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Railway applications — Axle design method

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National foreword

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European foreword

This document (CEN/TR 17469:2020) has been prepared by Technical Committee CEN/TC 256 “Railway applications”, the secretariat of which is held by DIN.

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Introduction

The first railway accident due to the fatigue failure of an axle occurred on 1842, May 8th, in France, near Meudon, on the Versailles-Paris line.

In those days, the fatigue phenomenon was unknown. This failure initiated numerous studies including German Engineer August WOHLER works on wheelset failures at the end of XIX century.

In the middle of XX century, M. KAMMERER, an engineer working for French railways, established the bases for the calculation of wheelset axles.

At international level, the report ORE B136 RP11 « Calculation of fret wagon and passenger coaches' wheelset axles » was edited in April 1979, using in particular the French approach.

This document allowed editing on 1994, July 1st of UIC leaflet 515-3 «Railway rolling stock – Bogie – Running gears – Axle calculation method».

The first edition of the European Standards about design of axles occurs on April 2001 (EN 13103 for non-powered axles for powered axles).

The ongoing European standardization has allowed the merging of EN 13103 in only one standard (EN 13103-1 Railway applications – Wheelsets and bogies – Part 1: Design method for axles with external journals) and the creation of a new Technical Specification about internal journal (CEN/TS 13103-2 Railway applications – Wheelsets and bogies – Part 2: Design method for axles with internal journals).

All these documents, including M. KAMMERER's work up to EN 13103-1 and CEN/TS 13103-2, use the beam theory calculation method. The stresses taken into account are then the nominal stresses. The fatigue limit is determined from full scale tests in which nominal stresses are taken into account. Concentration factors are defined from tests to consider the local geometry and to increase the nominal stress locally. The method is quite simple, with no need of sophisticated calculations or dedicated software.

On another hand, in the middle of XX century, the need in mechanics to have a tool to calculate complicated parts lead to the development of the finite element method.

Along with the theoretical study of this method, the use of new mathematical objects and the growth of calculation capacities of computers, the finite element method raised to a large and common use in design.

The stresses then calculated are local stresses, and not anymore nominal stresses, and the fatigue limit to be applied with this methodology are based on local stresses.

In the Euraxles project, the objective was to propose the use of a new assessment method based on load measurements, finite element method, experimental fatigue limit and new safety concept for the design of axles in particular for axle designs requiring more complex geometries. This design procedure is different from today's proven methods given by the EN standards and not in a status to substitute them. Nevertheless, it was considered interesting to gather the Euraxles project results inside this Technical Report. The content should be considered as partial and only for informative uses at this stage. For example, the reliability of the input data, the variability of parameters, boundary conditions and the confidence in the partial results should be assessed at full extent.

Where relevant, CEN TC256/SC2/WG11 comments and responses to preliminary enquiry inside the community were inserted for additional use for the reader, as Observations of CEN/TC 256/SC 2/WG 11. Besides, a general recommendation of use has been drafted by WG11 members in chapter 9.

This new method is described in this Technical Report in order to allow the possibility for wheelset designers to apply it and to collect return of experience for further improvement.

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Work Program summary

Clause 4 deals with the definition of a new fatigue design method which enables to assess the in-service reliability of axles with regards to fatigue failure. The proposed approach, based on the "Stress Strength Interference Analysis" (SSIA) and the "Fatigue-Equivalent-Load" (FEL) methods, aims at estimating the probability of axles' fatigue failure by characterizing the variability of in-service loads and the scatter of the axles fatigue strength.

First of all, the main lines of the SSIA method are recalled. This method aims at evaluating the in-service reliability of components for their design or their homologation. In the second part, the fatigue load analysis method that is proposed for railway axles is described. It starts with a post-processing of an axle load measurement: from a time signal of forces applied to both wheels fitted on the axle, fatigue cycles of bending moment applied to the axle are identified and transformed into a cyclic equivalent load, Meq, which is a measurement of the severity of the initial variable load. Then, virtual but realistic load spectra are generated, thanks to a classification operation followed by a random draw of elementary load data that considers the operation and maintenance conditions of the axle. All the spectra are then analysed with the FEL method in order to build the distribution of in-service load severities. This distribution gives a picture of the stress to which the axles are submitted. In the third and last part, the methods are applied to real data of SNCF. Sensitivity analyses are performed in order to quantify the effect on Meq of variations of parameters and to verify the convergence and robustness of the process. Finally, results obtained for a passenger coach are given. The comparison between the distribution of load severities and the normative load, defined as according to standards EN 13103-1, shows that, for the studied axle, the normative load is very conservative. Finally, using the axles fatigue limits identified on full-scale tests, a Stress Strength Interference Analysis is performed to calculate the probability of failure of the axle.

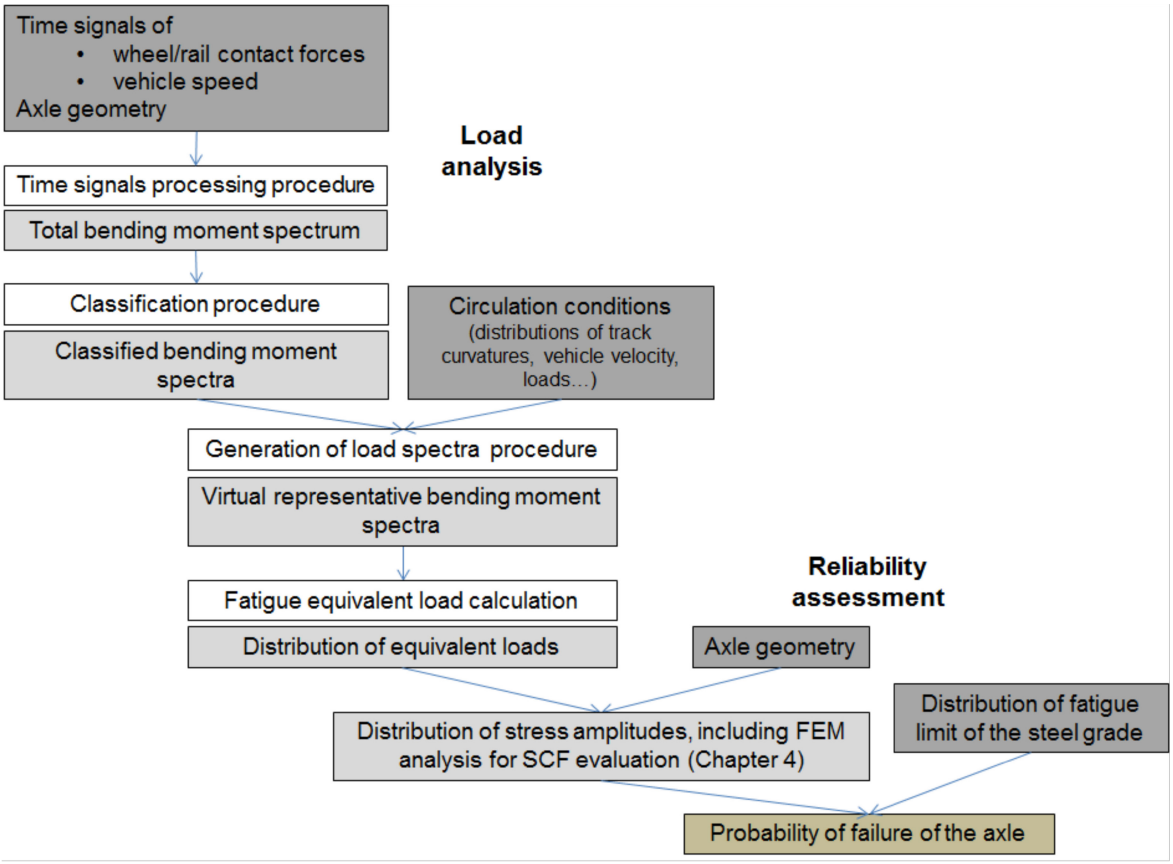


Figure 1 — Flowchart for load analysis and reliability assessment

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Clause 5 concerns the mechanical modelling of an axle and defines a procedure to obtain local stresses from the applied loads.

The characteristics of the finite element models to be applied to railway axles are analysed in terms of element definition, convergence analysis, boundary conditions. A parametric analysis was performed to assess the applicability of the models. The numerical models generated were validated through the comparison with experimental results coming from full scale fatigue tests. Finally, a methodology to design axles using modelling tools as a complement to current European norms is proposed looking for a compromise between the computational effort and the results obtained.

