introduced into numerical model. In general, energy loss is happening due to three main reasons in numerical models: nonlinearity of members, energy radiation and inherent damping. Damping capacity is defined as the ratio of the energy dissipated in one cycle of oscillation to the maximum amount of energy accumulated in the structure. There are many mechanisms of damping in a structure and the ones best identified are material damping and interfacial damping. The material damping contribution comes from a complex molecular interaction within the material, thus the damping is dependent on type of material, methods of manufacturing and finishing processes [100]. Since the material microscopic properties may differ from one sample to the other one, estimation of material damping may be complicated.

In presented model Rayleigh damping was applied. The damping parameters were chosen to produce low damping in the frequency range of interest. Derivation of this parameters was done according to [101] and consisted of following steps:

Assuming linear viscous damping focusing on Rayleigh damping the damping matrix can be defined as a function of mass and stiffness matrices. The damping ratio for the n-th mode of a system is:

$$\zeta_n = \frac{\alpha}{2\omega_n} \frac{\beta}{2\omega_n}$$

Convenient procedure used to determine α and β coefficients was introduced by [102]. Modes of interest were covered by selecting a frequency range of ω to 7ω (which means 1xRev to 7xRev mode) and desired amount of damping. Then ∆ was calculated using:

$$\Delta = \zeta \frac{1 + R - 2\sqrt{R}}{1 + R + 2\sqrt{R}}$$

where ∆ determines bounds on the damping ratios that are imparted to those modes within the specified frequency range, and R defines number of frequency orders under consideration (Rω). By following this approach the modes in the given frequency range had a damping ratio bounded between ξ+H and ξ-H. Next α and β were calculated from:

$$\alpha = 2\zeta \sigma \frac{2R}{1 + R + 2\sqrt{R}}$$
$$\beta = 2\zeta \frac{1}{\sigma} \frac{2}{1 + R + 2\sqrt{R}}$$
As it was proven in [101] larger R results in larger $\Delta$ which is not wanted. This means that although it seems better to choose a larger R to cover a larger range of frequencies, it results in larger H which decreases the accuracy of the damping ratio used as the basis of formation of the range. In presented work R covering up to 7xRev frequency range was used. Resulting damping ratio curve for 5\% damping and 10\% damping is presented in Figure 4.15. By introducing Rayleigh coefficients calculated in abovementioned way vibration signal decay could be investigated. For this purpose axle was fixed at one end and 5 [mm] deflection was defined at the opposite end. After sudden release of force causing deflection axle harmonic vibration could be observed with amplitude decay dependent on damping ratio. In Figure 4.16 free vibration of the axle is plotted, comparing vibration amplitudes corresponding to 10\%, 5\% and ‘no damping’ case. Its necessary to note here that ‘no damping’ means that no Rayleigh damping coefficients were introduced, but damping associated with volumetric straining introduced by bulk viscosity was present (both linear as well as quadratic, with coefficients equal 0.06 and 1.2 respectively).

![Figure 4.15 Rayleigh damping coefficients and resulting damping ratio](image)

![Figure 4.16 Influence of damping on axle vibration decay](image)
4.4. Simulation scenarios

During numerical simulations stage of research few different scenarios have been taken into account. The main division was made between laboratory experiment representation and simulation including disturbances coming from train-track interaction. Train-track scenario was considered to be more similar to real life situation in which train is operating, thus called ‘railway’ case. Details about differences in both models are presented in following subsections.

4.4.1. Laboratory case setup

In this simulation axle was modelled in a way to reproduce the configuration defined during laboratory test. Supports were defined at wheel seats, and rotational velocity was applied at axle end. Load value of 170 [kN] was applied at centre of the axle, at area of actuator attachment used in test bench. Rotational velocity was set at level of 590 RPM. Crack was located in between load area and wheel seat where supports were defined. No disturbances coming from other sources were introduced. Resultant vibration was related to the breathing mechanism in crack area only. Different sizes of crack were introduced and results plotted for comparison after finishing the simulation (presented in section 4.5). Axle deflection measurements were taken at location where lasers transducers were localized during experimental test campaign. Described arrangement is presented in Figure 4.17.

