A NEW RELIABILITY APPROACH FOR THE FATIGUE DESIGN OF RAILWAY WHEELS

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Abstract. Wheels are part of the safety critical railway components. Hence, they are designed, manufactured and maintained in order to avoid any in-service failure. One of the main potential causes of breaking is the fatigue of the web. Therefore, to make the homologation of a wheel, fatigue requirements are defined in national and European standards which describe the numerical simulations and the tests procedures, the loads to be tested and the acceptable limits which include some safety margins.

These design, production and maintenance rules enable the railway industry to offer one of the most safe transport technology. Though, as they have essentially been established on the basis of the feed-back on existing materials and traffics, they are interdependent and do not leave place to innovations that would bring lighter or environmentally friendly solutions. At least, when a new design is introduced, some precautions in the maintenance have to be taken, increasing considerably the costs for the operators.

These observations indicate that existing design and maintenance rules need to be reviewed - or at last verified - so as to be adapted to a fast changing market environment and to allow for the safe introduction of innovations.

The aim of the study presented in this paper is to develop a new comprehensive fatigue design approach for the railway wheels in order to guaranty their in-service durability and reasonable maintenance costs.

The methodology starts with a loading analysis consisting in transforming temporal in-service load measurements into more simple "Fatigue-equivalent" loads, building the operating loads severity distribution and deducing relevant specifications. The resistance distribution of the wheels is then characterized. A probabilistic treatment of the data, based on the Stress and Strength Interference Analysis, determines the probability of failure. A correlation between the results obtained in this project and the current standards for existing wheels is finally made. The final objectives are to bring some technical elements to facilitate the homologation of innovative solutions or to quantify the impact of an evolution of the operating usages and, from a scientific and technical point of view, to help to switch from a traditional "safety margin approach" to a reliable approach.

1 INTRODUCTION

The paper begins with the description of the current standards for wheels which defines both specification load values and the range of admissible fatigue stresses. The methodology of fatigue design method is then described:
- it starts with the description of the load measurement method in service.
- the stress and strength interference method [1] is introduced to define the "equivalent load" specification in a general context.
- the "equivalent load" relative to the wheel is defined and the associated distribution is build.
- the specification load is computed and compared to the standard.

The paper finishes with the description of some open questions and a conclusion.

2 CURRENT STANDARD FOR WHEELS

The wheel (Figure 2) is subjected to different situation of life. The figure 1 shows the 3 cases described in the standard [2].

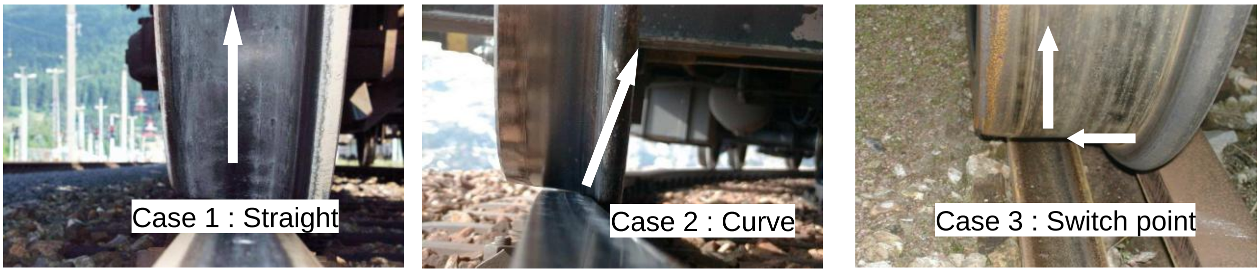

Figure 1: the 3 situations of life for the wheel and definition of the standard cases