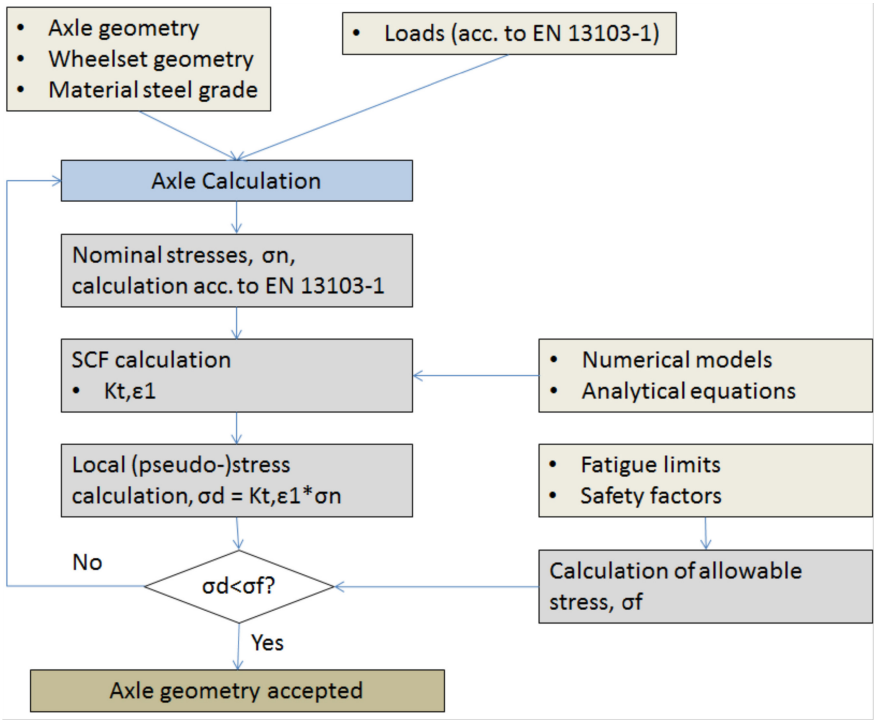


Figure 2 — Flowchart for modelling

The main scope of Clause 6 is to provide the fatigue limits for standard steel grades considering also the effect of surface conditions that may be different from the normal newly machined axles, like surface corrosion that can appear during the service or surface blasting as a method to improve paint adhesion.

The areas of the axles considered were the free body transitions or groves and the wheel seats where at high bending rates relative micro slips take place generating the so called fretting fatigue phenomena.

The paper provides in the conclusions a comparison with the fatigue limits that are today included in the European Standards.

Another aspect that is treated in this work is the stress concentration effect that takes place along the transitions where the body fatigue limit is verified. These parameters were measured by strain gauges during each test and used inside the Euraxle project to validate their estimation through FE model calculation.

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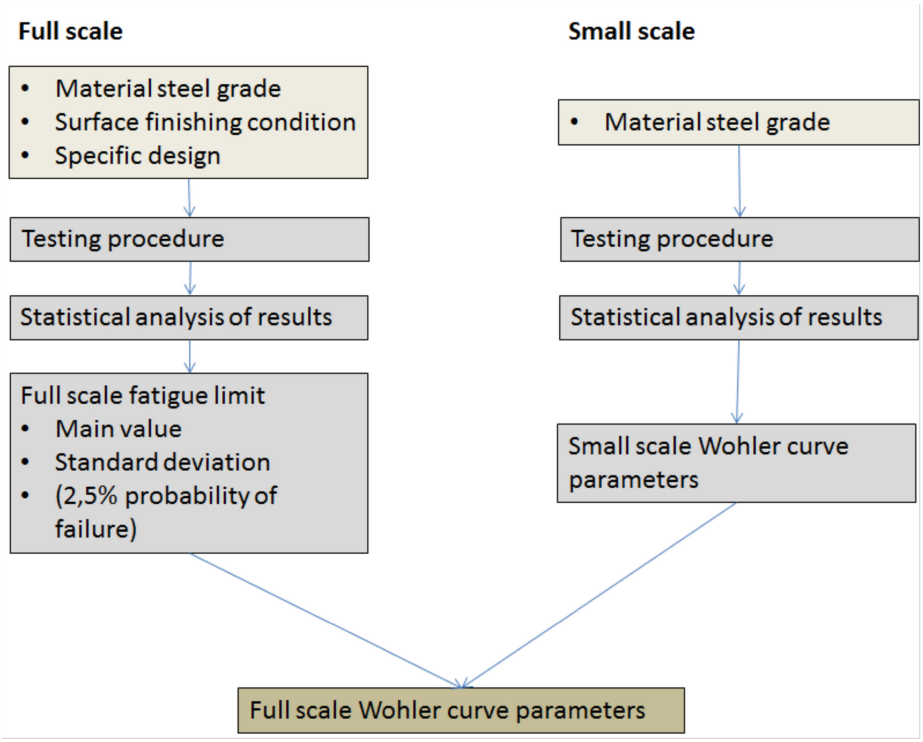


Figure 3 — Full-scale and small-scale fatigue tests

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1 Scope

This document presents the stage of knowledge resulting from the Euraxles project about the design of the axle, and further steps to be taken.

It is the support:

- to define the loads to be taken into account;

- to describe the stress calculation method using finite elements and the validation processes associated;

- to specify the maximum permissible stresses to be assumed in calculations and the safety factors to be used.

This technical report is applicable for:

- wheelset Axles defined in EN 13261 as “pure wheelset”;

- other axle designs such as those encountered in particular rolling stocks e.g. with independent wheels, variable gauges, urban rail…

This document has not for aim to replace EN 13103-1 and CEN/TS 13103-2 but to present a complementary method to the existing ones.

2 Normative references

There are no normative references in this document.

3 Terms, definitions, symbols and abbreviations

3.1 Terms and definitions

No terms and definitions are listed in this document.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— IEC Electropedia: available at http://www.electropedia.org/

— ISO Online browsing platform: available at https://www.iso.org/obp/ui

3.2 Symbols and abbreviations

FEL	fatigue equivalent load method
SSIA	Stress Strength Interference Analysis
KMR	Consequent Miner Rule
FEM	Finite Element Method

Nomenclature is given in Table 1.

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Table 1 — Nomenclature

$Y_1(t)$, $Y_2(t)$	lateral force applied to both wheels
$Q_1(t)$, $Q_2(t)$	vertical force applied to both wheels
P1, P2	Vertical loads applied on the journals
C_{min}	Minimum transition length
F	Fatigue load
F_{eq}	Fatigue equivalent load
F_n	Severe representative load that can be defined for test and simulation validation
x, y, z	Longitudinal, axial, vertical direction of wheelset reference axis
$M_x(y)$	Bending moment applied to the y-section of the axle in the x direction (train circulation direction)
M_x	Bending moment applied to the most critical section of the axle in the x direction
$M_{x,eq}$	Equivalent bending moment applied to the most critical section of the axle in the x direction
$M_{x,EN}$	Normative bending moment
MR	Resultant bending moment
P	Probability
P_f	Probability of failure
P_n	Probability of having a more severe load than F_n
E	Young modulus
K	Stress correction factor
K_t	Stress concentration factor
K_f	Fatigue stress concentration factor
$K_{t,s}$	Stress concentration factor based on strain measurements
D	Total fatigue damage
d_i	Partial damage generated by the ith class of a load spectrum
n_i	Occurrence of the ith class of a load spectrum
N_i	Number of cycles for a crack initiation for the ith class of a load spectrum
CV_X	Coefficient of variation for the X variable
m	Slope of the S-N diagram when using a one single slope curve
k	Slope of the S-N diagram for $S > S_D$
k’	Slope of the S-N diagram for $S < S_D$
N_D	number of cycle for the knee of the S-N diagram
S	Stress load
S_D	stress amplitude for the knee of the S-N diagram
K_{ref}	total mileage of an axle

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σ_f	Allowable fatigue stress
σ_d	Calculated dynamic stress
σ_{VM}	Von Mises stress
σ_h	Hydrostatic stress
τ	Shear stress
σ_1	Principal stress
ε_1	Principal strain
σ_n	Nominal stress
D/d	Diameter ratio (diameter of wheel seat divided diameter of nearby body)
D_N	Outer diameter of hub
d	Diameter of the axle shaft
r_{max}	Maximum value of the radius of the transition r
F1	Full scale axle body fatigue limit
F4	Full scale axle seat fatigue limit

4 Loads

4.1 Reliability analysis based on the Stress Strength Interference Analysis method

Fatigue is known to be a damage phenomenon which is very dispersive. The sources of variability are linked to the material properties that depend on its composition but also on the manufacturing process, the geometry of the structure, loads, usages, environment, etc. To ensure safety, margins applied to the specified loads and the prescribed fatigue limits, associated to a stress calculation method were defined in standards EN 13103-1 [21] for the design and validation of railway axles. They were established in the past decades, based on experience of railway experts and experimental and modelling works. Today, they enable to guaranty a high level of safety for the European railway sector, as feedback from operation shows. But, to gain competitiveness, it can be very useful to measure the available margins in order to ensure that when a new design or a new technology is introduced, the level of safety is maintained.