![Figure 4.17 'Laboratory case' model definition scheme](image)

4.4.2. ‘Railway’ case setup

To allow investigation of scenarios being closer representation of real axle working conditions so called ‘railway’ case was discussed and prescribed. For that matter load was moved from centre of axle to its ends, where axle boxes in real train are located. Velocity was applied at both wheel seats simulating torque coming from engine. This arrangement was common for all scenarios simulated under that ‘case’. Differences were occurring in introduced cracks size as well as excitations coming from rail and/or wheel irregularities. Excitations generated by wheel-rail contact were introduced at the supports as additional displacements in vertical direction, having direct influence on total axle vibrations. Special
interest was paid to wheel out-of-roundness excitations which are generating vibration signal with periodicity close to crack breathing mechanism (1xRev, 2xRev, etc.)

Common arrangement for all scenarios simulated under ‘railway’ case is presented in Figure 4.18.

![Figure 4.18 'Railway' case model definition scheme](image)

Under ‘Railway case’ different scenarios have been simulated, as details are presented in following subsections.

4.4.2.1. No excitation in model

In this simulation scenario axle with introduced crack was simulated without any additional excitation coming neither from rail irregularities nor from wheel out of roundness. Results from this simulation served as a reference situation for investigating the influence of introduced external source disturbances on vibration caused by breathing mechanism of crack.

4.4.2.2. Wheel out of roundness excitations

During simulations two cases of wheel out of roundness (OOR) and their influence on low frequency vibrations inducted by crack breathing mechanism were studied.
Profiles for both new wheel and worn wheel (300.000 km) OOR were created by means of mathematical approach.

Due to the fact that initial (low) level of wheel irregularity exists also in new railway wheels one of simulation scenario was created to account for this situation. For this reason irregularity profile with wide spectrum of wavelengths was created, corresponded to the lowest 20 harmonic orders of polygonalized wheel.

Calculations of assumed in this way wheel out of roundness profile were conducted according to work done by [103] and subsequent steps will be shortly described here.

The harmonic orders $\theta$ are defined in a way that wavelengths are given by:

$$\lambda_\theta = \frac{2\pi R}{\theta}, (\theta = 1, 2, 3, \ldots, 20)$$

Level spectrum $L_w$ of wheel irregularity can be defined as

$$L_w = 20 \log_{10} \left( \frac{\hat{w}}{w_{\text{ref}}} \right),$$

where $\hat{w}$ is the RMS value of irregularity profile $\omega(xw)$.

Initial wheel irregularity level spectrum was than calculated from:

$$L_{w\theta} = 24.7 \log_{10} (\lambda_\theta) + 8.47,$$

which as stated by [103], corresponds to an empirical formula determined by measured data from new wheels.

The amplitude of sine function of each wavelength could then be obtained as,

$$a_\theta = \sqrt{2} \cdot 10^{L_{w\theta}/20} \cdot w_{\text{ref}},$$

where $w_{\text{ref}} = 1[\mu m]$.

The wheel irregularity profile was then evaluated as a sum of 20 sine functions (number of wavelength harmonic orders under consideration) according to:

$$\omega(x_w) = \sum_{\theta=1}^{M} a_\theta \sin \left( \frac{2\pi x_w}{\lambda_\theta} + \psi_\theta \right).$$

Different irregularity profile for each wheel was created by assigning uniformly and randomly distributed phase angles to the sine functions ($\psi_\theta$ from 0 to $2\pi$).

In this way irregularity spectrum with wavelengths in the interval 0.14 to 2.88 [m] was established (where 2.88 [m] is wheel circumference). Wheel nominal radius versus circumferential coordinate is plotted in Figure 4.19(left). To allow visual comparison of irregularity profile with perfectly round wheel profile in Figure 4.19(right) irregularity values were magnified 1000x.

To account for periodicity in signal produced by rotating wheel, for simulation purpose irregularity profile was repeated to fill all the simulation time. Example of repeated OOR signal is presented in Figure 4.20 along with its corresponding spectra.
Different order amplitudes can be found on used railway wheels. In that case the measured deviation of wheel radius is dominated by the third order harmonic and by looking at overall contribution of several first orders, first and third orders are dominating ones. Second order harmonic in that case is decreasing as travelling distance is increasing [104]. Following this observations worn wheel profile for axle test was created using similar approach as for new wheel OOR but with increased amplitude of first and third harmonic, causing greater polygonization of wheel (periodic radial irregularity with three wavelengths around wheel circumference). Worn wheel profile (1000x magnified) in comparison with
ideal wheel is presented in Figure 4.21 while its time repeated signal and corresponding spectra with dominating 1\textsuperscript{st} and 3\textsuperscript{rd} order are plotted in Figure 4.22 and created based on [105, 106].