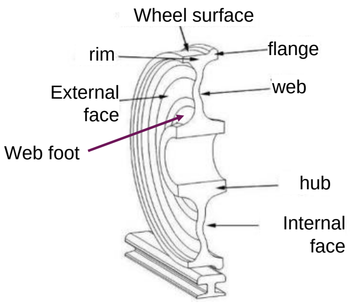

Figure 2: the wheel

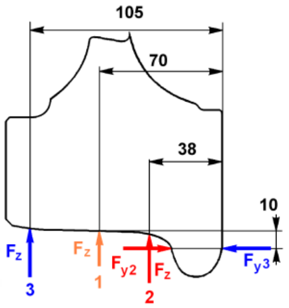

Figure 3 : standard loads

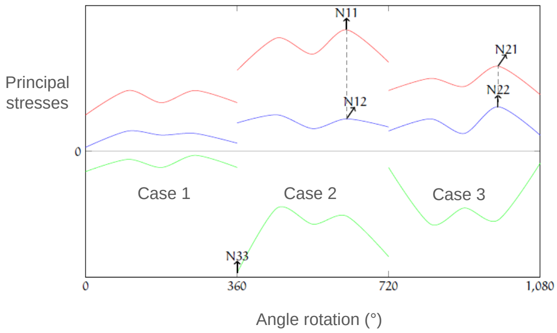

Figure 4: Principal stresses (at a point)

According to these situations of life, the standard defines the position of loads and the values : Fz = 1.25 P, Fy2= 0.7 P, Fy3 = 0.42 P (P is half of the axle load and the relation is dedicated to driving axles)
Theses values defines the Y (transverse) and Q (vertical) wheel forces in each 3 cases. The press fit load case is also considered in the standard (the interference between the hub of the wheel and the axle is about 0.18 mm).

The wheel is acceptable according to fatigue phenomenon if the dynamic stress is less or equal the admissible stress ($\sigma_{dyn} < S_a$)
S_a is material dependant whereas the dynamic stress is computed as follow:
The standard defines Nij principal directions (see figure 4): N11 and N22 are the directions associated to the maximum of the principal stresses for the 3 cumulated cases. N12 is the second principal direction at the same point and position where the first principal stress is maximum. N33 is the principal direction where the 3rd principal stress is minimal. For each N, $\sigma_{NN}(t)$ must be computed and the relation $\sigma_{dyn} < S_a$ must be true.
$\sigma_{NN}(t) = (\ \sigma(t) \cdot N\ ) \cdot N$
$\sigma_{max} = \max_t \sigma_{NN}(t)$, $\sigma_{min} = \min_t \sigma_{NN}(t)$ and $\sigma_{dyn} = |\sigma_{max} - \sigma_{min}|$.

From this fatigue criterion, we can observe that the variability of loads is not considered: the standard is a deterministic approach. The fact that the 3 cases are equally considered allows us to think that the standard is severe.

3 LOAD MEASUREMENTS IN SERVICE

The reliability approach is based on real measurements. It is described here how to build a "matrix load" from a wheel equipped by strain gauges. From such an equipped wheel, it is possible to acquire the real Y and Q forces in the time domain (Figure 5). Most of the cases, Y and Q forces are approximately constant over a wheel revolution. The hypothesis that we have one cycle fatigue for one revolution is done here. Consequently, a matrix load can be built counting each occurrence of load for each wheel revolution.
Figure 6 is plotted an example of such a Matrix load for a running test (hundred kilometers of data).