For that reason, it would be beneficial to switch little by little from conservative approaches towards reliability approaches. Maximalist approaches ensure safe designs by defining safety factors that make the load specifications more severe and underestimate the allowable fatigue limits. The consequence is that optimized solutions can't be found. Moreover, when a significant change occurs in the system, it is difficult to evaluate its impact on reliability. In reliable approaches, the aim is to have a "just necessary" design associated to a target probability of failure. For safety critical components, the probability of failure during the lifetime generally vary from 10^{-5} to 10^{-8}. In the example given in [4] on an automotive engine part, the target probability is 10-6. For railway safety applications, if one considers that the number of accidents due to mechanical failures is rather small, a target between 10^{-6} and 10^{-7} sounds reasonable.

Observations of CEN/TC 256/SC 2/WG 11: The target value quoted (10^{-6} to 10^{-7}) is a failure rate per axle during its whole life. It is approximately in line with the 10^{-9} failure rate per operational hour defined in the CSM (EU regulation 402/2013) for technical systems for which a functional failure with immediate disastrous consequences is assumed.

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Reliability approaches have been developed for many years in various industries. The methods always *consist* in characterizing the variabilities of the system to be designed and in calculating its probability of failure by propagating the uncertainties in a deterministic modelling, as shown in Figure 4.

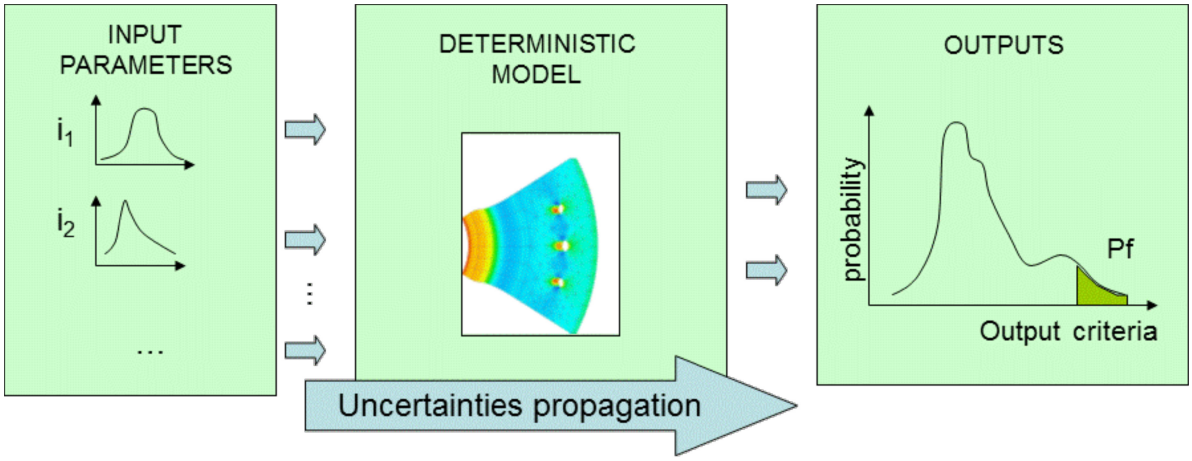
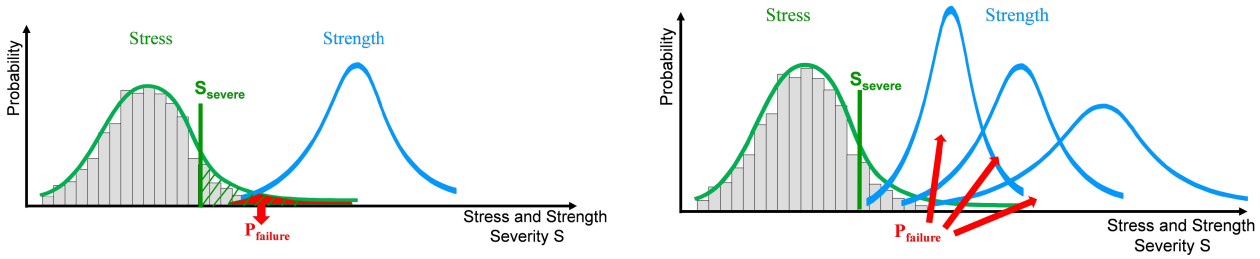


Figure 4 — Reliability approaches

The SSIA (Stress Strength Interference Analysis) is one of the reliability methods. It is widely used in industries, due to its simplicity. Within this approach, only two global parameters are used to characterize the variability of the problem: the load severity - referred to as STRESS - and the component strength - referred to as STRENGTH. These two parameters are chosen so as to be comparable (for instance, local stresses vs fatigue limits). Once the distributions of the Stress and the Strength are identified and modelled, a probability of failure can be easily calculated, by means of analytical expressions or numerical resolutions, as explained for example in [1][2][3][4]. The SSIA is therefore often used in the validation process of a new component. But it is even more powerful in the design process: from a pre-defined target probability of failure Pf, for a given distribution of Stress, the Strength distribution (mean value and scatter) can be specified. Let's notice that there is not a unique solution to this problem, as shown in Figure 5.



a) SSIA for the validation process **b) SSIA for the design process**

Figure 5 — SSIA approach

To apply the SSIA approach, it is necessary to characterize both distributions. The distribution of Strength can be identified thanks to laboratory fatigue tests. To have a good estimation of its distribution and not only its mean value, a sufficient number of tests is necessary. In the Euraxles project, new tests were performed for steel grades EA1N and EA4T. They are described in [17] and in Clause 4 of the report. For the Stress distribution, measurements of loads shall be carried out in various operating conditions in order to capture the variability of the behaviour of the component, as described in 4.2.3. Using complementary information on the variability of usages from an axle to another, a distribution of in-

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service load severity S can be generated. Accordingly, a representative and severe load can be defined as a specification to be used in the numerical and experimental design and validation process. This severe load is defined by its probability of occurrence $P(S > S_{severe}) = P_{severe}$ as shown in 2. This probability is a parameter that needs to be chosen by the industry and depends on the application. The idea is to define a severe load to accelerate the physical fatigue tests, but not to define a solicitation that would induce a damage process that is not the one that is studied. Typical values vary from 1/100 to 1/50000.

4.2 Fatigue load analysis method

4.2.1 General

The method for the axle fatigue load analysis is described in this part. The first step is the load signal post-processing: from a time signal of forces applied to the both wheels, elementary fatigue cycles applied to the axle are identified. Then, by calculating the total damage induced by all these elementary load cycles, a cyclic equivalent load, F_{eq}, which is as damaging as the initial variable load, is calculated. Its amplitude is taken as a measurement of the variable load's severity. The second step of the method is the generation of virtual load spectra which are realistic and representative of real usages of axles. To do so, information from operation and maintenance are used (total average number of kilometres before the axles reach the acceptable wear ratio limit, the way the axles are changed in a whole train during maintenance, the characteristics of the tracks on which the axles circulate, etc.). A strategy to generate those load spectra is described. The third and last step is the processing of all these virtual spectra with the FEL method which enables to build the distribution of in-service loads severities ("Stress" in the SSIA) to be compared to the standard load.