**Figure 4.21 Worn wheel out of roundness profile**
left: irregularity according to circumferential coordinate of wheel
right: irregularity versus ideal wheel profile comparison (1000x magnification)

**Figure 4.22 Extended worn wheel OOR profile (upper) and its corresponding spectra (lower)**
Wheel out-of-roundness profiles created in abovementioned way led to a simultaneous wheel-rail excitation depending on the train speed at several frequencies in a given frequency range, thus influencing directly vibrations inducted by crack breathing mechanism. Excitation frequency depends on speed of train and can be calculated as $F=\frac{v}{\lambda}$ where $v$ is train speed and $\lambda$ is a wavelength. If just first five wavelengths of wheel irregularity profile are taken into consideration generated excitation frequencies are in the range 5-125 [Hz] for train speed in the range of 50-250[km/h] [104], thus covering the entire frequency range of interest in this analysis.

4.4.2.3. Wheel-rail interaction excitations

To take into consideration influence of both types of excitations acting on the axle during its operation, namely wheel and rail irregularities, rail irregularity profile was introduced into simulation as additional external source excitation.

Rail irregularity profile was converted from space to time domain, allowing to be used as input for simulation case. Different irregularity profiles were used for left and right rail.

Rail irregularity signal covering 100 [m] of track is presented in Figure 4.23 a and b for left and right rail respectively.

![Figure 4.23 Rail irregularity profile](image)

upper: left rail  lower: right rail
To take into account effect of both type of excitation (rail related and wheel related) both signals were added together creating one signal with new amplitude and spectral components. This signal was then applied as variable displacement value acting at supports along vertical degree of freedom. Simulation input for common (rails and wheels) excitations is presented in Figure 4.24 a and b for left and right side of axle respectively.

Figure 4.24 Common excitation for wheel and rail used in simulation

upper: for left side of axle    lower: for right side of axle
4.5. Simulation results

Simulation results presented in all of following plots were post processed to subtract deflection caused by external static load (weight of vehicle), which allows to focus only on crack influence on overall vibration levels of the axle.

4.5.1. Laboratory case simulation results

For the laboratory test equivalent simulation, hollow axle was chosen to allow a comparison with laboratory tests, at least in qualitative terms. External diameter of the axle was equal 0.169 [m] and internal (bore) diameter equal 0.06 [m]. Axle was considered as made of A1N steel grade, with total length of 2.2 [m]. Static load applied at centre of axle and set to be equal to laboratory test one, thus resulting in value of 170 [kN]. Rotational velocity in that case was defined at 590 RPM. Boundary and loading conditions were defined according to description given in Section 4.4.1.

To investigate crack dimension influence on axle dynamics, simulations were conducted in a way to cover cases from non-cracked axle up to 50% of cracked cross section, evenly distributed in steps of 5% crack depth. As a result eleven simulations were conducted totally. Time domain signal of axle deflections obtained from each simulation was converted to frequency domain by means of FFT algorithm, allowing to observe its harmonic components. Spectra components are presented in Figure 4.25 for non-cracked axle and in Figure 4.26 for crack affecting 5% to 50% of axle cross section.

It’s necessary to note differences in vertical (deflection/amplitude) axes magnitudes, which were chosen for sake of results visibility.

![Figure 4.25 Non-cracked axle deflections and its corresponding harmonic components](image_url)
Figure 4.26 Deflection and spectra components for defined crack depth
Figure 4.26 (continued)
In Figure 4.27 trends of displacement signal harmonic components are presented. As can be seen, according to simulation results, crack up to 10% depth is not causing any significant change in dynamic behaviour of axle. Slowly increasing trend of vertical displacement harmonics can be observed when the crack is affecting 10% to 30% of axle cross section. After propagating more than 30% crack is affecting vibrations of rotating axle in considerable way, causing fast increase of vibration signal harmonics amplitudes, specially 1\textsuperscript{st} and 2\textsuperscript{nd} order. Another considerable change in crack influence on rotating axle can be observed at crack depth equal 45% of axle cross section. At this point rather exponential growth of amplitudes can be noticed, indicating that dynamic behaviour of axle is becoming dominated by crack breathing phenomena and consequences coming from this fact (bow, stiffness reduction, etc.). It’s worth noting that no resonance was found during analysis of crack depth between 40 and 50%, thus stable vibration amplitude was recorded.
The 'laboratory' scenario simulation results appear to be in good qualitative agreement with the measurements performed in the full-scale tests on cracked axles presented in Chapter 3. In particular the ratios of the amplitude for the 1st and 2nd harmonic shown in Figure 4.27 are in good agreement with these that can be observed from Figure 3.12. The absolute amplitude of vibration obtained at the end of the tests performed on cracked axles is consistent with the amplitude obtained by means of Finite Element simulations for a crack in the range of 25% to 35% which is roughly consistent with the final crack amplitude found on the specimen at the end of the laboratory test.
4.5.2. Railway case simulation results