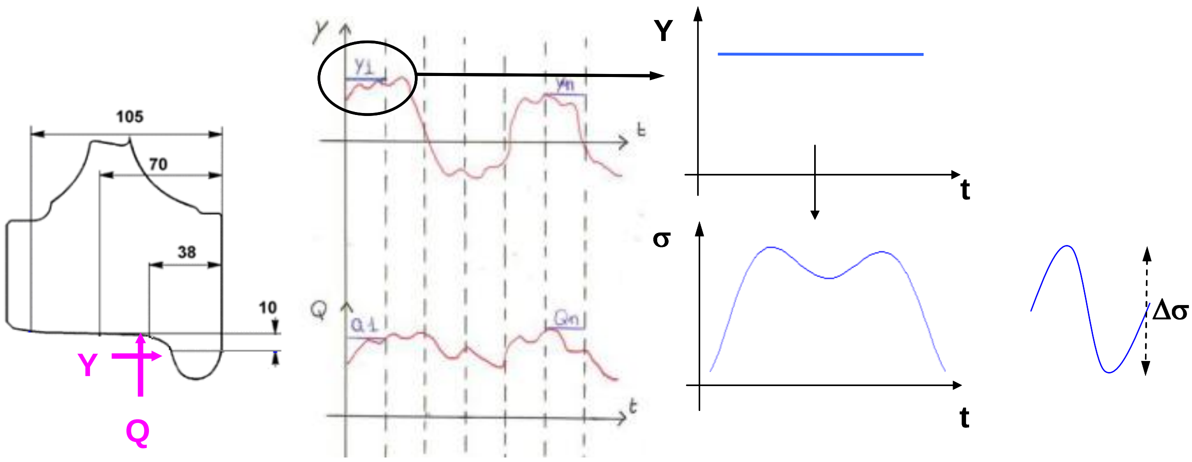

Figure 5: load measurements in service and cycle fatigue

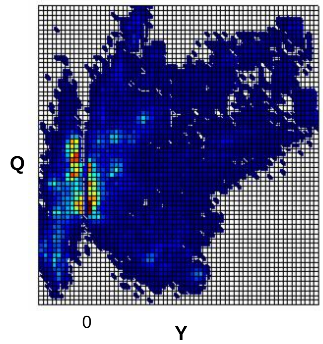

Figure 6: Load matrix from measurements

4 THE STRESS AND STRENGTH INTERFERENCE METHOD

The aim of the stress and strength interference method [5] is to specify the distribution of strength which gives the aimed probability of failure Pf in terms of the distribution of loads. In practice, the objective is to determine the equivalent load Fn which has to be tested in simulation and test (specification load).

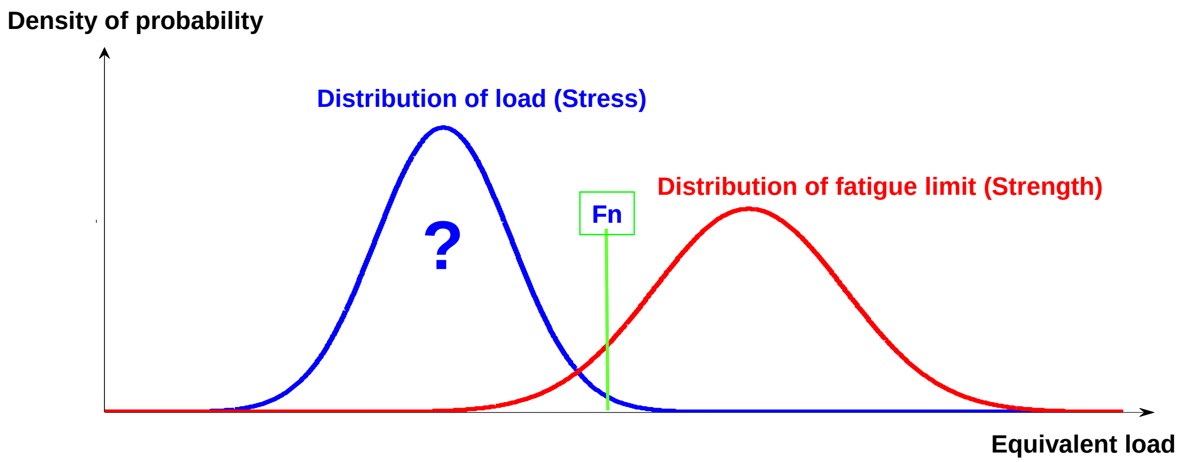

Figure 7: distributions of load and strength in term of equivalent load

A point of the distribution of load represents an equivalent load for a given life situation. The distribution of load represents the loads for all the life of the wheel.
The probability to reach an equivalent load greater than Fn is called the Severity: we will define Fn for a severity S = 10^{-4}. What is the equivalent load for the wheel? How to built the distribution? And then how to compute the load specification Fn ? This is the object of the next section.