4.2.2 Load signals processing and Fatigue-Equivalent-Load

In mechanical high-cycle fatigue, Wöhler works showed that the main parameters that affect the lifetime of a material under a cyclic one-dimensional load are the amplitude and, to a less extend, the mean value of the load. To estimate the total damage induced on a component by a load with variable amplitude, it is necessary to identify all the elementary cycles included in the load signal and finally cumulate the partial damages induced by all these cycles. When defined by their amplitudes and their mean values, the elementary cycles can be identified thanks to a Rainflow counting method, as explained in [6]. When the mean values variations of the elementary cycles are small, one can consider that only the amplitudes of the cycles have to be considered in the fatigue analysis. Hence, a simple cumulative damage law, as the linear Miner rule [7], can be used to calculate the total damage. Finally, a cyclic equivalent load, called the F_{eq}, producing the same total damage, can be easily calculated, using Wöhler curves. Its amplitude is taken as the parameter of load severity. The whole process is explained in details in [1] and is presented in Figure 6, where F(t) is the initial load time signal and F_{eq} is the amplitude of the final cyclic equivalent load.

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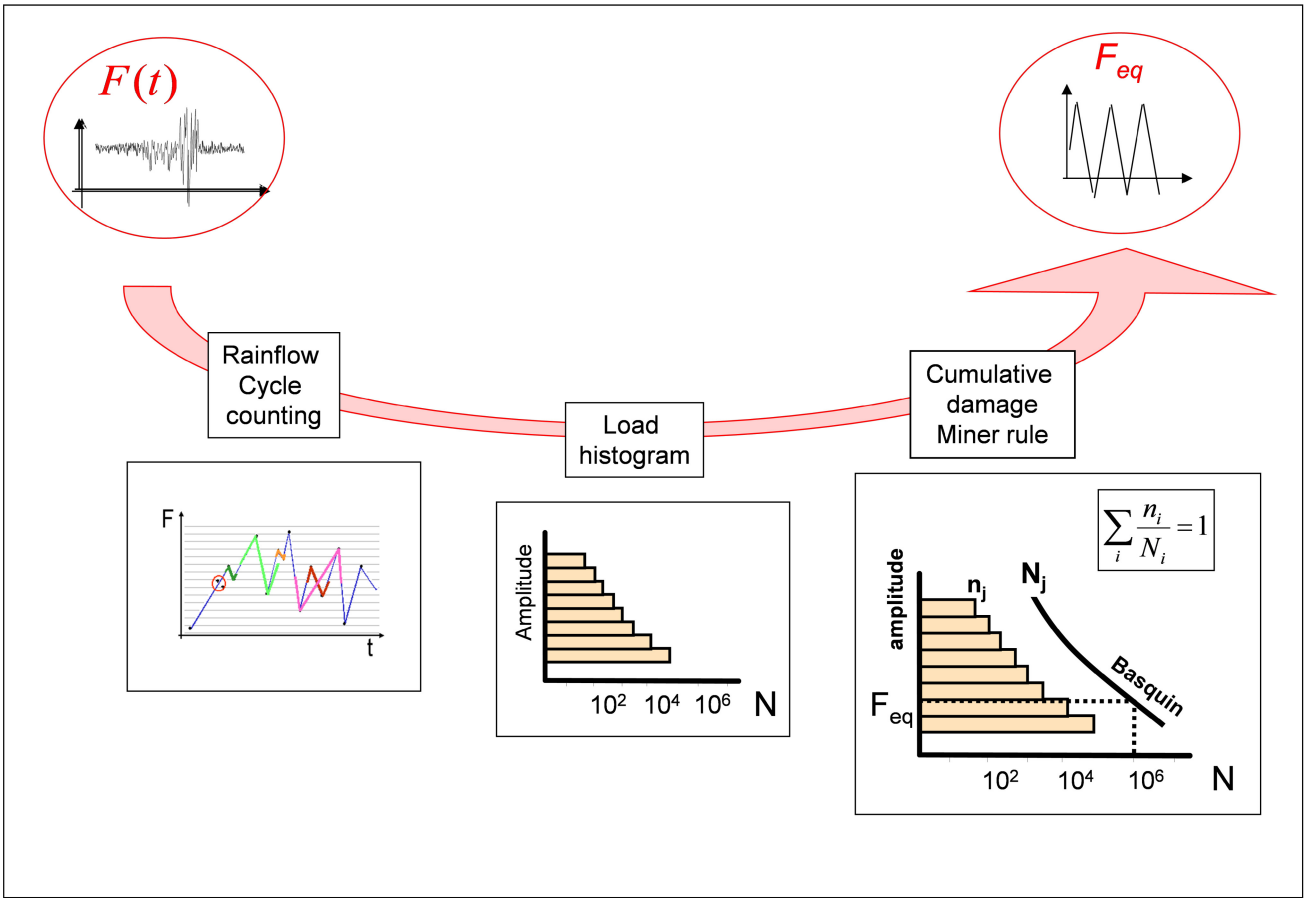


Figure 6 — The FEL method

An adaptation of this FEL approach to the railway axle is proposed and presented in the next paragraphs.

During operation, axles are subjected to vertical loads due to the train mass on both axle journals, the wheel/rail contact forces and the braking load. Let's call Y_1, Q_1, Y_2, Q_2 the lateral and vertical forces applied to both wheels and P1 and P2 the train loads, as shown in Figure 7. These forces generate bending moments in each cross section of the axle, defined by the coordinate *y*. It is assumed that the bending moment in the train circulation direction, Mx(y), sizes the axle to the first order and that the bending moments in the other directions and the other forces (torques) applied to axles can be neglected in the fatigue analysis since their occurrences are low.

Observations of CEN/TC 256/SC 2/WG 11: The assumption that only the bending moments in the train circulation direction are needed to design the axle should be demonstrated for a full extent assessment of an axle. The actual design philosophy of an axle in EN 13103 series standard takes into account all bending moment directions and torsion moments. Thus, in a cumulative damage approach, not only the number of occurrences is to be taken into account but also the inherent level of the damages.

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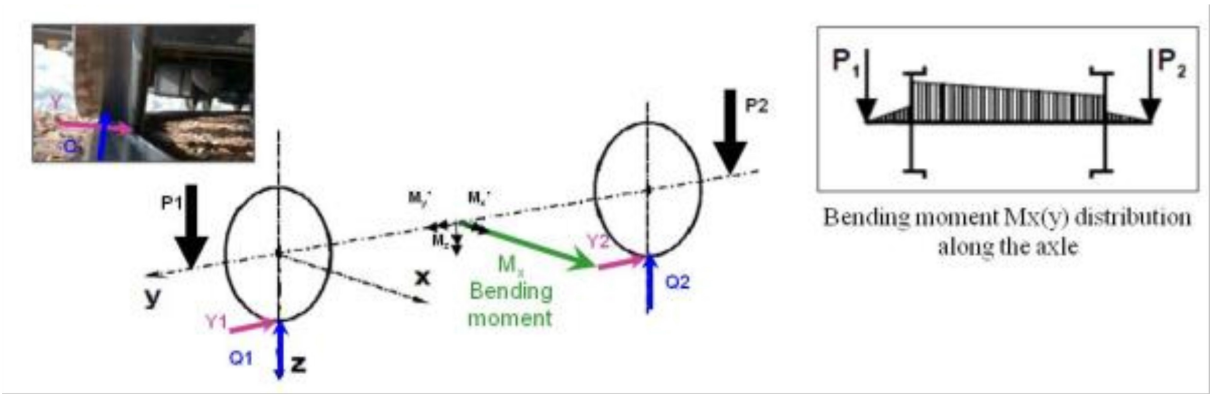


Figure 7 — Forces applied to wheels and axle

When submitted to a rotating bending moment $M_x(y)$, any material point of the axle, defined by a section y and a vertical location z, undergoes a stress cycle $\sigma(y,z,t)$ during each revolution of the axle. Its amplitude $\Delta\sigma(y,z)$ is proportional to M_x and can be calculated following Formula (1):

$$\Delta\sigma(y,z) = 2 \times max_t(\sigma(y,z,t) = \frac{r(y,z)}{I(y)} M_x(y) \quad (1)$$

To these stress variations, mean stresses can be added (mainly due to the press-fitting of wheels and pinions on the axle; residual stresses due to manufacturing are not considered as they are included in the fatigue tests made to determine the fatigue limits of full scale axles) but they can be assumed to be constant and to depend only on the y coordinate.

