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INTERNSHIP AT HUISMAN EQUIPMENT,
RESEARCH & DEVELOPMENT DEPARTMENT, IMEP TEAM

IMEP WHEEL & IMEP SECTIONS

Author: Jelmer Goerres (s1381644)

Faculty of Engineering Technology (CTW)
Mechanics of Solids, Surfaces and Systems (MS3)

Supervisors:
UT: André de Boer
Company: Ashley Nuttall

UNIVERSITY OF TWENTE.

Abstract

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This four month internship was done at Huisman's Research and Development department. It consisted of improving Huisman's internal Excel calculation software (IMEP). The two programs that were worked on are IMEP wheel, a wheel design validation tool, and IMEP sections, a beam profile validation tool.

The improvements to IMEP wheel are twofold: First two official standards for wheel design are added to the program. Secondly a calculation method for determining Fatigue for multiple load cases was added for the current method as well as the two extra standards.

For IMEP sections the use of an optimization tool is investigated. This tool should minimize the surface area and thus mass of the profile without Yielding. Two optimization methods are compared: A standard profile search and a build-in excel solver. The Standard Profile search algorithm efficiently searches through the known standard profiles and selects the profile that fulfills the stress requirement with the lowest surface area. The build-in excel solver varies profile dimensions to finds the lowest possible surface area. This yields a profile with a very low thickness, mainly because there is no criteria for Buckling added. Furthermore the solver is not very robust.

When increasing the amount of profiles from a single to multiple profiles, issues with intersecting of profiles occur. This is tackled by declaring relations between profile positions. Furthermore a large size difference between profiles can occur. This can also be prevented by adding relations between profile dimensions. With these relations it is possible to get an optimum for multiple profiles for the excel solver, as well as the Standard Profile search. However the solver is still not robust and the results from the Standard Profile search are therefore preferred.

Contents

1|Introduction

This four month internship was carried out at the Huisman Research and Development department. The Internship was focused on the Integrated Mechanical Engineering Program (IMEP). IMEP is a set of Excel worksheets used by Huisman engineers to design equipment. Two IMEP sheets have been improved: Wheel and Sections.

IMEP Wheel is a tool to check the design of driving crane wheel and rails. In the old program this check was done using the Horrowitz method. Two standards for wheel design have been added to this: EN 13001-3-3 and FEM 1.001 b4/ISO 16881-1. A rail failing from fatigue resulted in a demand to radically change the way the program handles fatigue: A calculation method for determining Fatigue for multiple load cases was added for the current method as well as the two extra standards. This meant the program had to be rebuild completely. Additionally the manual and work instructions for this program has been rewritten to include this new approach.

IMEP sections is a tool to check design of profiles from beam constructions (I-profiles, U-profiles etc.). An optimization tool is added to this program to minimize the area of the profile for a given load case.

The IMEP wheel sheet was successfully completed. Since the redesign of the IMEP wheel sheet was urgent and not intended, the time spend on the optimization tool was less then intended and indeed required. This results in a Optimization tool that works but is not yet ready for usage by the Huisman Engineers, as it still requires testing and optimization of the program.

2|Wheel Standards

The current IMEP sheet is outdated and requires an update. First of all the wheel design is only checked by a non-standard method of Horrowitz. For legal reasons three actual standards for validation of crane wheel design are added. Next to this, the load cases acting on the wheel are defined wrong: Each load case is considered individually. As the failure that is checked for, Fatigue, is a result of all accumulative load cases. The sheet only checks if the defined force is below the allowed force, which gives very limited insight in the behaviour of the wheel. The amount of information given by the sheet has to be increased and the fatigue caused by all the load cases has to be taken into account.

First the old method for wheel validation and the new standards are explained. The added Standards are:

- EN 13001-3-3[1]
- FEM 1.001 b4 [2]
- ISO 16881-1 [3]

From these standards, FEM and ISO give the exact same definition and will thus be regarded as the same.

The methods differ in a number of ways. For one the Horrowitz method from the old sheet and the EN 13001-3-3 standard give an allowed force for both the wheel and the rail, While FEM/ISO only give the force for the wheel. That is why these standards are absent for all rail related matters. The full definition of the three methods is given below. This definition consists of:

- The Allowed Static Force, given as the Allowed force for a very small amount of wheel rotations
- The Allowed Fatigue Force, given as the Allowed force for any given number of wheel rotations higher than the static wheel rotations.

The forces acting on the wheel are defined slightly different for the different methods:

- Horrowitz uses the Allowable stress method, which uses non factored Forces and adds a safety factor to the Allowable stress.
- EN 13001-3-3 and FEM/ISO use the Limit state method, which adds partial safety factor to the Forces instead of the Allowed stress.

For the static load cases this difference should be taken into account when declaring load cases. For the Fatigue load cases the partial safety factors are set 1 and the LSM forces are equal to the ASM forces.

The Load cases on the wheel are defined in ISO 8686 [4] / EN 13001-2 [5]. These standards make a distinction between three types of load cases:

- Load cases A: Regular loads occurring during normal operation. E.g. :Gravity, braking, accelerating and Hoisting

- Load cases B: Occasional loads that occur infrequently. E.g. : in-service wind ice and snow loads and skewing
- Load cases C: Exceptional loads that occur very little or may even not occur. E.g.: loads caused by testing, out of service wind, buffer forces and tilting, emergency cut-out, failure of drive components and external excitation of the crane foundation.

Load case C occurs so little it can be considered as static forces. That is why these forces are considered as static forces. For FEM/ISO the fatigue load cases contain load cases A and B. EN 13001-3-3 only take load cases A into account for fatigue. For Horrowitz the load cases to take into account are not specified. When adding Forces that are not defined in ISO 8686/EN 13001-2 consider in which load case category (A, B or C) this load case would be.

2.1 Horrowitz

The Horrowitz method is the method used in the old sheet. This method is created by Horrowitz based on the Hertze stress, A stress calculation for contact between round objects. Horrowitz declared an allowed Hertze stress as a function of the amount of rotations based on the Hardness. It is applicable for each contact situation for which the Hertze stress can be determined so the objects need to have a radius. The Allowed Hertze stress Yields an allowed force.

2.1.1 Allowed Static Force

This method does not give a definition of the Allowed Static Force. This Force is estimated by filling in the amount of rotation that is considered static into the Allowed fatigue force. The allowed Fatigue Force is given below and the amount of rotations used is given in Chapter 3.10, specifically equation 3.10.

2.1.2 Allowed Fatigue Force

As mentioned Horrowitz gave the Fatigue criteria as an allowed Hertze stress. This criteria states that:

$$\sigma_H \leq k_h 10HB \tag{2.1}$$

With:

- σ_H is the Hertze stress(N/mm^2);
- HB is the Brinell Hardness of the wheel, roll, ball or counter surface(N/mm^2). The difference in hardness between the surfaces should be above 100 N/mm^2 to prevent galling. despite not being mentioned in it this applies for all standards;
- k_h is the Stribeck value(-);

The Stribeck value is a function of the amount of wheel rotations:

$$k_h = \max(0.812 - 0.07\log(n), 0.27) \tag{2.2}$$

Where any Stribeck value lower than 0.27 yields such a low allowed force that the wheel will never fail due to fatigue, hence the cap. The Hertze stress is given by:

$$\sigma_H = \sqrt{\frac{FE}{\pi b_{eff} D_{eq}}} = 0.591\sqrt{\frac{FE}{b_{eff} D_{eq}}} \tag{2.3}$$

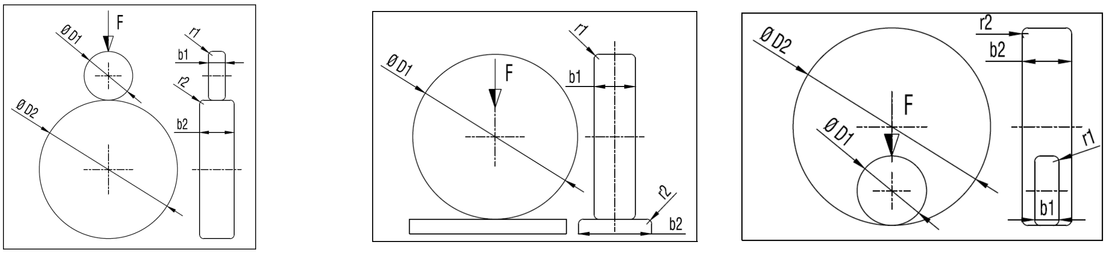
with:

- F the force on the wheel(N);

- E is the Young's modulus(N/mm^2);
- b_{eff} is the effective width(mm);
- D_{eq} is the diameter of the wheel(mm);

The equivalent diameter depends on the contact method between the wheel and the rail. Three contacts are defined, as shown in figure 2.1:

- Wheel on Wheel $D_{eq} = \frac{D_1 D_2}{D_1 + D_2}$
- Wheel on Surface $D_{eq} = D_1$
- Wheel in Wheel $D_{eq} = \frac{D_1 D_2}{|D_1 - D_2|}$



(a) Wheel on Wheel (b) Wheel on Surface (c) Wheel in Wheel

Figure 2.1: Contact methods

The effective width is give by:

$$b_{eff} = \min(b_1 - 2r_1, b_2 - 2r_2) \tag{2.4}$$

With, as shown in figure 2.1:

- b_1 is the wheel width(mm);
- b_1 is the counter surface width(mm);
- r_1 is wheel edge radius(mm);
- r_2 is counter surface edge radius(mm);

Although increasing the edge radius increases the Hertze stress it is advised to have at least some edge radius to prevent high edge stresses, which are not seen with the Hertze stress.

The Hertze stress equation can be rewritten to a maximum allowed fatigue force:

$$F_{fatigue} = \frac{b_{eff} D_{eq}}{E} \left(\frac{k_h 10HB}{0.591} \right)^2 \tag{2.5}$$

2.2 EN 13001-3

The EN 13001-3-3 standard is limited to wheels or rails with a radius in the width direction (crown radius) less than 5 times the edge radius.