As described in Section 4.4.2 ‘railway’ simulation scenario was designed to more closely represent actual working conditions of railway vehicle. For this reason location of boundary conditions was redefined, as well as additional external excitations were introduced into model (details in section 4.4.2.2 and 4.4.2.3). All of the simulations included in this set were conducted on the same model of railway axle, with the same crack shape (elliptical), location (axle mid-span) and depth defined (35%). This approach allowed to exclude eventual influence of differences in finite elements grid definition on numerical results.

The first simulation was conducted in accordance with description made in section 4.4.2.1, thus no external source excitations were introduced into model, neither from wheel irregularities nor from rail profile imperfections. In this case resultant harmonic spectra of axle vibration signal generated by breathing mechanism of 35% deep crack were equivalent to ones obtained during ‘Laboratory’ simulation setup. Peak to peak deflection amplitude in measuring section of axle was recorded to be equal approximately 0.65 [mm]. Corresponding harmonic spectra of deflection observed after application of Fast Fourier Transform are presented in Figure 4.28.

![Figure 4.28 Vertical displacement components (FFT) for 35% crack, no external source of excitation in model](image)

Next realised simulation scenario was including additional excitation coming from wheel imperfections, namely wheel out of roundness (OOR) profile. Initially profile of wheel irregularities resulting in small amplitudes of excitations was used. This OOR profile was reflecting so called ‘new’ or ‘initial’ wheel OOR, named in this way due to the fact that it can be found even on re-profiled or new railway wheels (more details on ‘new’ OOR definition are given in section 4.4.2.2). Fourier expansion of excitation caused by ‘new’ wheel OOR is presented on Figure 4.20. It’s worth recalling that two different OOR profiles were used to excite left and right side of axle. The difference was only due to change in
phase of both signals, thus excitation amplitudes were kept on the same level. Due to this fact, for sake of brevity only one wheel OOR and its corresponding spectra are presented in this work, as described earlier.

Simulation result of rotating axle including excitations coming from 'new' wheel OOR are presented in Figure 4.29.

![Harmonic components (FFT) of axle vertical vibrations influenced by 'new' OOR profiles](image)

**Figure 4.29** Harmonic components (FFT) of axle vertical vibrations influenced by 'new' OOR profiles

upper: axle without crack    lower: axle with 35% crack

In Figure 4.29 (upper picture) harmonic components of vibration signal generated by axle without crack are presented. After comparing the upper subplot in Figure 4.29 with Figure 4.20, can be said, that at measuring section of axle, 2xRev harmonic component of vibration signal encountered reduction in comparison with wheel OOR profile (more
than 3 times reduction in amplitude), while 1xRev and 3xRev components exhibit only minor reduction. Decrease in 2XRev harmonic component can be coming from the fact that wheel irregularity wavelengths that were comprising OOR profiles were generated with randomly assigned phase angle, thus there is possibility that 2xRev harmonic of wheel OOR on one side of axle was in counter phase with corresponding harmonic on the opposite side, thus causing reduction in their resultant, mutual amplitude.

Situation has changed over after introducing crack into axle. Due to very low initial vibrations of axle caused just by wheel OOR excitations, after introduction 35% deep crack, dynamical behaviour of axle became completely dominated by crack breathing phenomena. Vibration spectra components measured close to cracked section were showing much increased amplitudes, thus representing closely results obtained from simulation not including any external excitations (Figure 4.28). Comparison of amplitudes of first three harmonic components for non-cracked and cracked axle under influence of ‘new’ wheel OOR profile is presented in Figure 4.30.