During load measurements, time signals of lateral and vertical forces Y_1, Q_1, Y_2, Q_2 at the wheel/rail contact applied to both wheels are recorded. To identify all the elementary fatigue cycles included in the time signals $Y_1(t)$, $Q_1(t)$, $Y_2(t)$, $Q_2(t)$, it is therefore necessary to identify all the axle revolutions and to determine the bending moment value M_x, assumed to be constant during an axle revolution. To be conservative, it is suggested to use the maximum value of M_x encountered during each revolution. Finally, the post-processing of a test campaign gives a bending moment spectrum $(M_x(i), n_i)$ where n_i is the occurrence of the $M_x(i)$ cycles for any point of the axle.

To measure the severity of this bending moment spectrum, the FEL method is used. It consists in calculating the cyclic force that would reproduce the same damage as the load spectrum, when repeated millions of times. In the specific case of the railway axle, the amplitude of the stresses cycle at each material point depends on the value of the bending moment M_x, while the mean value mainly depends on the location of the point on the axle and the value of the press fitting of the wheels on the axle. For this reason, only the value of the bending needs to be recorded in the load analysis; there is no need to take into account the mean value of the cycles.

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Generally, to calculate the equivalent bending moment M_{eq}, the traditional and simple linear cumulative damage Miner rule is used: the total damage produced by the whole load spectrum is given by Formula (2):

$$D = \sum_i d_i = \sum_i \frac{n_i}{N_i} \tag{2}$$

where

N_i is the number of cycles at failure for the load $M_x(i)$.

The cyclic load M_{eq} which, repeated N_{eq} times, produces the same damage D is calculated with the same damage law. It is expressed with Formula (3):

$$\forall i \quad N_i.M_i^m = N_{eq}.M_{eq}^m \qquad D = \sum_i \frac{n_i}{N_i} = \frac{1}{N_{eq}} \sum_i \frac{n_i.M_i^m}{M_{eq}^m} \qquad \Rightarrow \qquad M_{eq} = \left(\frac{1}{N_{eq}} \sum_i n_i.M_i^m \right)^{1/m} \tag{3}$$

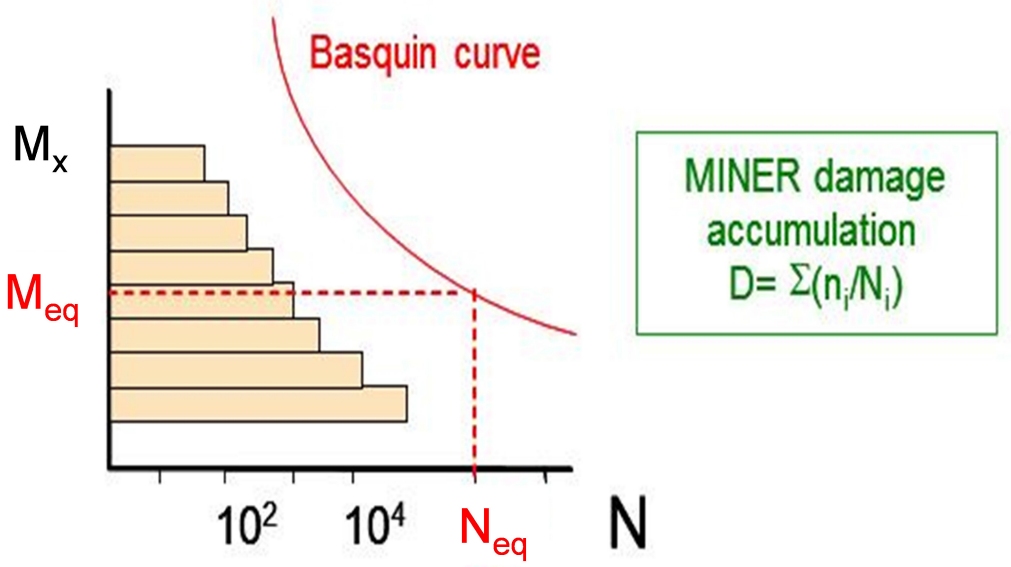


Figure 8 — Calculation of M_{eq} for a bending moment spectrum

In the special case of railway axles, the number of cycles supported by a railway axle is generally more than 10^8 and experimental tests show that the Wöhler curves of the axle steel grades exhibit a change of slope after 10^7 cycles. Hence, instead of using the traditional Basquin law with one slope, a Basquin law with two slopes following the rule proposed by Haibach (k for $N < N_D$ and k' = 2k-1 for $N > N_D$) can be used [8][9].

Another alternative method for the calculation of the cumulated damage generated by a load spectrum is the "consequent miner-rule method" that is explained in FKM guidelines for structural assessment [10] or in [11]. The consequent-Miner-rule assumes that a totally new component has a Wöhler curve with an asymptotic tail S_D, which can be identified by constant amplitude tests. When submitted to a random variable amplitude spectrum, the application of cycles above the fatigue limit progressively increases the damage D and reduces little by little the current endurance limit of the component $S_{D,D\neq0}$. As a consequence, more and more cycles of the spectrum contribute to the damage (even those who were underneath the initial fatigue limit). The process to calculate the total damage and the associated lifetime induced by a load spectrum is represented in Figure 9. The initial load spectrum is in black plain line and

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the initial and current damage laws are represented by the two red curves $S_{D,D=0}$ and $S_{D,D\neq0}$. . The KMR calculation gives the lifetime *N(D)* associated to a damage *D* as the load cycles are progressively imposed.

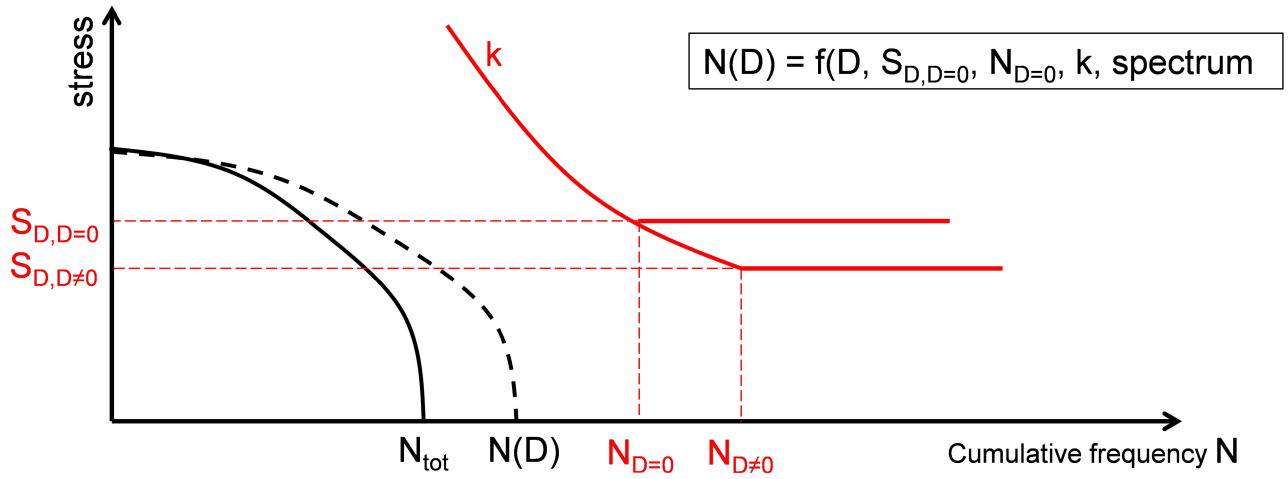


Figure 9 — Consequent Miner Rule lifetime calculation

The formulas to calculate the total number of cycles *N(D)* in terms of the total damage *D* within the KMR are the following:

$$(D) = \left[(A_{kon} - 1).D + 1 \right] . \left(\frac{S_{D,D=0} = 0}{S_{a,1}} \right)^{k} . N_{D,D=0}$$

$$A_{kon} = \left(\frac{S_{a,1}}{S_{D,D=0}} \right)^{k-1} . \left[\frac{Z_1}{N_1} + \sum_{v=m}^{j} \frac{Z_2}{N_2} \right]$$

$$Z_1 = \left(\frac{S_{D,D=0}}{S_{a,1}} \right)^{k-1} - \left(\frac{S_{a,m}}{S_{a,1}} \right)^{k-1} \qquad N_1 = \sum_{i=1}^{m-1} \frac{h_i}{H} . \left(\frac{S_{a,i}}{S_{a,1}} \right)^{k}$$

$$Z_2 = \left(\frac{S_{a,v}}{S_{a,1}} \right)^{k-1} - \left(\frac{S_{a,v+1}}{S_{a,1}} \right)^{k-1} \qquad N_2 = \sum_{i=1}^{v} \frac{h_i}{H} . \left(\frac{S_{a,i}}{S_{a,1}} \right)^{k} \qquad (4)$$