According to this standard the wheel or rail has failed at a permanent radial deformation of 0.02 % of the wheel radius.

2.2.1 Static Force

The allowed static load is defined as:

$$F_{static} = \frac{A_s^2 \pi D_{eq} b_{eff}(1-v^2)}{\gamma_m E_m} f1 f2 \tag{2.6}$$

with:

- γ_m is the general resistance coefficient; γ_m= 1,1(-);
- D_{eq} is the diameter of the wheel(mm);
- b_{eff} is the effective load-bearing width taken as $b = min[br; bw]$(mm) ;
- v is the Poisson's ratio (0.3 for steel)(-);
- $f1$ and $f2$ are Influence factors, which are explained below;
- E_m is the equivalent Young's modulus (N/mm^2), given by:

$$E_m = \frac{2E_{wheel}E_{rail}}{E_{wheel} + E_{rail}} \tag{2.7}$$

- A_s is a variable depending on whether hardening is applied. For non-Hardened material it depends on the Brinell harness (HB):

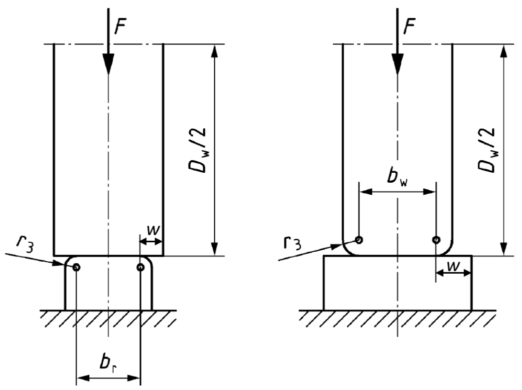
$$A_s = 7HB \tag{2.8}$$

  For surface hardened material, where surface hardness is equal to or greater than 0.6 times the yield stress(σ_y), it is assumed that the yield stress is the main failure mechanism, so A is given by:

$$A_s = 4.2\sigma_y \tag{2.9}$$

This Factor shows a very important difference between the Horrowitz standard and EN 13001. EN 13001 assumes that when the material is Hardened the Hardness it is no longer the criteria for failure. Instead the yielding caused by the internal stress will cause failure. For Horrowitz an infinitely hardened material would produce an unbreakable wheel, while for EN 13001 the plastic deformation will cause failure. Obviously the latter is more realistic, but the criteria for Hardening ($0.6\sigma_y$) is very low. A much higher hardness can be achieved. In Chapter 4 this difference and the implications are discussed in depth.

Influence factor $f1$ depends on the edge pressure in line contact. For point contact this factor is 1. The factor depends on the edge radius and the width of the non-contacting area as shown in 2.2



Ratio r_3/w	Factor f_1
$r_3/w \le 0,1$	0,85
$0,1 < r_3/w < 0,8$	$[0,58 + 0,15 (r_3/w)] / 0,7$
$r_3/w \ge 0,8$	1,0

where
w is the width of the projecting non-contact area and
r_3 is the radius of the edge of the non-projecting part (wheel or rail).

Figure 2.2: Influence factor $f1$

Influence factor $f2$ depends on the non-uniform pressure distribution in line contact. An ideal uniform distribution across the tread of the wheel in the line contact case is dependent upon sufficient elasticity of the rail fixing or its support and/or wheels with self-aligning suspension. Otherwise, deformation of the crane structure (e.g. bending of main girders) or tolerances in rail alignment result in non-uniform pressure distribution, decreasing the limit design contact force. The value of this factor can be found in figure 2.3a. The tolerance class is shown in figure 2.3b. Huisman cranes are almost always in Tolerance class 3 or 4.

	Tolerance class of ISO 12488-1			
	1	2	3	4
Wheels with self-aligning mounting	1,0	1,0	0,95	0,9
Non-aligning wheel mounting, rail mounted on elastic support	0,95	0,9	0,85	0,8
Non-aligning wheel mounting, rail mounted on rigid support	0,9	0,85	0,8	0,7

(a) value of f2 for certain wheel mounting

Tolerance class	Limits of travelling and traversing distance km
1	50 000 ≤ L
2	10 000 ≤ L < 50 000
3	L < 10 000, for stationary erected tracks
4	Temporarily erected tracks for building and erection purposes
NOTE L is calculated as the product of the normal travel speed and the specified working time of the relevant travel/traverse mechanism, either by application of customer specified values or through reference to the classification of the mechanism (see ISO 4301-1).	

(b) Tolerance class according to ISO 12488-1

Figure 2.3: Influence factor $f2$

2.2.2 Fatigue force

The allowed fatigue force for EN 13001-3-3 is given by:

$$F_{fatigue} = \frac{f_u}{\gamma_{cf} \sqrt[m]{s_c}} f_f \tag{2.10}$$

with:

- f_u is the reference contact force(N);
- s_c is the contact force history parameter, calculated separately for wheel and rail(-);
- γ_{cf} is the contact resistance factor for fatigue $\gamma_{cf} = 1.1$(-);
- f_f is the factor of further influences;
- m is the exponent for wheel/rail contacts, m = 10/3 = 3,33(-).

The reference contact force (f_u) is given by:

$$f_u = A_f^2 \frac{\pi D_{eq} b_{eff} (1 - v^2)}{E_m} \tag{2.11}$$

For fatigue force, the hardening factor is a bit different from the static hardening factor. For Non-hardened material A_f is now:

$$A_f = 3.0HB \tag{2.12}$$

For hardened material it is:

$$A_f = 1.8\sigma_y \tag{2.13}$$

The contact force history parameter s_c is given by:

$$s_c = k_c v_c \tag{2.14}$$

with:

- k_c is the contact force spectrum factor. This factor depends on the contact forces in each rolling contact;
- v_c is the Relative number of contacts;

k_c is given by:

$$k_c = \frac{n_i}{n_{tot}} \sum_{i=1}^{n_{tot}} (\frac{F_i}{F_{max}})^m \tag{2.15}$$

with:

- n_{tot} is the total amount of rotations for a wheel and passings for rails.
- n_i is the amount of rotations/passing of the current load case (i)
- F_i and F_{max} are the force in the i^{th} load case and the maximum force of all load cases;

v_c is given by:

$$v_c = \frac{n_{tot}}{i_D} \tag{2.16}$$

with:

- i_D is the reference amount of rolling contact used to scale the force to the reference force. $i_D = 6.4e6$.

The influence factor f_f takes into account multiple separate factors and is given by:

$$f_f = f_{f1} f_{f2} f_{f3} f_{f4} \tag{2.17}$$

with:

- f_{f1} is the edge pressure factor, which is the same as f1 for the static force;
- f_{f2} is the non-uniform pressure distribution factor, which can be taken 1 for fatigue;
- f_{f3} is the skewing factor, which depends on the skewing angle α of the crane, which is given in figure 2.4. f_{f3} is 1 for $\alpha \leq 0.005$ rad.α must always be lower than 0.015 rad. f_{f3} is given by:

$$f_{f3} = \sqrt[3]{\frac{0.005}{\alpha}} \tag{2.18}$$

- f_{f4} is the mechanical drive factor, which is 1 for non-driven wheels or wheels in environment without abrasive particles and 0.95 for for driven wheels in environment with abrasive particles

Skew angle resulting from		Flanged wheels	Guide rollers
α_g	Track clearance	$\alpha_g = s_{g\,min} / w_b$ when $s_g \leq \frac{4}{3} s_{g\,min}$ $\alpha_g = 0{,}75 \cdot s_g / w_b$ when $s_g > \frac{4}{3} s_{g\,min}$	
		Crane travelling	
		$s_{g\,min} = 10$ mm	$s_{g\,min} = 5$ mm
		Trolley traversing	
		$s_{g\,min} = 4$ mm	$s_{g\,min} = 2$ mm
α_t	Tolerances (wheel alignment and straightness of rail)	$\alpha_t = 0{,}001$ rad	
α_w	Wear	$\alpha_w = 0{,}1 \cdot b_h / w_b$	$\alpha_w = 0{,}03 \cdot b_h / w_b$

where

w_b is the wheel base (i.e. distance between guide rollers or between first and last wheel)

s_g is the actual track clearance of the guide means

$s_{g\,min}$ is the minimum track clearance of the guide means for the purpose of calculations

b_h is the width of rail head

Figure 2.4: Classification of crane skew angle in accordance with EN 13001-2

2.3 ISO/FEM

ISO and FEM use the same calculation for allowed static and Fatigue force. For these standards the wheel diameter needs to be below $1,250mm$ and the wheel Ultimate strength needs to be above $500N/mm^2$

$$F_{mean} = \frac{F_{minA,B} + 2F_{maxA,B}}{3} \tag{2.19}$$

This method does not calculate an allowed force for the rails. However, it does provide a minimum ultimate strength f_u for the rail from figure 2.5

2.3.1 Static force

The maximum allowed static force is given by:

$$F_{static} \leq 1.9 P_L b_{eff} D_{eq} \tag{2.20}$$

with:

- P_L is a limiting pressure dependent upon the ultimate strength of the metal used for the wheel (f_u(MPa)) as shown in 2.5.
- b_{eff} is the effective width (mm);
- D_{eq} is the equivalent wheel diameter(mm)

Ultimate strength of metal used for rail wheel f_u N/mm²	P_L N/mm²	Minimum ultimate strength for rail N/mm²
> 500	5,00	350
> 600	5,60	350
> 700	6,50	510
> 800	7,20	510
> 900	7,80	600
> 1 000	8,50	700

Figure 2.5: Values of PL

2.3.2 Fatigue force

The fatigue force is given by:

$$F_{fatigue} \leq P_L b_{eff} D_{eq} c_1 c_2 \tag{2.21}$$

with the factors c_1 and c_2. c_1 depends on the rotation speed of the wheel in RPM according to figure 2.6.