5 DISTRIBUTION OF LOAD AND COMPARISON TO THE STANDARD LOAD SPECIFICATION

5.1 Equivalent load definition and calculation

The generic equivalent load defined in the last section has to be defined in the context of the wheel. The loads are the Y, Q and the press fit. In the standard, as it is defined in the section 2, the values of Y,Q forces and their application points depends of the load case. A study of the stress path [3,4] has shown that at web foot point, the damage is mainly governed by the bending moments created by the Y and Q loads. Starting from this result, we propose to define generalized forces Qg and Yg that are equivalent to the standard cases. The application points of Qg and Yg are always the same for all the standard cases.

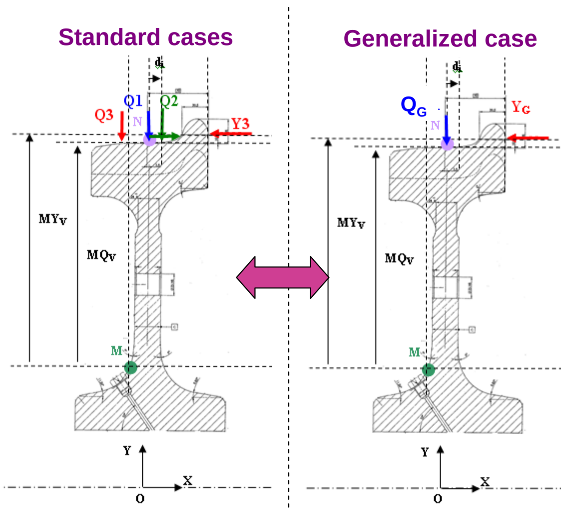

Figure 8 : Definition of a generalized load case

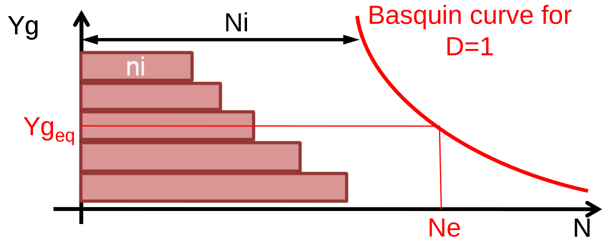

Figure 9 : Equivalent load calculation method

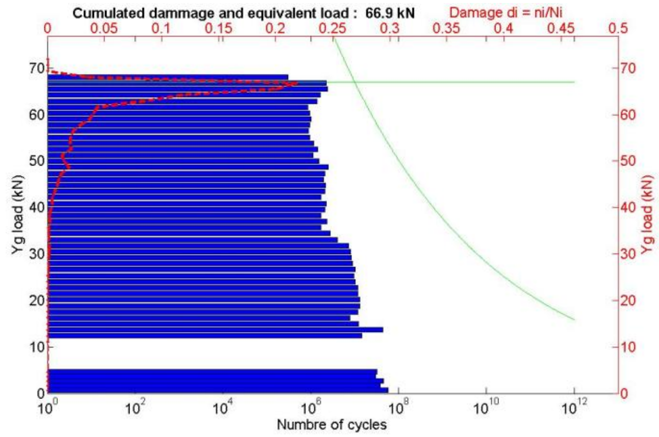

Figure 10 : Computed Yg_{eq} load for a running test data

The values of Yg depend on the cases:
Case 1 (-5kN<Y< 5kN) : Yg = Y,
Case 2 (Y> 5kN) : Yg = Y + α_2 Q,
Case 3 (Y< -5kN) : Yg = -Y - α_3 Q,
α_2 and α_3 are chosen to be sure that Yg creates a bending moment equivalent to the bending moment in the standard cases at the web foot point. For the definition of the equivalent load, we propose to neglect the influence of Qg and the press fit: the equivalent load defined figure 7 is Yg.