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The parameters in the formulas correspond to:

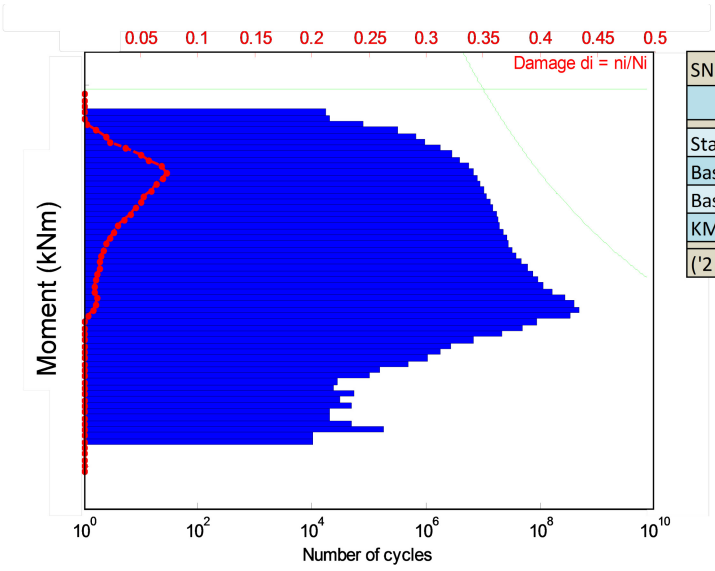
k	slope of the Wöhler-line
$S_{D,D=0}$	initial endurance limit
$N_{D,D=0}$	number of cycles at knee-point
D	miner sum
i	index of step in the spectrum (beginning with 1 for the maximum value)
h_i	number of cycles in step i of the spectrum
x_i	relative load amplitude (related to the maximum load) of step i of the spectrum
$S_{a,i}$	absolute value of the load amplitude of step i of the spectrum
$S_{a,1}$	absolute value of the maximum load amplitude
j	number of steps in the spectrum
m	number of the step smaller or less than the endurance limit
N	permissible number of cycles
Z1, N1, Z2, N_2 and A_{kon}	are for purposes of the calculation
γ	parameter describing the influence of the damage on $S_{D,D>0}$ and $N_{D,D>0}$ set equal k-1 (see [11]).

A new method to define a Fatigue-equivalent load which is consistent with the KMR (Consequent Miner Rule) is necessary. Since only stress ratios are necessary to calculate N, and since the amplitudes of stresses in the axles can be assumed to be proportional to the bending moment, the formulas remain correct when replacing stresses ratios by bending moments ratios. The Consequent Miner damage law applied to bending moments can therefore be directly used for the Fatigue Equivalent Bending Moment calculation: the bending moment corresponding to the lifetime N_D calculated for a damage $D = 1$ is taken as the Fatigue-Equivalent-Bending-Moment.

The methodology described above is applied to an SNCF axle made of steel EA1N and put on a passenger coach running up to V = 160km/h on a French classical line, for which load data are available (111 *km* of load acquisition). This small sample is simply repeated many times to simulate a much higher total mileage K_{ref}.

In Figure 3. 7, the SNCF load spectrum can be seen, with, in red, the value of the partial damage generated by each level of bending calculated with the Basquin law with one slope. It can be seen that with this damage law, the small cycles do not damage the axle. The Fatigue-Equivalent-loads obtained with the Basquin laws with one or two slopes (with $k = 8$; $k' = 15$; change of slope for $N_D = 1{,}00E+07$ cycles) or with the Consequent Miner Rule are given in the table in Figure 10. The total circulated distance (defining the total number of cycles) used to define the fatigue-equivalent-load is $K_{ref} = 7{,}10^6$ km, being the average mileage observed by SNCF for such an axle. Minimum and maximum typical encountered values during operation are $4{,}10^6$ km and 10^7 km. These values are also tested. The normative loads defined as according to standard EN 13103-1 are also given.

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SNCF data analysis : Values of $M_{x,eq}$ in kNm :			
$N_{ref\ (km)}$	1,E+07	6,E+06	4,E+06
Standard EN13103	47,1	47,1	47,1
Basquin 1 slope	41,9	40	37,4
Basquin 2 slopes	35,4	34,6	33,2
KMR	33,4	32,9	32,3
('2 slopes'-KMR)/'2 slopes'	5,6%	4,9%	2,7%

Figure 10 — SNCF data analysis - Fatigue-Equivalent-Bending-Moments calculated with various damage laws and comparison with the standardized load

These calculations show that:

- The standardized load does not vary with the target lifetime of the axle, while the Fatigue Equivalent Load takes into account the effect on the severity of the load of the target lifetime of the axle, which makes physical sense.

- The fatigue equivalent bending moments obtained with the KMR are significantly lower than those obtained with the Miner rule. This is consistent with the initial motivation of the KMR development: the classical Miner rule used with a Basquin law with an asymptotic tail is known to be too severe, especially when most of the loads included in the spectrum are lower than the initial fatigue limit of the component.

- The Fatigue equivalent Loads obtained with the KMR are always significantly lower than the normative loads, which is reassuring! The distance between the normative load and the equivalent-load calculated with KMR seem reasonable. This means that, for this example, the standardized loads are on the safe side. But one shall keep in mind that the load sample which is used is quite short. If the measurements were longer, higher load amplitudes could be encountered and the final equivalent load could be higher. Therefore, the conclusions shall be taken with caution because the studied loads are not various and representative enough.

4.2.3 Method to generate the distribution of in-service load severities

The philosophy of the work that is led is to compare the normative load not to the fatigue-equivalent-load of a “representative load spectrum” but with a distribution of fatigue-equivalent-loads. The idea is to take into account, in the final overall reliability approach, the fact that railway axles have various lives. The loads undergone by an axle that is on a trailer position during its whole life and that circulates on a track with lots of straight lines are definitively different from the loads applied to an axle that is put ahead of a wagon and circulates on a line with many tight curves. Since it is impossible to put sensors on all axles that circulate on the European railway network, it is necessary to develop a method that generates virtual but realistic load spectra.

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Ideally, long measurements should be performed in order to capture all the various situations that can be encountered by the component being designed. But to reduce the duration and the cost of such test campaigns, the French automotive company PSA Peugeot Citroën had defined an experimental strategy. It considered that the sources of variability in loads can be divided into two types:

- the behaviour of the system (the mechanical system itself - car and roads - and the driving style of the driver);
- the uses of the car by its owner (does he often take the highway? does he only take city roads to go to work? does he live in a mountainous area? does he drive alone? does he use his car for vacation with the whole family and luggage in the car boot? etc.).

So the strategy was:

- to carry field measurements with a small sampling of drivers on pre-defined and limited routes, with instrumented cars to capture the "behaviour" of the system "type of road/car/driver";
- to use surveys to obtain information on the uses (ratios in kilometres for each situation of life being defined by the type of road and the transported mass).

Then the combination of both sources of information, using appropriate mathematical methods, led to the generation of virtual loads. Once virtual load spectra were generated, the distribution of their severities could be determined, using the FEL calculation, generally performed with the Basquin law with one slope. From this distribution, a normative load could be defined for the calculation and test validations. Detailed description of the method can be found in [1][2][4]. This approach was used in other industrial domains: aircraft, nuclear, etc. For example, in the project DEFFI led in 2010 by the Cetim, a French research and technical centre for the mechanical industries, with the participation of several companies such as Snecma/Cnes, Mecalac, etc., many applications were studied: elements of the Vulcain engine for Ariane 5 launcher, cable of a ski lift, boom and arm of earth moving machines, bogie of a TGV [3]. Similar work was also led by SNCF on wheels load analysis [14].