Wheel rotation speed r/min	c_1	Wheel rotation speed r/min	c_1	Wheel rotation speed r/min	c_1
200	0,66	50	0,94	16	1,09
160	0,72	45	0,96	14	1,10
125	0,77	40	0,97	12,5	1,11
112	0,79	35,5	0,99	11,2	1,12
100	0,82	31,5	1	10	1,13
90	0,84	28	1,02	8	1,14
80	0,87	25	1,03	6,3	1,15
71	0,89	22,4	1,04	5	1,16
63	0,91	20	1,06		
56	0,92	18	1,07		

Figure 2.6: Values of c_1

c_2 depends on the Group classification of the mechanism, as shown in table 2.1. The classification of mechanism depends on T, which depends on the total operation time of the crane given in figure 2.2, and L, which depends on the Force spectrum factor k_c given in figure 2.3. The total operation time depends on the rotation speed of the wheel and the amount of rotations and the force spectrum factor depends on the Force and rotations (see figure 2.15)

Spectrum	Utilization Class									
Class	T0	T1	T2	T3	T4	T5	T6	T7	T8	T9
L1	1.25	1.25	1.25	1.25	1.12	1.12	1	0.9	0.8	0.8
L2	1.25	1.25	1.25	1.12	1.12	1	0.9	0.8	0.8	0.8
L3	1.25	1.25	1.12	1.12	1	0.9	0.8	0.8	0.8	0.8
L4	1.25	1.12	1.12	1	0.9	0.8	0.8	0.8	0.8	0.8

Table 2.1: values of c_2

Utilization class	total duration of use T(h)		
T0	0	$\leq$T$\leq$	200
T1	200	<T$\leq$	400
T2	400	<T$\leq$	800
T3	800	<T$\leq$	1600
T4	1600	<T$\leq$	3200
T5	3200	<T$\leq$	6300
T6	6300	<T$\leq$	12,500
T7	12,500	<T$\leq$	25,000
T8	25,000	<T$\leq$	50,000
T9	50,000	<T$\leq$	∞

Table 2.2: Values of T

Spectrum class	Spectrum factor k_c(-)		
L1	0	$< k_c \leq$	0.125
L2	0.125	$< k_c \leq$	0.250
L3	0.250	$< k_c \leq$	0.500
L4	0.500	$< k_c \leq$	1.000

Table 2.3: Values of k_c

3|Wheel Fatigue

The three methods describe an allowed force for a given amount of rotations. However for real application the wheel is subjected to a range of different forces active for different amounts of rotations. E.g. different crane loading and weather. In this case one cannot just consider each load case separately, as fatigue can cause nonlethal damage after each separate load case that causes failure after a number of load cases. In the current sheet this effect is not taken into account, with disastrous consequences.

In the new drive sheet a universal method for determining fatigue of multiple load cases based on a force and an amount of rotations is required. This can be done using Miner's rule.

3.1 Miner's rule

The Miner's rule is a method for determining fatigue damage. It states that:

$$C = \sum_{i=1}^{k} \frac{n_i}{N(F_i)} \tag{3.1}$$

with:

- C is the lifetime consumed by the load cases
- i is the index of the load cases and k is the total amount of load cases
- n_i is the amount of rotations of the current load case
- $N(F_i)$ is the total amount of rotations allowed for the current load case as a function of the Force of the current load case

The load case the user has to fill in consists of a force and an amount of rotations that force is active for. Based on the force, the maximum amount of rotations is determined and with this the consumed lifetime (C) is found. If the sum of C for all load cases exceeds 1 failure occurs. The total lifetime of the system is found by dividing the time the load cases take by the Miner's ratio C.
The maximum amount of allowed rotations as a function of the force for the different standards can be found by rewriting the Allowed force equation. This is explained below.

3.1.1 Horrowitz

for Horrowitz the Allowed stress is given as a function of the amount of rotations:

$$\sigma_{allowed,i} = (\max(0.812 - 0.07\log(n_i), 0.27))10HB \tag{3.2}$$

For finding the maximum allowed rotations the minimum stribeck value (0.27) is omitted. The allowed stress is the Hertze stress:

$$\sigma_{Hertze,i} = 0.591\sqrt{\frac{F_i E}{bD}} \tag{3.3}$$

Filling in this stress and rewriting to the amount of rotations yields:

$$N(F_i) = 10^{\left(-0.591\sqrt{\frac{F_i E}{0.49HB^2 bD}} + \frac{0.812}{0.07}\right)} \tag{3.4}$$

3.1.2 EN 13001-3-3

For EN 13001-3-3 the allowed fatigue force is:

$$F_{fatigue,i} = \frac{F_{u,i}}{\gamma_{cf} \sqrt[m]{\frac{n_i}{i_D}} k_{c,i}} f_f \tag{3.5}$$

Since the amount of rotations for one specific load case (F_i) is needed, the force spectrum value is always 1(as all rotations are with the same force). Rewriting the force equation to the amount of rotations yields:

$$N(F_i) = \frac{\left(\frac{A^2 \pi D b(1-v^2) f f}{E_m \gamma_{cf} F_i}\right)^m}{k_{c,i}} i_D \tag{3.6}$$

Where A is the part that depends on hardening. Since the Miner's rule is used instead of the Force spectrum, k_c is 1 for each load case. This reduces the equation to:

$$N(F_i) = \left(\frac{A^2 \pi D b(1-v^2)}{E_m \gamma_{cf} F_i} f_f\right)^m i_D \tag{3.7}$$

Rotations static load case

It was already shown that for low rotations the allowed fatigue force is higher than the allowed static force. This means that this standard considers a force that is active for less than a certain amount of rotations as a static force. With the equation for the amount of rotations it is possible to find the amount of rotation for the allowed static force($N(F_s)$). This is done by filling in the allowed static force (equation 2.6) in equation B.31. This yields, after rewriting:

$$N(F_s) = \left(\frac{A_f^2}{A_s^2} \frac{f_{f3} f_{f4}}{f_2}\right)^m i_D \tag{3.8}$$

With A_f and A_s are the hardening factor for the equivalent fatigue force and the static force. This ratio is always constant independent of the hardening:

$$\frac{A_f^2}{A_s^2} = \frac{(3HB)^2}{(7HB)^2} = \frac{(1.8\sigma_y)^2}{(4.2\sigma_y)^2} \approx 0.18 \tag{3.9}$$

with this $N(F_s)$ reduces to:

$$N(F_s) = \left(0.18 \frac{f_{f3} f_{f4}}{f_2}\right)^m i_D \tag{3.10}$$

This amount of rotations is used to find a comparable allowed 'static' force for Horrowitz.

3.1.3 ISO/FEM

For ISO/FEM finding the allowed rotations is much more difficult to determine, as it is not directly found in the force equation:

$$F_{fatigue} \leq P_L b D c_1 c_2 \tag{3.11}$$

The amount of rotations does depend on c_2, Finding the allowed c_2 yields the allowed amount of rotations. This is done by first determining each possible value of c_2 based on the Force spectrum factor of the load cases. Then calculating the allowed fatigue force for each value of c_2 and comparing these results to the mean force of the load cases. The first allowed fatigue force that is higher than the mean force is taken. The Operation class of this force can now be translated to the allowed rotations with the rotation speed of the wheels for the load cases.

Rotations static load case

Just like done for EN13001-3-3, the amount of rotations for the static load case can be determined. To find the static rotations the Fatigue force is equal to the Static force, so the influence factor c_1 times c_2 has to be equal to 1.9. The c_1 for 0 RPM (static) can be extrapolated from the data: 1.19. c_2 has to be equal to 1.6 to reach 1.19. Since the maximum c_2 is 1.25, the static rotations force cannot be reached: the static force is higher than the fatigue force for a stationary wheel. There is not an amount of static rotations for FEM/ISO.

4|Wheel Hardening Depth

When a wheel loaded, the maximum shear stresses do not occur at the wheel surface, but at a certain depth (z). To ensure that the Hardness of the material here suffices, the hardness at this depth has to be assured. That is why the depth to which a material is hardened needs to be specified. The wheel Standards give such a specification: EN13001-3-3 and Horrowitz base the Hardening depth on the depth of the maximum shear stress and ISO/FEM just assumes the depth as 1$\%$ of the Diameter. EN 13001-3-3 and Horrowitz make a distinction between point contact and line contact. Since a wheel on a rail will always reduce to a line contact during rolling of the wheel, point contact is discarded.

for Horrowitz, the hardening depth (δ) is given by

$$\delta = 4 * 0.78 \frac{\sigma_H(1-\upsilon^2)}{E_m} D_{eq} = 3.12(1-\nu^2)\sqrt{\frac{F D_{eq}}{\pi b_{eff} E_m}} = 3.12(1-\nu^2)\sqrt{\frac{1}{\pi}}\sqrt{\frac{F D_{eq}}{b_{eff} E_m}} \tag{4.1}$$