![Figure 4.30 Comparison of harmonic components amplitudes ('new' wheel OOR)](image)

where 1x, 2x, 3x corresponds to non-cracked axle, and 1xCR, 2xCR, 3xCR to cracked (35%) axle vibration signal harmonic components

By analysing the results of this simulation scenario it is possible to conclude that for wheel irregularity profile corresponding to new or reprofiled wheel, the influence of external source excitations on axle dynamical behaviour is minor. In that case a 35% deep axle crack could be detected by measuring and trending axle deflections. Due to the fact that axle dynamic behaviour was almost completely dominated by crack influence, by comparing typical ‘new’ wheel OOR can be assumed that cracks having depth of at least 15% could be fruitfully detected by means of LFV in case the wheel is new or newly reprofiled.

Another simulation scenario was taking into account influence of wheel irregularity profile representing worn railway wheel. In this case excitation harmonics were higher, according to estimations described in details in section 4.4.2.2. Similarly to previous simulation, different irregularity profile was used for left and right side of axle respectively. Results of this simulation are presented in Figure 4.31.
As can be seen in Figure 4.31 (upper subplot) dynamic behaviour of axle in which crack is not present is completely affected by external source excitation profile, which in this case is representing wheel out of roundness irregularities in correspondence to worn railway wheel. Vibration signal spectra obtained in this case are closely representing first harmonic components of worn wheel irregularity profile (Figure 4.22). Some reduction in 3xRev harmonic can be observed, but it can be caused due to opposite phases of irregularity at this range of frequency for left and right side of axle. In comparison with case of simulation scenario which was excluding external source excitations (Figure 4.28) only minor changes are taking place when external excitation is introduced. By comparing both plots in Figure 4.31, for non-cracked and cracked axle respectively, high increase in
amplitude can be observed at 1xRev and 2xRev harmonic. Specifically 2xRev harmonic is showing high sensitivity for fault occurrence, by increasing its amplitude more than 5 times. In spite of still visible slight reduction of vibration components amplitudes in comparison with case without additional excitation, still the ratio of their change after introducing the crack gives the possibility for fault detection by means of low frequency vibration analysis. Change in vibration spectra components can be caused by increased bow of axle due to crack occurrence (1xRev) as well as reduced local stiffness of axle cracked cross section resulting in additional deflections enhanced by external excitation signal (2xRev). Different behaviour can be observed in case of 3xRev harmonic, which is undergoing reduction of its amplitude. It could be explained by the fact that 3xRev harmonic component is mostly representing crack breathing phenomena and due to external excitation influence crack for a longer time stays in opening condition compared with situation where no external excitations were defined (this would also justify increase of 1x and 2xRev components). Comparison of first three harmonic components of vibration signal obtained from simulation case including ‘worn’ wheel OOR profile is presented in Figure 4.32.

![Figure 4.32 Comparison of harmonic components amplitudes ('worn' wheel OOR profile)](image)

where 1x, 2x, 3x corresponds to non-cracked axle, and 1xCR, 2xCR, 3xCR to cracked (35%) axle vibration signal harmonic components.

Closer to real life railway vehicle operating conditions are presented in following set of simulations. In those simulation scenarios rail related excitations as well as wheel related excitations were taken into account. Rail related irregularities were created assuming long wavelets, acting on the distance of approximately 100 [m], thus allowing around 33 rotations of axle under influence of this type of external excitation source (details given in section 4.4.2.2 and 4.4.2.3). As already mentioned, different rail irregularity profile related excitations were used for left and right side of axle correspondingly. Wheel OOR profiles were used without any changes from previously described scenarios, namely ‘new’ wheel OOR and ‘worn’ wheel OOR. No changes were also made to axle model, and its finite element grid.
First simulation in this series was comprised of axle model without crack and excited both by 'new' wheel OOR as well as rail irregularities. After obtaining axle vibration signal, a filter was applied in order to minimise the influence of low frequency component introduced by slowly changing track irregularity (below 1xRev harmonic component). For this reason IIR mode, bandpass type, sixth order Inverted Chebyshev filter was used with 5.5 [Hz] low cutoff frequency. Results of simulation including filtered signal are presented in Figure 4.33.

Figure 4.33 Axle vertical vibration from 'Mixed case' simulation results (under 'new' wheel OOR and rail profile influence)
upper: time domain axle vibration signal (blue: original, red: filtered)
lower: filtered vibration signal harmonic components (FFT)
As could be expected dominating 1xRev harmonic component of vibration signal can be observed in simulation result presented in Figure 4.33. 2xRev and 3xRev components remained at very low level; similar to one resulted from simulation without rail irregularity (upper plot in Figure 4.29).