It is proposed to develop the same process and adapt it to railway axles. The graphic in Figure 11 summarizes the methodology. The different steps of the method are the following:

Step 1: load measurements are carried out on a sufficiently long distance in order to capture variability due to the tracks characteristics, the train speed and other sources of variability (friction coefficient, position of the axle in the train, etc.)

Step 2: bending moments time signals are calculated, depending on the measurement methods (gauges or accelerometers on wheels or on axles)

Step 3: the time signals are cut and classified into different elementary life situations classes

Step 4: information on operation conditions is gathered (circulated tracks characteristics, train velocity, etc.)

Step 5: virtual load spectra are generated, thanks to the classified load measurements database and operation information

Step 6: the fatigue–equivalent-loads of all virtual load spectra are calculated

Step 7: the distribution of the fatigue-equivalent-loads is assessed and modelled by a statistical distribution

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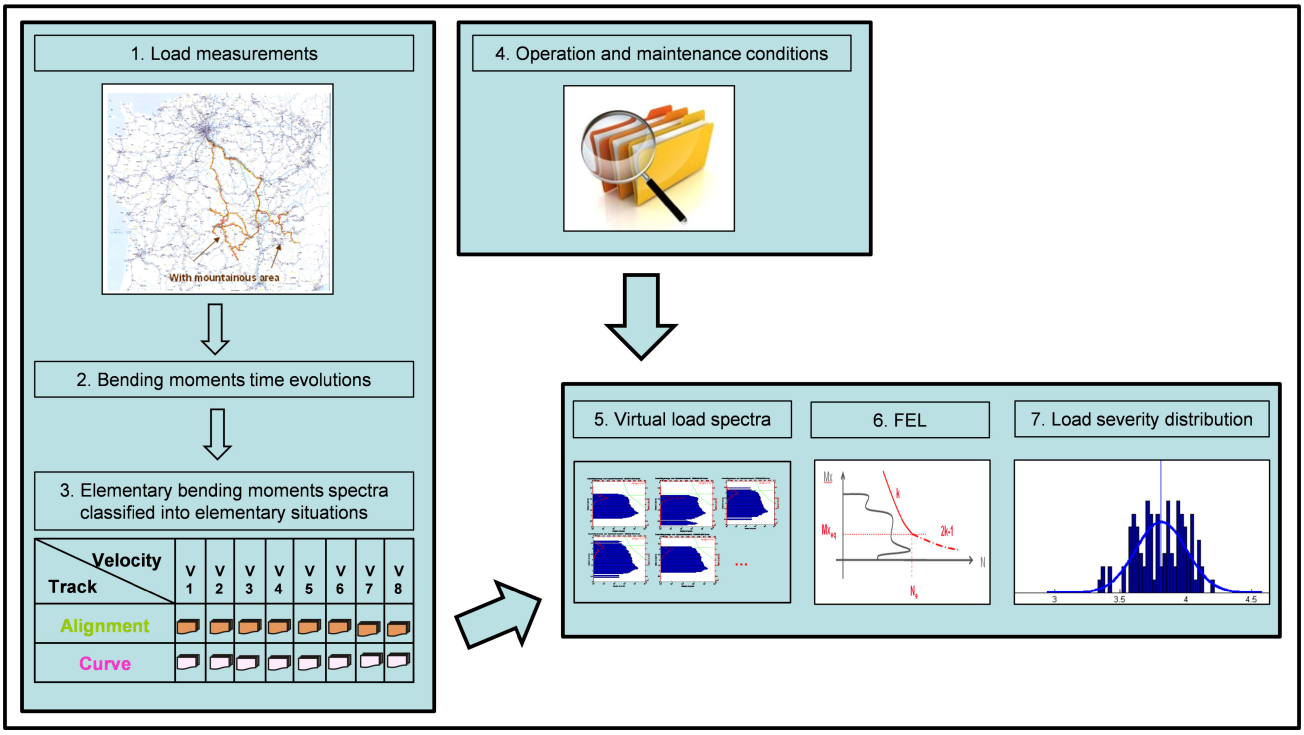


Figure 11 — Generation of virtual load spectra and determination of the load severity distribution process

Steps 1 and 2 were described in the previous chapters. Details on an in-service load measurement study are given in 4.1.

Step 3. The questions raised at this stage are the following: which parameters should be used to define a relevant classification according to fatigue of railway axles? What kind of operation information is needed (and is available) in order to generate virtual uses?

Many parameters that affect the fatigue loads can vary during an axle life: the track geometry (tangent tracks, curvatures, cant deficiencies that depend on the cant and the train speed, the defects of the track – defects of the geometry, defects in rails, hanging sleepers, track stiffness…), the train speed and more generally its dynamical behaviour, the carried mass and its distribution on the different axles of the train, the friction coefficient in the rail/wheel contact area, the profiles of the wheel and the rail, the position of the axle in the train, etc. It is important that the major sources of variability are considered to build the load database. Depending on the easiness of characterizing and measuring them, the strategy will consist in either using them as a parameter for defining elementary situations of life, either assuring that all the ranges of the parameters are met during the tests. Many studies on the dynamical behaviour of the trains have been carried out in the railway sector but one shall admit that the overall understanding of the nature of the fatigue loads applied to axle is still partial. One of the most recent studies on the subject is the one that was performed by DeltaRail [12]. In this work, the main sources of variability are assumed to be:

- The track characteristics: track curvatures and track quality;
- The train velocity;
- The position of the axle: in front or at the rear of the vehicle;
- The masses supported by the vehicle: empty to fully loaded vehicle.

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The track quality is difficult to characterize. A first approach could consist in using the track quality indicators as defined in leaflet UIC518 for train homologation purposes [22]. But as noticed in [12], there is no scientific work that exhibited a clear relationship between the track quality indicators and the axle loads. For this reason, it is suggested not to use this parameter to classify loads but to ensure that when performing load measurements, different types of tracks are circulated (with different traffics and different strategies of maintenance).

Observations of CEN/TC 256/SC 2/WG 11: The effect of track quality on the axle loads and the underlining fatigue impact on the axle design is significant and cannot be neglected. As a limited test campaign can never cover at full extent the track quality and maintenance of a network infrastructure (in quantity and in time), the variability of the loads measured should be covered, in the axle design, by the use of an adapted safety coefficient. This safety coefficient could be assessed by statistical analysis or alternative methodology. In the current axle design standards (EN 13103 series) the track quality effect on the fatigue assessment is taken into account through safety factors. These safety factors assessments are documented in NF F 01–118 [4].

Step 4. Once the load signals are cut and classified according to the three parameters (curvature, train speed, axle position), it is necessary to generate new load spectra.

First of all, it is assumed that any axle is 50 % of the time in the front of the vehicle and 50 % of the time at the rear of the vehicle since, in general, a train does back and forth paths without making U-turn in the stations. So for each virtual load spectrum that is generated, half of it derives from signals corresponding to a front position of the axle in the vehicle and half of it derives from signals corresponding to a rear position.

Observations of CEN/TC 256/SC 2/WG 11: The load spectrum of bending moments is a result, among others, of the Y guiding forces and a portion of these forces, within the spectrum, is unknown. For this reason, using an existing load spectrum of a vehicle to design a new wheelset for a new vehicle is challenging. Therefore, the generation of specific load spectra can only be done by measurements on the intended rolling stock application. One solution to avoid this constraint would be to aggregate the bending moment load spectrum with the portion of the relevant loading parameters (Y, Q forces, etc.) measured at the same time on an existing comparable vehicle and service conditions (for example, track quality and geometry), in order to be able to convert one type of wheelset to another one.

Secondly, the train speed and the curvature classes are defined. Typically, 10 classes can be defined for the train velocity, from zero to the maximum speed allowed on the lines. The curvatures can correspond to tangent tracks or curves (with 2 or 3 classes of curvature). Information on the ratios of curvatures in the railway network that is studied and on the associated maximum allowed speeds can be used to determine probabilities of having a couple ($speed_{class\ i}$, $curvature_{class\ j}$).

Step 5. Finally, random drawings in the database of elementary load spectra, following the probabilities determined above, can be held. Attention shall be paid to be sure that the method does not generate unrealistic data (for example, very high speeds in tight curves).

But there is one important parameter left, which is the mass that is being carried by the axle.