For EN13001 the hardening depth is defined as:

$$\delta = \sqrt{\frac{F \pi D_{eq}(1-\nu^2)}{b_{eff} E_m}} = \sqrt{(1-\nu^2)\pi}\sqrt{\frac{F D_{eq}}{b_{eff} E_m}} \tag{4.2}$$

with ν is 0.3 for steel :

$$3.12(1-\nu^2)\sqrt{\frac{1}{\pi}} \approx \sqrt{(1-\nu^2)\pi} \tag{4.3}$$

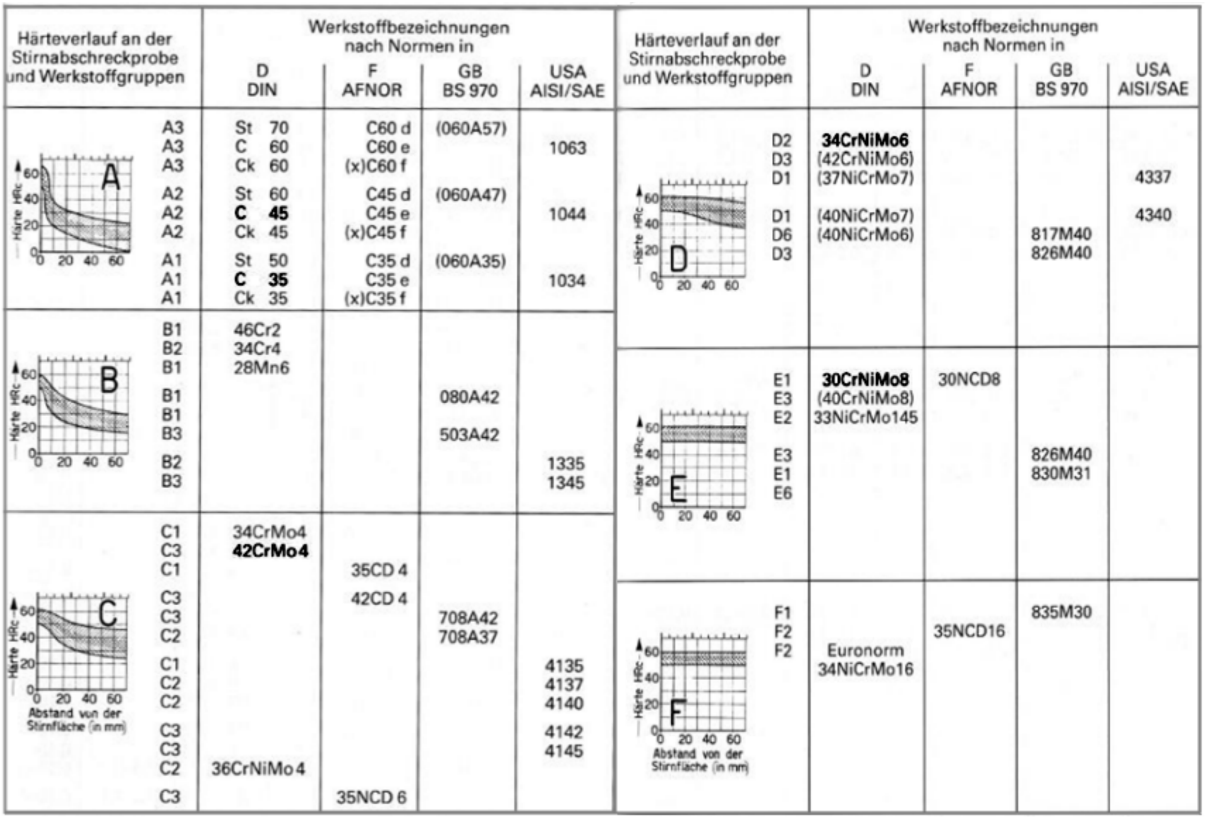
So the Hardening depth according to EN 13001-3-3 is almost equal to the hardening depth for Horrowitz.

4.1 Hardened Hardness

The hardened Hardness depends on the material that is used. Some materials are more fit for Hardening than others. An overview of hardenability of materials is given in The Hardening EWI. For example Stainless steel is not suitable for most hardening methods. Hardening methods include:

- Induction hardening
- Quenching followed by tempering
- Thermochemical hardening
- Laser hardening
- Flame hardening

The hardened Hardness of a number of Hardenable materials is given in figure 4.1 based on Maag Taschenbuch 1985. The material commonly used by Huisman and contained in the material list are bold. The values are based on Jominy tests, where a bar is heated to around 1000 $^{\circ}C$ and then cooled from the bottom to be able to get the Hardness as a function of the depth of the material. Graphs A and B represent normalized steels. C to F represent quenched and tempered steels. The numbers indicate where the Hardness is most likely to occur: In the lower (1), middle (2) or upper(3) range of the indicated values. Some materials may result in values over the full range(6). Using Excel's Trendline function the graphs have been translated into equations of the hardened Hardness as a function of the hardening depth. It should be noted that in most cases only the wheel is hardened. As the wheel is often made of material with high Chromium content, They are good for hardening and the hardness varies only little along the thickness.



Härteverlauf an der Stirnabschreckprobe und Werkstoffgruppen	Werkstoffbezeichnungen nach Normen in D DIN	F AFNOR	GB BS 970	USA AISI/SAE
A3	St 70	C60 d	(060A57)	
A3	C 60	C60 e		1063
A3	Ck 60	(x)C60 f		
A2	St 60	C45 d	(060A47)	
A2	**C 45**	C45 e		1044
A2	Ck 45	(x)C45 f		
A1	St 50	C35 d	(060A35)	
A1	**C 35**	C35 e		1034
A1	Ck 35	(x)C35 f		
B1	46Cr2			
B2	34Cr4			
B1	28Mn6			
B1			080A42	
B1				
B3			503A42	
B2				1335
B3				1345
C1	34CrMo4			
C3	**42CrMo4**			
C1		35CD 4		
C3		42CD 4		
C3			708A42	
C2			708A37	
C1				4135
C2				4137
C2				4140
C3				4142
C3				4145
C2	36CrNiMo4			
C3		35NCD 6		

Härteverlauf an der Stirnabschreckprobe und Werkstoffgruppen	Werkstoffbezeichnungen nach Normen in D DIN	F AFNOR	GB BS 970	USA AISI/SAE
D2	**34CrNiMo6**			
D3	(42CrNiMo6)			
D1	(37NiCrMo7)			4337
D1	(40NiCrMo7)			4340
D6	(40NiCrMo6)		817M40	
D3			826M40	
E1	**30CrNiMo8**	30NCD8		
E3	(40CrNiMo8)			
E2	33NiCrMo145			
E3			826M40	
E1			830M31	
E6				
F1			835M30	
F2		35NCD16		
F2	Euronorm 34NiCrMo16			

Figure 4.1: Hardened Hardening from Jominy test

For the materials that are not included in this overview, the hardened Hardness is given as the minimum required hardened Hardness according to EN 13001-3-3:

$$HB_{hardened} = 0.6\sigma_y \tag{4.4}$$

This Hardness is very conservative and should be replaced by actual tested values. As briefly mentioned before, the Hardness of the material plays a huge role in the Fatigue criteria of the wheels. This is why it is so important to get a good overview of the achievable Hardness. As for the decision in EN 13001-3-3 to use the yield stress instead of the Hardened hardness; this assumption makes EN 13001-3-3 very conservative for Hardened materials compared to Horrowitz. However neglecting this decision and using the Hardness in every situation means that the standard is violated. Even with conclusive proof that using the yield stress is incorrect it will not be changed.

For the material that are unsuitable for hardening the Hardened Hardness is assumed equal to the Unhardened Hardness It is possible for the user of the drive sheet to fill in a custom Hardened Hardness. This should be done with care as the fatigue conditions depend heavily on the Hardness of the material. As mentioned before, to prevent pitting the difference in Hardness between the wheel and the rail should be at least 100 N/mm^2.

5|Wheel Worksheet Implementation

The two standards, EN 13001 and FEM/ISO, have been added to the existing IMEP wheel worksheet in Excel. To test whether this is done correctly results from the sheet have been validated hand calculations. Furthermore the validity of the Standards in general has been measured via a comparison to wheel fatigue tests done by Huisman. These comparisons can be found in Appendix B

5.1 Old worksheet

Figure 5.1 shows the interface of the old IMEP wheel worksheet. On the left top, the user can fill in the parameters and choose a contact situation: Wheel on wheel, wheel on surface, wheel in wheel and ball on surface. This choice influences which parameters need to be filled in e.g. for ball on surface the diameter of the counter surface and the width of the wheel are not required. The equivalent diameter, which is used as the diameter for calculations, also differs for the contact situations.

This diameter, as well as the Stribeck value and the allowed force is displayed below the input cells.

Below these calculated value cells there is another input range for Forces. The result cells next to these cells show the Hertz stress and hardening depth for this force. Furthermore it shows whether this force is below the allowed force. Next to the result is a guideline for approximating the hardness of a material.

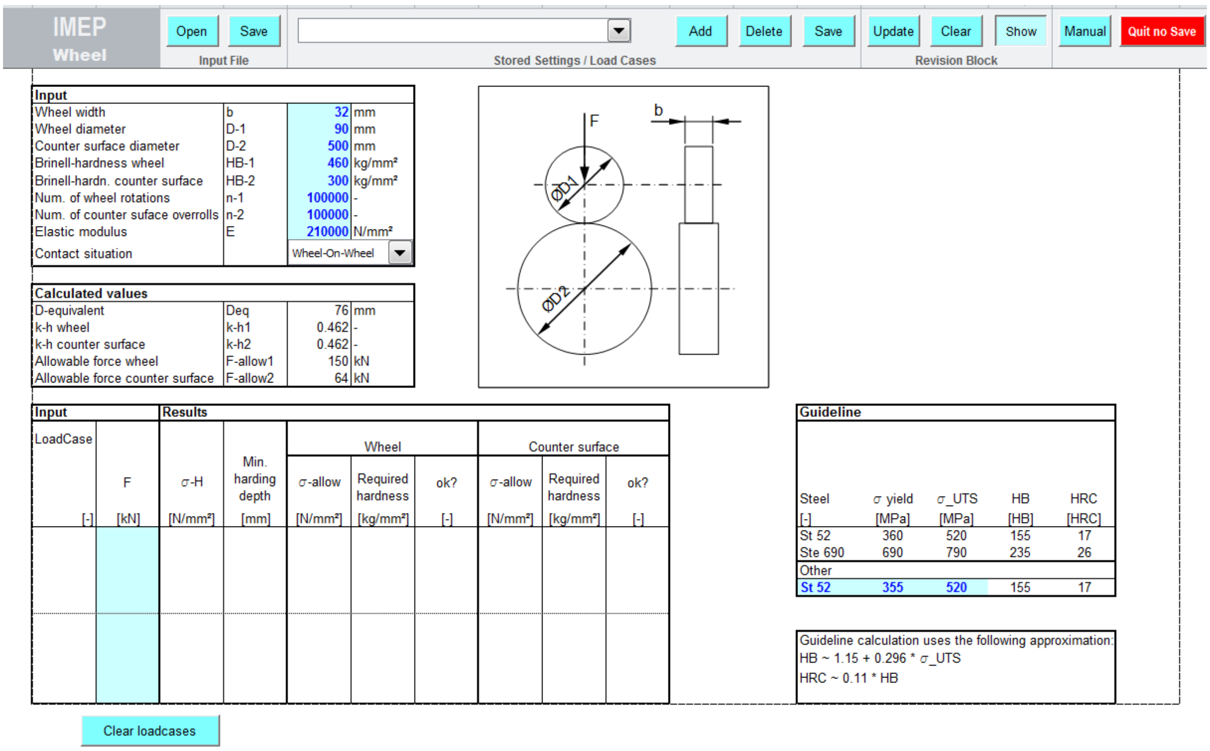


Figure 5.1: Interface of the old worksheet

5.2 New worksheet

The new worksheet is vastly different from the old one. First of all two standards; EN 13001-3-3 and ISO/FEM, are added. The user can choose which standard to use in the top drop down. Based on this choice, the required inputs are determined. Figure @@@ shows the input area for the different standards. If the Horrowitz method is selected, the contact method can be specified, just like for the old worksheet. EN and ISO/FEM only applies to Wheel on rail contact. another big change to the old worksheet is the load case input section. In the old worksheet the load cases were all separate load cases which did not result in any useful results apart from the hardening depth. The new worksheet does combine the load cases and bases the total lifetime used on it.