From a given set of loads Y,Q (a given situation of life), we propose to define a scalar equivalent load Yg_{eq} in term of damage to this set of loads. The general method is described in [1]. The general idea is the following:

a. The Basquin damage law is identified from Wöhler curve: $N.(\Delta\sigma)^m = C1$
From the fatigue tests, we can identify Ne = 10^7 cycles which is a good number to reach the fatigue limit and the Basquin coefficient m = 8.
b. From the hypothesis that the damage is directed linked to Yg, we have $\Delta\sigma$=k. |Yg|. The damage law can be expressed in term of Yg: $N.|Yg|^m = C2$
c. If the Basquin curve in Yg is plotted with the occurrences in Yg (figure 9), it is possible to fit the C2 coefficient to get a damage D equal to 1. (the damage is evaluated with the miner law : D= Σ(ni/Ni))
d. The Yg_{eq} can be chosen for a Ne number of cycles.

Application: in the Figure 10 are plotted occurrences in Yg from the running test data (from the load matrix figure 6) and Yg_{eq}. The number of cycles are extrapolated from the hundred kilometers of data taking into account the total number of kilometers viewed by a wheel in it entire life: about 1 million kilometers.
For the running test, we obtain Yg_{eq}=66.9 kN.

5.2 Distribution of load and load specification calculation

To build the distribution of load defined in the section 4 (and compute the specification load), it is proposed to build virtual load matrices from the unique running test data (Y & Q loads).
To achieve that, we have decomposed the original signals to isolate some particular configurations: we consider 3 classes of train velocity and 3 classes of track curvature:

Train speed V (km/h) classes	Track curvature r (m) classes
VL : 60 < V < 90	GC : r > 610 m
VM : 90 < V < 105	PC : r < 610 m
SV : 105 < V < 125	A : $1/r \cong 0$ m

From the original temporal loads, new elementary load matrices are computed for each class. The figure 11 shows these matrices.

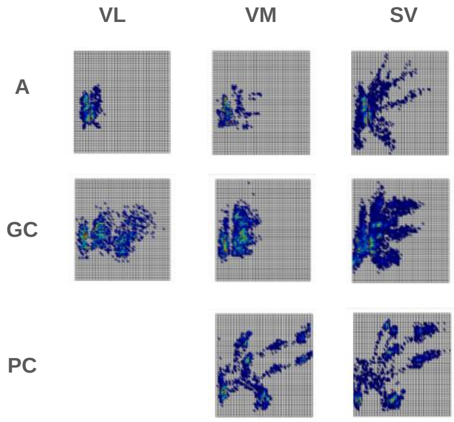

Figure 11 : Example of elementary load matrices

From these elementary load matrices, hundreds of virtual load matrices are computed (by linear combinations of the normalized elementary load matrices). It is then possible to compute a set of hundreds values of Yg_{eq} representative of the fatigue loads for the entire life of the wheel.

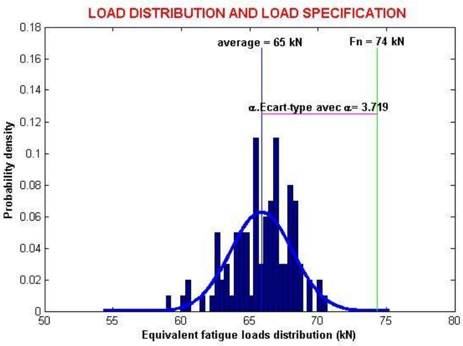

Figure 12 : Load distribution
and load specification (for case 2 only)