After introducing 35% deep crack into model vibration response of axle has become more intense. Result of this simulation is presented in Figure 4.34. Also in this case, red line on upper plot is representing filtered vibration signal in time domain, by using same filter characteristic as in abovementioned simulation.

![Axle vertical vibration from 'Mixed case' simulation results](image)

**Figure 4.34** Axle vertical vibration from 'Mixed case' simulation results (under 'new' OOR profile, rail profile and 35% crack depth influence)

upper: time domain axle vibration signal (blue: original, red: filtered)
lower: filtered vibration signal harmonic components (FFT)
As can be clearly seen from simulation result (Figure 4.34) 35% deep crack introduced in model affected its dynamical behaviour in sufficient way to cause increase in all first three harmonic components. Especially important from the point of view of monitoring axles structural integrity 2xRev harmonic [49] has increased significantly, resulting in amplitude level measured during simulation of cracked axle without external excitations (Figure 4.28).

This allows to assume that providing proper measurement system and conducting trending of vibration signal spectra components in this case detectability of assumed 35% crack is achievable. Ensuring proper filtering mechanism in frequency range of crack breathing mechanism it may be possible to detect crack with smaller size, although big enough to overcome 2xRev component of wheel/rail contact excitations. According to simulations results, already 15% crack should be detectable by means of LFV.

In Figure 4.35 comparison of amplitudes change of first tree harmonics of vibration signal are presented, which resulted from simulation of non-cracked and cracked (35%) axle undergoing external source excitations representing ‘new’ wheel OOR and rail irregularities.

![Comparison of harmonic components amplitudes](image.png)

**Figure 4.35 Comparison of harmonic components amplitudes ('new' wheel OOR and rail irregularities included)**

where 1x, 2x, 3x corresponds to non-cracked axle, and 1xCR, 2xCR, 3xCR to cracked (35%) axle vibration signal harmonic components

The last two simulated cases were different from previous ones in terms of wheel irregularity profile. For following scenarios ‘worn’ wheel irregularity profile was used, thus generating significantly higher excitation amplitudes in combination with rail irregularities. Both of mentioned signals were provided as excitation into the model. Two cases were simulated, one without crack defined in the axle (as reference situation) and one with 35% axle cross section cracked.

Simulation results of non-cracked axle influenced by excitations coming from ‘worn’ wheel OOR and rail irregularities are presented in Figure 4.36.
As simulation results shows (Figure 4.36.,) in comparison with results obtained from equivalent simulation excluding rail related excitations (Figure 4.31 upper) no significant change of vibration harmonics can be observed as influence of rail irregularities. This result will as usual serve as reference for simulation of axle with crack introduced.

The last, most severe from vibration point of view, and most demanding from calculation point of view was the simulation of cracked axle under influence of vibration
excitations coming from ‘worn’ wheel OOR profile as well as rail irregularity. As in all of previously described simulation scenarios which were including external source excitations, also in this one different irregularity profiles were used for left and right side of axle. Among all simulations conducted in described set, this one is reflecting the real operating conditions of railway vehicle in closest way. Simulation result coming from this case is presented in Figure 4.37.

Figure 4.37 Axle vertical vibration from 'Mixed case' simulation results (under ‘worn' OOR profile, rail profile and 35% crack depth influence)
upper: time domain axle vibration signal (blue: original, red: filtered)
lower: filtered vibration signal harmonic components (FFT)
As can be observed from Figure 4.37 introducing rail irregularity profile combined with excitations representing worn railway wheel didn’t overcome specific type of behaviour caused by breathing mechanism of developed crack. Also in this case 2xRev harmonic is undergoing very significant amplitude change (easily seen when compared with Figure 4.36), which confirms its feasibility in structural health monitoring of railway axle. To enable quick visual comparison, in Figure 4.38 first three vibration signal harmonics coming from this simulation case are presented. The biggest ratio of change is experiencing 2xRev harmonic, resulting in 5 times increased amplitude. Also 1xRev and 3xRev harmonic are showing increasing trend, which for the latter one would mean that effect of crack breathing mechanism was enhanced by rail irregularity excitations.

The observed sensitivity of 3xRev component (which is correlated specifically to crack breathing phenomena [49]) to rail irregularities, thus its increasing amplitude can be meaningful for the change of crack growth rate by enabling higher amplitudes of crack opening and closing thus reducing fatigue life of component.