5.3 User interface

The user interface consists of four input areas: The standard, the geometry, the material and the External influences, which are mainly only used for EN 13001-3-3.

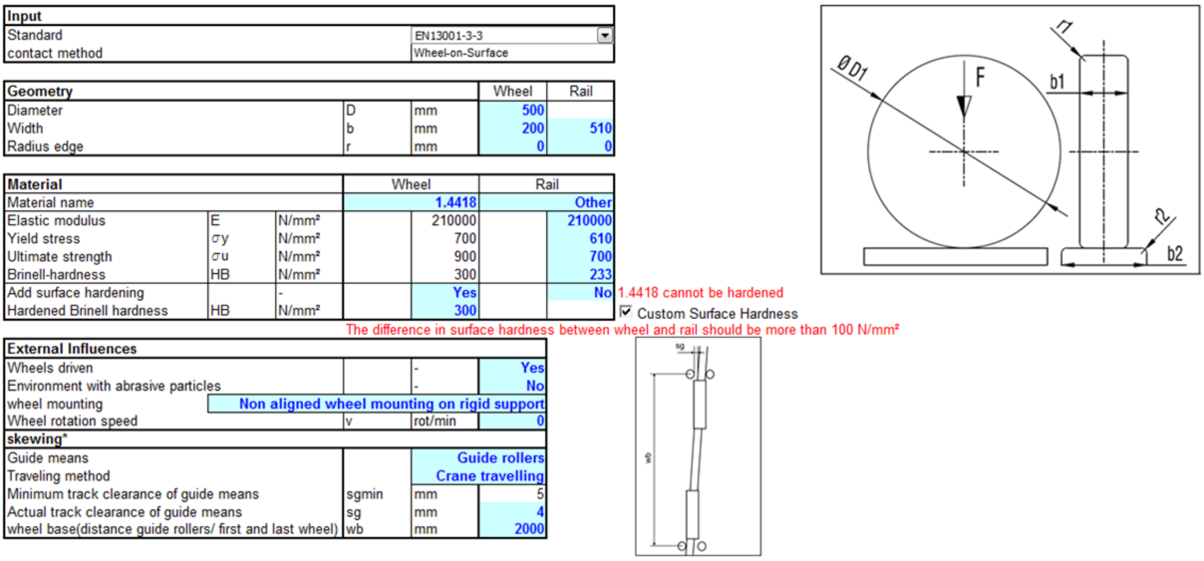


Figure 5.2: User Interface

Standard

The standard input contains a drop down menu to switch between the standards. The user can also choose to show all standards. If the Horrowitz method is selected, the user can also specify the contact method. If another method is selected, the Horrowitz contact method is set to Wheel on surface, since this is the contact method of the other standards.

Geometry

The Geometry contains the Diameter (D), width (b) and edge radius(r) for the Wheel and the rail and the ratio between Rail over rolls and Wheel rotations. The Diameter of the rail is only relevant if the rail does not have a (nearly) infinite diameter, so for Wheel on Wheel and Wheel in Wheel contact. If this is not the case it is made blank

Material

For the material choice the IMEP material database is used, along with Hardox material, which is specifically designed for high fatigue applications and has a very high Hardness. Furthermore the user can fill in another material by choosing `Other`. When choosing this the characteristics of the previously selected material remain as visible, but can be changed. If the user wants to harden the material, the hardness of the hardened material is required. The user can choose to use the hardness as defined in chapter 4 or put in a self defined hardness. An overview of the available material and their characteristics is given in figure 5.3. The hardness values are those of the hardened materials. The values with a yellow background are those that are determined by the Jominy test as discussed in chapter 4. For the Hardox material, not all parameters are available. The unknown σ_u are estimated based on the Hardness of the material.

Rail	Wheel	σy	σu	HB (h)
Hardox Extreme	Hardox Extreme	1400	1950	840
Hardox 600	Hardox 600	1400	1800	840
Hardox 550	Hardox 550	1400	1700	840
Hardox 500	Hardox 500	1250	1500	750
Hardox 450	Hardox 450	1100	1350	660
Hardox 400	Hardox 400	1000	1250	600
1.4418	1.4418	700	900	300
30CrNiMo8	30CrNiMo8	723	898	455
34CrNiMo6	34CrNiMo6	616	814	503
42CrMo4	42CrMo4	489	742	555
StE690	StE690	610	700	366
StE690-Forging	StE690-Forging	610	700	366
C45	C45	313	587	529
StE460-Profile	StE460-Profile	376	515	226
GS20Mn5	GS20Mn5	320	512	192
C35	C35	269	509	455
StE460-Plate	StE460-Plate	393	501	236
St52-3N-Forging	Other	295	470	177
S355-HP		295	450	177
St52-3N		285	450	171
S355-HST		290	442	174
St37-2		185	340	111
Other				

Figure 5.3: Material list

5.3.1 External influences

This input section mainly contains input for the EN 13001-3-3 influence factors. For FEM, the only external influence is the crane speed and for Horrowitz none of the external influences are required. That is why this section is (partly) hidden if FEM or Horrowitz is selected. If the 'show all standards' button is clicked the external influences will always be shown.

5.4 Calculations

To get an overview of the calculations done in the drive sheet The 'show calculations' button as shown in figure 5.4 can be clicked. This shows the constant values used for calculations, as well as intermediate answers , the influence factors and the static allowed force and rotations for each standard. Like the external influences only the results that are relevant to the current standard are shown and clicking the 'show all standards' button shows the values for all standards. For EN 13001-3-3, the skewing angle needs to be below 0.015 rad. If it is not this cell will be red.

5.5 Load cases

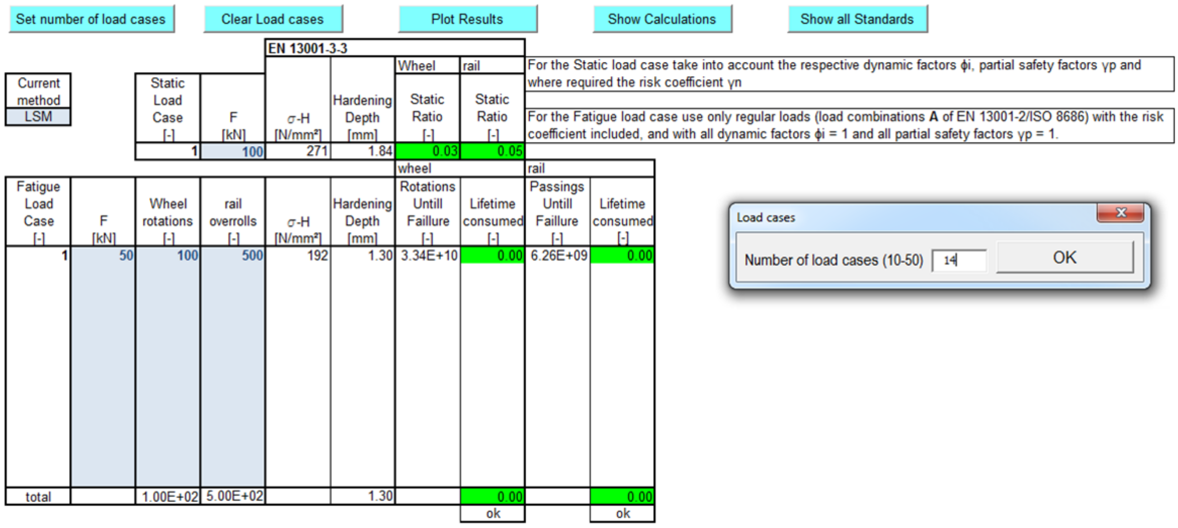


Figure 5.4: Results

In this section the Load cases are declared. As FEM/ISO and Horrowitz and EN13001-3-3 yield different results, the load case section looks different for the standards. Again, when the 'show all standards' button is clicked, all results are shown. The buttons are equal for all standards:

- Amount of load cases triggers a form where the user can declare the required amount of load cases. The load cases that are not filled in do not influence the program; reducing the amount of load cases only contributes to the compactness of the sheet.
- Clear load cases does what it says
- Plot results plots the results as described in chapter 5.6.
- Show calculations shows the calculations as described in chapter 5.4.

In the first input row the maximum static load case is required. This Force is the maximum occurring force in the life of the wheel/rail and thus includes exceptional loads, which the fatigue load cases below do not include. For that reason the static force is at least equal to the maximum fatigue force. As the static force is static, it does not contribute to the consumed lifetime. Only static failure can occur here.