To take into account this parameter, it is first necessary to establish its effect on the M_{eq}. A short piece of load data (30km) generated in the European project Dynotrain [23] is used for this study. Two similar freight wagons circulated with empty or fully loaded conditions. By analysing the signals of both freight vehicles in tangent track portions, the mean values of the vertical loads can be estimated (Q1+Q2)/2 = 23 kN for the empty vehicle and 106kN for fully loaded one. The calculated M_{eq} are: M_{eq}(empty) = 11,85 kNm and M_{eq}(fully loaded) = 49,94 kNm. It shows that the ratio between the masses supported by the empty and loaded axles and the ratio between the M_{eq} of the empty and the loaded axles are quite similar. Other simulations are performed with other SNCF data and they confirm this result: the M_{eq} is almost linearly dependent of the carried load.

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In other words, if a fatigue-equivalent-load $Mx_{Formula\ 1}$ is calculated for an axle supporting a mass M1, the fatigue-equivalent-load $Mx_{Formula\ 2}$ for the same axle circulating in the same area with the same velocity but supporting a mass M_2, can be estimated using the following formula:

$$Mxeq2 = \left(M2 / M1 \right) Mxeq1 \quad (5)$$

By considering a realistic distribution of mass during an axle life, based on operation data, each fatigue-equivalent-load of each virtual load spectrum is "multiplied" by this distribution of mass, generating a new load spectrum for which, finally, a final fatigue-equivalent-load can be calculated.

Step 6. All virtual load spectra are analysed with the FEL method described in 3.1.

Step7. The values of the calculated M_{eq} are gathered and their distribution can be modelled (Gaussian or other statistical distributions).

4.3 Fatigue reliability assessment of a railway passenger coach axle

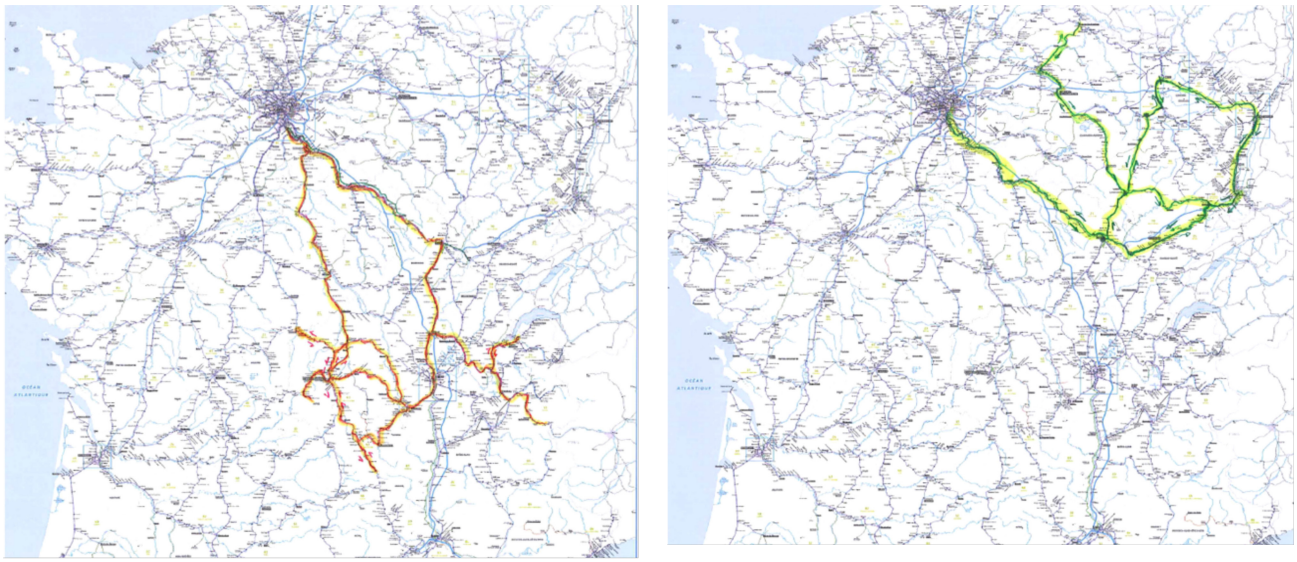
4.3.1 Load measurements

A long test campaign was carried out by SNCF on a non-powered axle of an empty passenger coach circulating on various French lines (see Figure 12) in 2012. The total circulated distance is more than 2000km. The instrumented train was travelling several times on the same paths, with small variations of the speed to allow variability but to remain realistic. The maximum speed was V = 160km/h. The axle load was 14t.

A light measurement acquisition system (black box recorder with 16 recorded signals, with a battery allowing a good autonomy, GPS for the localization) was used. To measure the forces, holes with strain gauges around were implemented in the wheels in order to capture the stresses generated by the deformation of the holes. A calibration was done with a specific bench to calculate the Y and Q values from the stresses. With this method, a relatively good estimation of the forces at the wheel-rail contact can be made. The advantage of such a measurement strategy, compared to a direct measurement on the axle, is to have information on the wheel loads themselves, useful for other research studies carried out on the wheels. The measured wheel/rail forces on the left and right wheel were low pass filtered with a cut off frequency of 20 Hz for Q and 10 Hz for Y, in order to be consistent with the leaflet UIC518 [22].

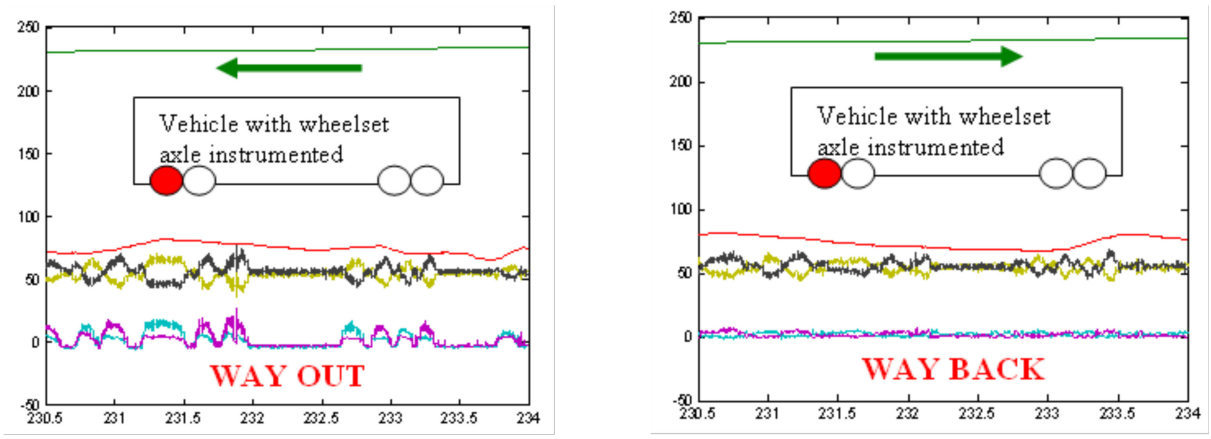
For both test campaigns, the position of the tested axles is 50 % in front of the vehicle and 50 % at the rear of the coach in terms of circulated distance of circulated path, as illustrated in Figure 13.

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a) First test campaign b) Second test campaign

Figure 12 — SNCF Test campaigns



a) Tested axle in front of the vehicle b) Tested axle at the rear of the vehicle

Figure 13 — Axle position during the SNCF tests

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The main features of the passenger coach are reported in the following table:

Table 2 — Passenger coach features

m1	13 960	kg
m2	1 200	kg
H1	1 425	mm
b	1 000	mm
s	750	mm
P1	102 668	N
P2	68 517	N
Y1	47 932	N
Y2	23 966	N
Q1	114 770	N
Q2	54 665	N

The following data can be extracted:

- Instant of acquisition (s);
- Position in the track (PK);
- Velocity of the train (km/h);
- Loads Y and Q on both wheels (kN);
- Curvature (1/m) after conversion of the gyro meter's signal information (°/s).

The load spectra of both test campaigns and their fatigue-equivalent-load values $M_{x,eq,}$ calculated with a Basquin curve with two slopes following the "2k-1" rule, with k = 8, and an axle lifetime equal to $7{,}10^6$ km, are given in Figure 14. The left one corresponds to the mountainous area. The right one corresponds to the other test area.

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