The sensitivity of 2xRev component for fault existence was also proved by conducting additional simulation, in which crack of 30% depth was introduced in axle. Simulation conditions were kept unchanged with respect to the ‘mixed case’ for which 35% crack depth was simulated combined with influence of worn wheel OOR and rail irregularity. The simulation result and rate of change of vibration spectra amplitudes for ‘no crack’, 30% and 35% crack can be seen in Figure 4.39 and Figure 4.40 accordingly.

![Comparison of harmonic components amplitudes](image)

**Figure 4.38 Comparison of harmonic components amplitudes (‘worn’ wheel OOR and rail irregularities included)**

where 1x, 2x, 3x corresponds to non-cracked axle, and 1xCR, 2xCR, 3xCR to cracked (35%) axle vibration signal harmonic components.
Figure 4.39 FFT of axle vertical vibration from ‘Mixed case’ simulation results including ‘worn’ OOR profile, rail irregularity profile and 30% deep crack influence

Comparing the rate of change of vibration components amplitudes in Figure 4.40 it can be clearly seen that the harmonic components most sensitive to increase of crack size are 1xRev and 2xRev. Due to the fact that simulation case from which abovementioned results are taken was representing the closest to reality operating conditions (worn steel railway wheel-related irregularities and rail-related irregularities) it is possible to assume that these
vibration components are the most suitable to enable proper estimation of structural integrity of railway axle.

Simulations including additional sources of excitation coming from wheel-rail interaction showed that irregularity levels reported on worn railway wheels are not overcoming vibrations caused by 35% deep crack in the axle, thus allowing to assume that the detection of a crack in similar size is highly probable during vehicle service. 1xRev and 2xRev harmonics occurred to be the most suitable to indicate presence of crack. Comparison of two cases with different crack depth (30% and 35%) revealed that NxRev vibration components are undergoing higher rate of change of amplitude due to crack propagation than from developing OOR profile of wheel allowing to assume that that axle condition monitoring based on LFV is capable of delivering diagnostic results even during further degradation of wheel profile.
CONCLUSIONS

The work presented in this thesis was intended to cover the existing open points in structural health monitoring of railway axles. Despite the high level of safety and reliability ensured by the use of many advanced methods in the production and maintenance of these components, the lack of system capable of monitoring the condition of an axle during its operation, thus providing its actual health parameters still exists. The need for such a system is justified by many factors. Just mentioning few of them, like working conditions, where due to the fact that axle is operating in open space area is prone to destructive effects of corrosion and track ballast impacts; demand for increasing loads that railway vehicles can carry as well as speed at which they operate and, last but not least, foreseen growth of railway transport by 2020 and correlated society and operators expectations for increased safety and reliability levels of railway stock.

Due to the fact that railway axle is a safety critical component of the railway vehicle, which as a part of wheelset provides guidance of vehicle, transfer loads, traction as well as braking forces, the importance of monitoring its structural integrity during operation process becomes obvious.

The proposed method for online health monitoring of railway axle is based on axle vibration measurements, specifically harmonic components in the axle bending vibration having periodicity which is an integer sub-multiple of the axle revolution period. These specific types of vibrations are induced by the so called crack-breathing mechanism and by asymmetry in the bending stress of the axle, as produced by a propagating crack.

Due to the fact that railway axles are operating at relatively low speeds, vibrations correlated to axle angular speed are concentrated in the low frequency range. The advantage in that case is in the possibility to measure low-frequency vibration by using simple, robust and inexpensive transducers.

Methods of estimating condition of rotating machines are widely introduced and tested in the field of rotordynamics, from where also this approach is taking its root. Abovementioned method of measuring vibration signal low frequency components was initially proposed for crack detection in the shaft of turbo-machinery [49] where it proved its accuracy and reliability, however due to different work specification of railway axle (speeds much lower than first critical, not stable working conditions due to additional excitation sources, weather condition influence, etc.) successful application in railway field was not obvious. For this reason feasibility of this approach in railway application was initially checked by means of experiments performed on a laboratory test rig [43] and demonstrated to be a promising method for railway axles crack monitoring application.

For further, more detailed, investigation bending vibrations measurements of full size railway axles were conducted by means of test bench located in Politecnico di Milano laboratories, which by applying load and speed equal to real railway vehicle operational values allowed to investigate both the axle resistance to crack occurrence as well as vibrations caused by its eventual development.