5.5.1 Horrowitz/EN13001-3-3

For EN 13001-3-3 the load case input is shown in figure 5.4. The user fills in a Force and an amount of rotations this Force is active for. After that the results are given:

- Rail over rolls, which is related to the wheel rotations by the ratio declared in the geometry input
- The Hertz stress, which is given by equation 2.3
- The hardening depth, which is based on equation 4.1 for Horrowitz and equation 4.2 for EN 13001-3-3
- The static ratio, which is the ratio between the allowed static force and the load case force. Where the allowed static force is given by equation 2.1 for Horrowitz and equation 2.6 for EN 13001-3-3
- The Maximum allowed rotations, given by equation 3.4 for Horrowitz and equation B.31 for EN 13001-3-3

- The consumed lifetime, which is given by equation 3.1

Note that if either the Force or the rotations is not filled in, not all equations can be calculated. The results are limited to those that can be calculated with the current input.
The most important row of the results is the bottom row. Here the total results are shown. For the Hardening depth the maximum value is shown, for the static ratio the minimum value is shown and for the rotations, over rolls and consumed lifetime the sum is shown. If the values are within the allowed limit, their cells will be shown in green, if not they are red. Only when this row is green the wheel/rail will fulfill the required load cases without failure.

5.5.2 FEM/ISO

The FEM Load case is shown in figure 5.5. As FEM is does not calculate each load case individually, the results are a lot more compact. The load case input is equal to the other standards. The results include:

- The Hertz stress, which is given by equation 2.3
- The hardening depth, which is 1% of the wheel diameter
- The Mean fatigue force, which is given by 2.19
- The force spectrum factor, which is given by equation 2.15
- The static ratio, which is the ratio between the allowed static force and the load case force. Where the allowed static force is given by equation 2.20
- The Maximum allowed rotations, which is shown in Chapter3.1.3
- The consumed lifetime, which is given by equation 3.1
- The minimum required σ_u, which is shown in figure 2.5

Next to the results the Allowed fatigue force for each Utilization class is shown. If the allowed fatigue force is higher than the mean force the corresponding rotations is allowed. The current Utilization class is shown in orange.

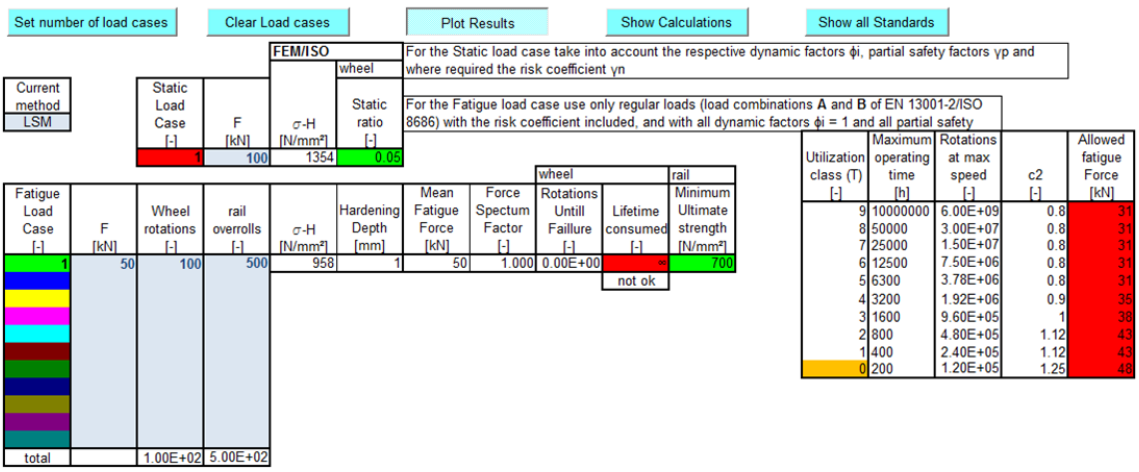


Figure 5.5: Load cases and results for FEM

5.6 Plot results

When the plot results button is clicked, the allowed force for the current standard is plotted against the amount of rotations/over rolls for the wheel and the rail. While the rotations are below the static rotations the static allowed rotations are shown. If the 'show all standards' button is clicked, all standards are plotted. The load cases are also plotted in these graphs as dots.

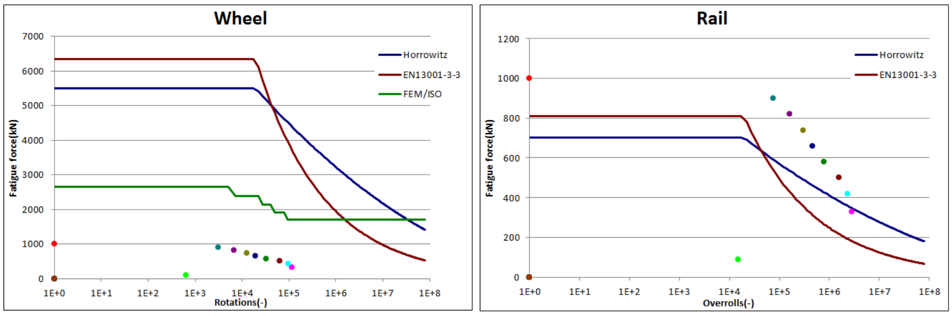


Figure 5.6: Plotted load cases

5.7 Manual and EWI

Next to the excel sheet, the manual for this datasheet has been completely rewritten to match the new sheet. Furthermore the Work instruction (EWI) for wheels has been improved by addition of the new standards and Fatigue calculations.

6|Sections Optimization

The second part of this internship focuses on investigating the possibility of adding An optimization tool to the IMEP program: sections. In this program users can declare cross-sections of beam sections of a number of standard profile types e.g.: I-profile, Pipe or box. The program determines whether the section is sufficient to withstand the defined load cases. The objective of the optimization is to find the geometry that withstands these load cases with the smallest surface area, and thus the smallest mass.

To achieve this, first the program is discussed, next the optimization methods are discussed and lastly the implementation of an optimizer in IMEP sections is discussed.

6.1 IMEP sections Program

With IMEP sections a user can create profiles for beams and find the stress that occur in this profile for a certain load case. The profiles are a combination of plate profiles, as these are easy to work with when determining inertia and stresses. The program has six profile types to choose from, but the user could create any combination of plates. An example of a programmed profile is the I-profile, which is a combination of two horizontal plates and one vertical plate. Below the different sheets in the program are discussed

6.1.1 Geometry

The main sheet of IMEP sections is the geometry sheet. Here the user can create the profile. Clicking one of the profile options buttons pops up a form. Here the user can fill in the desired dimensions of the chosen profile or pick one of the standard profiles.This profile is added to the geometry as a series of plates at a certain position. The program supports up to 200 plates, so it is possible to add different profiles to the geometry. The profile is plotted next to the geometry input. Furthermore the user can choose the material of the profile. This will influence the stresses in the profile, which is discussed later

6.1.2 Forces

In the Forces sheet the user can declare load cases. The user can add forces and moments to the load case and add a safety factor for the specific load case.

6.1.3 Stresses

In this sheet the stresses resulting from the load cases as declared in the forces sheet on the geometry declared in the geometry sheet are calculated. The sheet gives a stress safety factor as the ratio between this von Misses stress and the yield stress. The stress constraint of the optimization tool is based on this factor.

7|Sections Optimization Strategy

The objective of the optimization is to minimize the mass of a beam, while fulfilling the constraints opposed by the yield stress.The optimization looks as follows:

- Objective
  - Minimize Surface Area: A(h, w, t_f, t_w)
- Variables
  - h, Profile height
  - w, Profile width
  - t_f, Profile Thickness vertical plates
  - t_w, Profile Thickness horizontal plates
  - x_{cog}, x centre of gravity
  - y_{cog}, y centre of gravity
- Constraints
  - Stress safety factor
  - Boundary conditions
    - * Maximum and minimum allowed value for variables

The main criteria this profile has to fulfill is the Stress safety factor, which was already discussed in chapter 6. This safety factor depends on the method declared in the forces sheet. When the Limit State Method (ASM) is used, the safety factor should be :

$$s_f = \frac{\sigma_y}{\sigma_{vonM}(h, w, t_f, t_w, x_{cog}, y_{cog})} \geq sf_{req} \tag{7.1}$$

with σ_y the yield stress and σ_{vonM} the von Mises stress and sf_{req} the user defined allowed safety factor. When the limit stress method (LSM) is used, the safety factor should be lower than the inverse of the ASM safety factor.

Next to the stress, the available area can be a limiting factor. In application, the profile has to be placed in a construction, meaning there is a finite space to place it. That is why the user can define the box in which the profile needs to fit. This is done by setting the x-and y- limits of the entire profile. The user can also define a maximum profile thickness and the program defines a minimum profile width, height and thickness to make sure the profile remains. If for example the height of the i-profile is lower than the thickness of the two horizontal plates the vertical plate need to have a negative length. To prevent this the minimum height is set to three times the thickness of the vertical plate. It is also possible to manually put in the minimum dimensions, but these are overruled if they are below the program defined minimum dimensions.

Two solving methods are used: One that searches the best profile from a predefined set of standard profiles, Standard Profile Search, and one that uses a solving engine present in excel to optimize geometry parameters of the given profiles, The excel solver. For the Standard profile search the COG position is not variable. These Methods are described below.

7.1 Standard Profile search

The first optimization method used is a discrete method. It cycles through the array of standard profiles and chooses the one that fulfills the constraints with the lowest area. Although this does not yield the absolute optimum, it does yield the best standard profile and this is often more desired than the absolute optimum, as this profile is always available and does not require fabrication, often weighing up to the costs saved when using a lighter profile. There are four profiles in IMEP Sections that have standard profiles declared: I-, RHS- UNP- and PIPE-Profiles. If the user requires another profile, first a set of standard profiles will have to be defined.