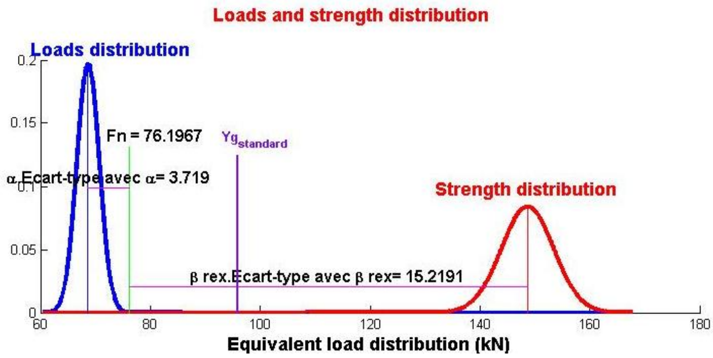

Figure 13 : Load distribution, load specification Fn, standard load $Yg_{standard}$ and the strength distribution (for cumulated cases)

The method has been applied in 2 contexts:
(a) First, the cases have not been cumulated: the goal is to compare the obtained specification value by the method (for $S = 10^{-4}$) and the standard load values for separate cases (we suppose for that the wheel is subjected to the considered case in his entire life).
(b) Secondly, the cases have been cumulated: the goal is to compare the obtained specification value by the method (for $S = 10^{-4}$) and the $Yg_{standard}$ load which gives the same severity than the 3 cumulated standard cases.

<u>Results in the context (a)</u>
In the figure 12, we can observe the distribution of the equivalent Yg loads restricted to the case 2. To compare to the standard case 2 from Yg, it is necessary to traduce it in term of Q and Y values. We have decided to fix the Q value according to the Q standard value (120 kN) and to evaluate the corresponding Y value ($Y = Yg - \alpha_2 Q$) :
Standard approach: Qn = 120 kN, Yn = 67 kN
Probabilistic approach: Qn = 120 kN, Yn = 64.1 kN (equivalent to $Fn=Yg_n$=74 kN)
We can conclude that the results are very near in this case.

Results in the context (b)
In the figure 13, we can observe the distribution of the equivalent Yg loads for the cumulated cases. The specification load value obtained is Fn = Yg_n=76.2 kN. Comparing this value to the standard is not so easy as before:
We have used a Finite Element model of an existing wheel and we have calculated the stress path for the 3 standard cases. We have evaluated a fatigue criterion $DV_{standard}$ (Dang Van fatigue criterion) associated to the cumulated 3 stress path. The $Yg_{standard}$ value is the Yg that gives the same fatigue criterion $DV_{standard}$. We found $Yg_{standard}$ = 96 kN.
The distribution of strength has also been plotted in the figure 13. The knowledge of the fatigue limit of the material combined with the Finite Element model has been allowed us to build this distribution in the Yg domain.

We can observe that the specification value is lower than the standard value. This is because the standard considers that the wheel is subjected to the 3 cases alternatively (and equally considered) in his entire life which is very severe. For an existing wheel we have verified than the probability of failure (Pf) is very low.

6 OPEN QUESTIONS AND CONCLUSIONS

A probabilistic approach has been developed for the definition of load specifications for wheels at the web foot. A comparison of the results has been done with the standards. The main results are:
- for the separate case (case 2), the found load specification is very near the standard one (for a severity $S=10^{-4}$).
- for the cumulated cases, the standard is more severe in terms of load specification.
We have checked that the probability of failure is very low.

There are still open questions after this study. All the obtained results come from hundred kilometers of a running test. The representativity of this running test must be checked with the use of more data.
These results concerns the web foot of the wheel. For different geometries of wheels, critical points can be in other positions where the damage can depends on Y,Q and the press fit. The probabilistic approach must be extended in a multi dimensional approach in this case.
The cases transition must be studied in details to check that new cycle fatigue with high magnitude is not created.

7 ACKNOWLEGEMENTS

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8 REFERENCES

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