Results of laboratory tests presented in Section 3.5 proved that Low Frequency Vibration analysis in laboratory conditions is capable of detecting changes of bending vibration signal due to occurrence and propagation of crack in railway axle. During these experiments a number of NxRev harmonics appeared in the measured signal and after performing trending process the increase in the level of vibration became evident for some
of them. As the result of test, it was possible to observe that the most sensitive for crack occurrence are the 1xRev to 3xRev harmonics. The 1xRev component showed the largest increase in amplitude at the final stage of the tests along with the 2xRev component which proved to be also sensitive to the number of cumulated loading blocks. The 3xRev harmonic component also experienced almost monotonic increase but its trend was less clear, probably on account of disturbances such as thermal effects that may produce a bow of the rotating axle and thus reduce the breathing mechanism by keeping the crack partially opened during revolutions.

Due to the fact that finally the 1xRev and 2xRev components allowed to assume that they are the most promising indicators of eventual fault development in the axle, their amplitudes measured at axles which experienced crack propagation were plotted at Figure 3.13 and Figure 3.14 for 1xRev and 2xRev harmonic respectively.

After obtaining promising results from laboratory tests numerical, 3D finite element model of axle was created to enable investigation of influence of crack depth to dynamics of railway axle under different working conditions.

Simulation campaign was divided into two parts, first one covering conditions found in laboratory tests, and second one investigating possible vibrations levels that can occur during train operation (thus taking into account additional sources of disturbance).

Due to the fact that railway wheels and rails are undergoing wear effects, thus introducing additional excitations to the axle coming from the wheel-rail contact, the influence of these excitations on crack breathing phenomena was under biggest concern.

Results of simulations representing laboratory conditions appeared to be in a good qualitative agreement with measurements performed in the full – scale tests on cracked axles. Specifically 1st and 2nd harmonic showed highest sensitivity for crack presence and its size, which is also consistent with laboratory measurements results. The amplitude levels obtained by means of Finite Element model for a crack size in a range of 25% to 35% of axle cross section were consistent with vibration amplitudes recorded during final stage of laboratory tests, when crack was undergoing high propagation ratio.

So called ‘track’ case simulations including additional sources of excitation coming from wheel-rail interaction revealed that even for irregularity levels reported on worn railway wheels (dominating 1st and 3rd harmonic) still vibrations caused by 35% deep crack were visible, thus allowing to assume that the detection of a crack having a size in thus order of magnitude is highly probable despite the disturbances acting on the wheel during real service. 1xRev and 2xRev harmonics occurred to be the most suitable to indicate presence of crack. This is also true specifically when further degradation of wheel circumference is considered, due to the fact that as simulations and field measurements are showing [103] during increasing wear of wheel, OOR harmonics tends to develop into higher orders leaving 2xRev harmonic slightly affected or causing reduction in its amplitude.

Finally, nevertheless of possible influence of wheel OOR irregularity on NxRev vibration component, the latter ones are undergoing higher rate of change of amplitude due to crack propagation and thus affecting vibration signal to a larger extent than the introduced by more slowly developing OOR thus opening the way to properly assess the structural integrity of the axle.

Observing the actual developments in railway auxiliary systems, the energy harvesting method from rotating railway axles seems to serve a good platform for implementation of
LFV measurement system. Due to the fact that equipment for both these systems will be located at the axle side bearings reduction in additional space requirements, as well as implementation costs can be achieved.

LFV application for monitoring structural integrity of railway axles can possibly be extended to monitor wheel OOR profiles and track quality (latter one when LFV is combined with GPS interface) due to their mutual influence on axle vibration levels.
ACKNOWLEDGEMENTS

The SHM results shown in the present research were obtained in the frame of the SUSTRAIL Collaborative European Project (Grant Agreement #265740). The full-scale crack propagation test is carried out in the frame of the MARAXIL Project co-funded by Regione Lombardia (ID 16973, Rif. n° MAN-15).

The author would like to thank Prof. Stefano Bruni, Prof. Stefano Beretta, Dr Michele Carboni and Ing. Mauro De Mori from Mechanical Engineering Department at Politecnico di Milano for the help and useful discussions.
BIBLIOGRAPHY:


[34] "Final Report Summary " WOLAXIM (Whole Life Rail Axle Assessment and Improvement).