7.1.1 Input

The Standard profile search requires a bit of user input to channel its search. First of all the standard profiles types and the materials to search through. In general the user will already have an idea of which profiles to use based on the load cases. The search time can be reduced by limiting the amount of profiles to choose from. The material choice influences the results of the optimizer. First of all the material influences the Yield stress. Furthermore available standard profiles differ among materials. For example IPE profiles are only available for St.37 and St.52. If these materials are not desired for whatever reason the IPE profile will be excluded from the search. For a fair comparison between profiles of different materials an optimization to the lowest price instead of the lowest area would be better. Lastly the position and orientation of the profile is required as input. The search will not optimize this, as it would require for the search to be repeated until this optimal is found, which takes a long time. Furthermore the position of the profile is often defined depending on other aspects and the orientation can easily be determined by the user based on the load cases.

Search strategy

It is possible to cycle through all available profiles to find the best, however this method is very time consuming. To speed up the search a more directed search strategy is desired. This method determines the if the profile should be larger or smaller than the previous found profile and chooses a new profile based on this.

IPE BEAM [D0000017]							
SIZE	H	B	Tw	Tf	R	A [mm2]	M [kg/m]
80	80	46	3.8	5.2	5	764	6.00
100	100	55	4.1	5.7	7	1030	8.09
120	120	64	4.4	6.3	7	1320	10.36
140	140	73	4.7	6.9	7	1640	12.87
160	160	82	5	7.4	9	2010	15.78
180	180	91	5.3	8	9	2390	18.76
200	200	100	5.6	8.5	12	2850	22.37
220	220	110	5.9	9.2	12	3340	26.22
240	240	120	6.2	9.8	15	3910	30.69
270	270	135	6.6	10.2	15	4590	36.03
300	300	150	7.1	10.7	15	5380	42.23
330	330	160	7.5	11.5	18	6260	49.14
360	360	170	8	12.7	18	7270	57.07
400	400	180	8.6	13.5	21	8450	66.33
450	450	190	9.4	14.6	21	9880	77.56
500	500	200	10.2	16	21	11550	90.67
600	600	220	12	19	24	15600	122.46

Figure 7.1: IPE standard profiles

For a given profile (e.g. I-profile) the standard profiles are saved in a database (figure 7.1) ordered from small to large area. Due to this the stress occurring in the profiles for any given load case can be assumed to be ordered from high to low. This ordering is useful for an effective search. As results for a profile can give an indication in which direction to search. The search is started at the profile in the middle of this database, at search index $m = m_{tot}/2$.

This profile is filled in into the Sections sheet, which yields a stress safety factor, based on the defined load cases. This safety factor is compared to the allowed safety factor:

- If the safety factor is lower than the allowed safety factor, the stress needs to be decreased and m is increased
- If the safety factor are higher than the allowed safety factor, the stress could be increased and m is decreased

The amount of change in m is equal to the rounded off ratio between the yield stress and the current stress as long as the new m is within the range of the profiles. This way m changes more if the current profile is far from the optimal solution.

this procedure is repeated until m changes direction and dm of the previous iteration is 1. This means that the desired safety factor is passed. dm needs to be 1 if it more than 1 the optimum may lay between the current and the last profile. In that case the script is not stopped. In the example the procedure is repeated twice. after the first iteration the safety factors are are still not met, so m is increased again. Since dm is now 1 the new m is 10. after the second iteration the safety factors are satisfied, so the optimum is passed. Since the previous dm is 1 the optimum is now found. The profile that meets this safety factor with the lowest area is deemed the optimum standard profile.

7.2 Excel solver

A different method for optimization is to variate dimensions of a given profile instead of comparing standard profiles. This is done with a solving algorithm. Excel has such an algorithm, called the 'solver'. This solver uses an optimization engine to find the minimum surface area of a selected cross-section by changing a defined set of variables while fulfilling a set of defined constraints.

7.2.1 Input

The input for the solver is slightly different from that of the Standard profile search. First of all the solver needs boundary conditions and initial values for the variables. Also the solver will require a profile to optimize, as it cannot determine this itself. It would be possible to let the solver optimize each possible profile for the given boundary conditions, but this would be very time consuming and often unnecessary. Like the profile, the material has to be defined as well.

7.2.2 Search Engine

The solver has three search engines the user can choose from, ordered from fast to slow:

- Simplex LP
- GRG-Nonlinear
- Evolutionary

Simplex LP

The reason Simplex LP is the fastest is that it is a linear solver. As the optimization problem is nonlinear, this solver is not usable. This leaves only the slower Non-linear solvers.

GRG nonlinear

GRG stands for Generalized reduced Gradient. This method uses the gradient of the area to find out how the variables should be changed to achieve the optimal area. When the gradient is zero the optimum is found. If the function that should be solved is smooth, this solver finds the optimal solution very fast. The downside of this method is that the solver will stop at any optimum it finds. It is possible that the solver converges to an optimum that is not the global optimum but the local optimum.

Evolutionary

This method optimizes based on the principle of natural selection. The solver takes a population of random initial configurations and selects the configuration for which the solution is closest to the optimal solution. Next a set of mutations of this profile are compared and again the best solution is selected. This process is repeated until there is only very little difference between the mutated profiles. Because this method uses a wide range of initial conditions, the change that the optimum is in fact the global optimum is very large. However for this method to work, the optimal area needs to be known, which is not the case. This makes this method unusable as well.

Multistart

It can be concluded that the only usable method is GRG, however the fact that local optima can be found is a big problem. This problem is averted by adding multistart. The GRG will repeat the optimization process a couple of time with different (pseudo-random) initial conditions to make sure the different optima are found and the correct optimum is used. The downside of this method is that it takes a longer to find a solution, depending on the population size The two optimization methods are added as separate forms in the IMEP sections sheet. The input of the optimized profiles is comparable to the input of a user defined profile.

7.2.3 Plate thickness

The plate thickness (t_f and t_w) found by the solver are never standard plate thicknesses, so in order to produce the optimized profile a custom plate has to be produced. This is very costly and should be prevented. To make sure the optimized profile has a standard thickness, the thicknesses resulting from the solver are adjusted to the nearest higher standard plate thickness. With this standard thickness the solver is ran again without the thickness as a variable, yielding the optimized profile for this standard thickness.

7.3 Multiple profiles

In some situations two beams parallel beams are used in construction. It is possible to optimize such a geometry. However a couple of problems can occur that the optimization tool does not automatically take into account. The main problem for the solver is relative sizing. If all dimensions of all profiles are variable. The optimized geometry is often one very small and one very large profile. This problem is easily solved by declaring relations between the dimension of the different profiles. The initial size ratio between profiles is kept. This reduces the amount of variables to those of one profile.

A more complex problem is intersection between profiles. This problem occurs for both the standard profile search and the Solver. Below an explanation of this problem as well as a solution to the problem is given.

7.3.1 Profile intersection

When two profiles intersect, the intersecting areas will contain double mass here and will be used for calculations twice. This situation is realistically impossible and thus these results are incorrect. That is why this situation should be avoided.

This can be done two ways. One way is checking for intersection and reject results that intersect. Another way is making sure no intersection can occur by declaring relations between the position of profiles. e.g. specifying the distance between the edges of the profiles. These relations are described by a pattern. For now two Patterns have been declared: Line pattern and Circle pattern.

The two methods for preventing intersections are described below.

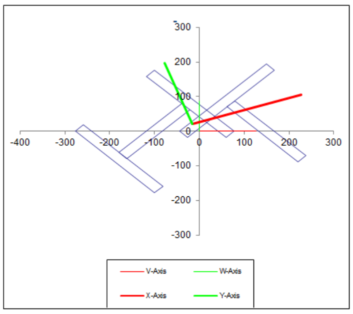


Figure 7.2: Intersecting profiles

7.3.2 Checking for intersection

For checking intersection the lines that make up the blocks of the profiles are checked individually. Each of these lines can be described as a linear equation. For each line in the profile the intersection point with each line in each other profile is determined. If this intersection point lies on the line for which intersection is checked the two lines intersect. Given the four corner points of line A and B: $x_{A1}, y_{A1}, x_{A2}, y_{A2}$, the linear function describing these lines, $y = ax + b$, can be found:

$$y_A = \frac{dy}{dx}_A (x_A - x_{A1}) + y_{A1} \tag{7.2}$$

$$y_A = \frac{dy}{dx}_B (x_B - x_{B1}) + y_{B1} \tag{7.3}$$

$$\tag{7.4}$$

with:

$$dx_A = (x_{A2} - x_{A1}) \tag{7.5}$$

$$dy_A = (y_{A2} - y_{A1}) \tag{7.6}$$

The intersection point is the point where $y_A = y_B$:

$$\left(\frac{dy}{dx}\right)_A (x_{int} - x_{A1}) + y_{A1} = \left(\frac{dy}{dx}\right)_B (x_{int} - x_{B1}) + y_{B1} \tag{7.7}$$

This yields a function for x_{int}:

$$x_{int} = \frac{x_{A1}\left(\frac{dy}{dx}\right)_A - y_{A1} - x_{B1}\left(\frac{dy}{dx}\right)_B + y_{B1}}{\left(\frac{dy}{dx}\right)_A - \left(\frac{dy}{dx}\right)_B} \tag{7.8}$$

$$x_{int} = \frac{(y_{B1} - y_{A1})dx_A dx_B + x_{A1} dy_A dx_B - x_{A1} dy_B dx_A}{dy_A dx_B - dy_B dx_A} \tag{7.9}$$

When x_{int} is known, y_{int} can be found by filling it in either of the line equations:

$$y_{int} = \frac{dy}{dx}_A (x_{int} - x_{A1}) + y_{A1} \tag{7.10}$$

There are three situation where the method for determining line intersection is slightly different: parallel lines, a horizontal line and a vertical line. For parallel line no intersection occurs, so it does not need to be checked. If one of the lines is a horizontal line the equation reduces. If line B is horizontal, it is given by $y_B = y_{B1}$:

$$\left(\frac{dy}{dx}\right)_A (x_{int} - x_{A1}) + y_{A1} = y_{B1} \tag{7.11}$$

$$x_{int} = (y_{B1} - y_{A1}) \frac{dx_A}{dy_A} + x_{A1} \tag{7.12}$$

and $y_{int} = y_B$

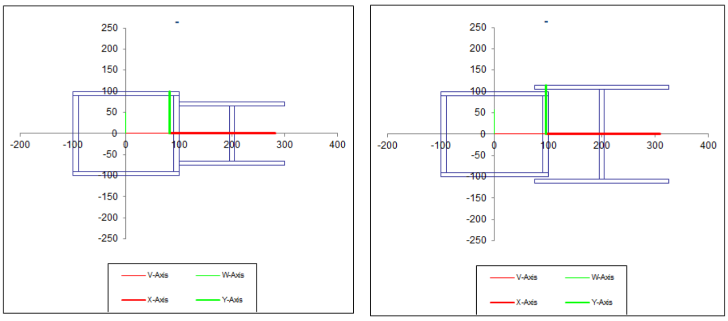
If line B is a vertical line, $x_{int} = x_{B1}$ and y_int is:

$$y_{int} = \left(\frac{dy}{dx}\right)_A (x_{B1} - x_{A1}) + y_{A1} \tag{7.13}$$

Intersection occurs if the intersection point found is between A_1, A_2, B_1 and B_2.

Having the solver check for Intersection may give problems as it depends on the variables discretely: changing the variables a little bit often does not influence the amount of intersections. The amount of intersections does not translate to specific requirements for the variables. As a result the solver will get stuck and assume the optimum is found when profiles intersect. An example of this problem is shown in figure 7.3a. The optimizer requires a higher width for the I-profile on the right. As this will cause intersection it is not allowed and the solver assumes this is the best possible solution. If the solver would first increase the height as shown in figure 7.3b this would not directly yield a better result, but it would allow for a consecutive increase in width without intersection, yielding a better solution. This kind of problem solving could in theory be implemented in the solver, but it requires a lot of code to take special situations like this one into account.

Using Multistart does offer a bit of a solution to this problem by approaching the intersection point from different initial positions. However it would be best if intersection could be prevented instead of counting the amount of intersections. This is done by defining relations between the positions of profiles with patterns.



(a) Optimizer gets stuck at intersection

(b) Solution to prevent intersection

Figure 7.3: Optimizer Intersection

7.3.3 Preventing Intersection

As mentioned intersection can be prevented by declaring relations between profile positions. Two patterns have been defined: Line and Circle Patterns.

Line

For the line pattern all profiles are placed in a line with a certain step size between the COG's. When the first profile moves, the other profiles move accordingly. However there is still a possibility for intersection if the distance between the profiles is too small. To prevent this the step size can be declared as the distance between the edges of the profiles. Figure 7.4 shows this difference. The step size between the COG is d_1 and the distance between the edges of the profiles is d_2. When d_2 is given by the user, d_1 is declared as:

$$d_1 = d_2 + \frac{w_1 + w_2}{2} \tag{7.14}$$

where d_2 is the user defined distance between the profiles and w is the width of profiles 1 and 2. It should be noted that the user needs to take into account the painting of the profiles when declaring d_2: If the distance is zero the touching sides do not need to be painted, otherwise the distance has to be at least 50mm to make sure a painting brush can reach between the profiles.

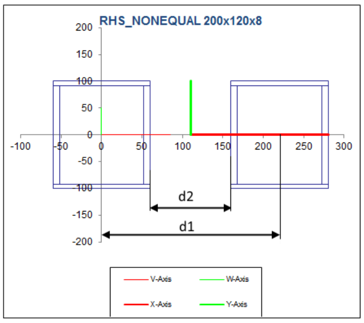


Figure 7.4: Stepsize

This is the most simple example of a line pattern. the line contact becomes more complex when the profiles are rotated, as shown in figure 7.5a. In this case the distance d_1 is:

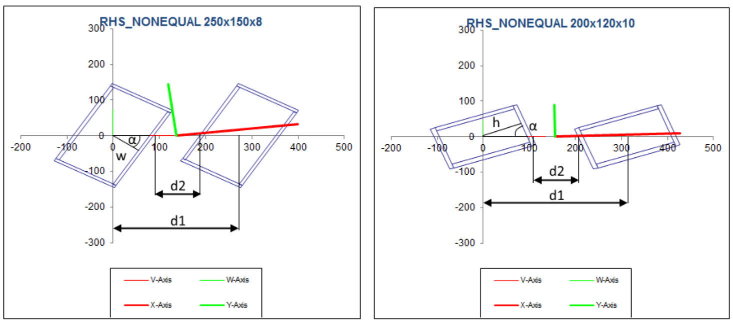
$$d_1 = d_2 + \frac{w_1 + w_2}{2\cos(\alpha)} \tag{7.15}$$

However, if the angle α is large, the profiles will not be neighbouring in width direction but in height direction, as figure 7.5b shows for an angle of $70°$. In this case the distance d_1 is given by:

$$d_1 = d_2 + \frac{h_1 + h_2}{2\sin(\alpha)} \tag{7.16}$$

Whether one has to use the distance definition based on the width or on the height depends one whether the corner point of the profile is above or under the line between the two COG's. Besides this the definition that should be used is always the one that yields the smallest distance:

$$d_1 = d_2 + \min\left(\frac{h_1 + h_2}{2\sin(\alpha)}, \frac{w_1 + w_2}{2\cos(\alpha)}\right) \tag{7.17}$$



(a) $\alpha = 30°$

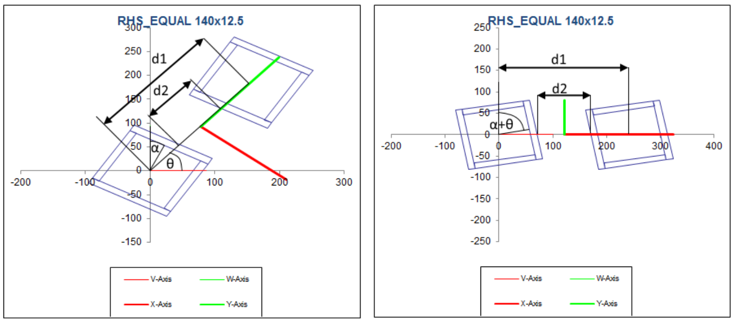
(b) $\alpha = 70°$

Furthermore it is possible to change the direction of the line pattern, given as an angle θ. This is shown in figure 7.6a. This angle influences the step size, as well as the direction. The step size can be determined by adding θ to α, yielding the situation in figure 7.6b. Step size d_1 is now

$$d_1 = d_2 + \min\left(\frac{h_1 + h_2}{2\sin(\alpha + \theta)}, \frac{w_1 + w_2}{2\cos(\alpha + \theta)}\right) \tag{7.18}$$

for the x and y position of profile 2 this step size is used:

$$x_2 = x_1 + d_1 \cos(\theta) \tag{7.19}$$
$$y_2 = y_1 + d_1 \sin(\theta) \tag{7.20}$$



(a) $\alpha = 30°$ and $\theta = 50°$ (b) $\alpha = 80°$ and $\theta = 0°$

Figure 7.6: Stepsize for different direction

Circle

The circle pattern places the number of required profiles in a circle, as shown in figure 7.7. For the circle the user defines the position of the first profile (x_p, y_p), as well as the center of the Circle (x_c, y_c). In this case the position of the first profile is (100, 100) and the center of the circle is (0, 0) The Radius of the circle is defined as:

$$r(x, y) = \sqrt{(x_p - x_c)^2 + (y_p - y_c)^2} \tag{7.21}$$

the angle of the first profile (θ_0) relative to the positive x axis of the center of the circle is given as:

$$\theta_0 = \arctan(\frac{y_p - y_c}{x_p - x_c}) \tag{7.22}$$

The angle of the other profile is given by:

$$\theta_n = 2\pi \frac{n-1}{n_{tot}} + \theta_0 \tag{7.23}$$

With n the index of the number of profiles. With each θ_n known, the position of each profile as a function of the position of the first profile can be determined the same way as the Line contact.

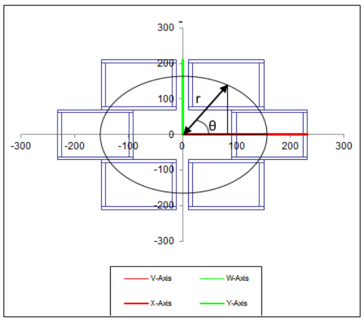


Figure 7.7: Circle pattern

8|Sections Otimization Results

To test the quality and comparability of the Standard Profile Search and the Solver, an I-profile is optimized. This is done for a single profile first. In this test the problems that occur while optimizing are discussed. Next the test is done for multiple profiles with different contact methods to investigate the effects of increasing the complexity of the problem as well as checking the solutions proposed in Chapter 7.

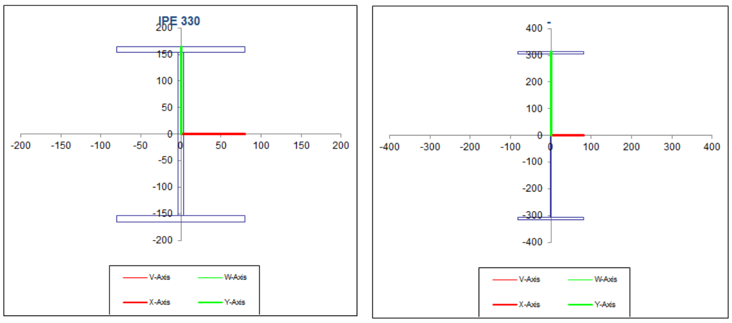
The optimization is done for a single load case of a moment around the x-axis of 200kNm and a force in y-direction of 100kN. The required safety factor is 1.18

8.1 Single profile

For a single I-profile the optimized standard profile and solver profile are given here.

8.1.1 Standard Profile search

There are three types of I-profile: IPE, HEA, HEB and HEM. Where the different HE are variations in flange size. The search is done for all types and all available material separately. The search takes 6.4 seconds to find the optimal profile, which is IPE 360 with St.52-3N material. This profile can be seen in figure 8.1a and has a surface area of $6995mm^2$ and the safety factor is 1.47. To check if this is indeed the best profile, the search has been done without IPE profiles. This time the best profile is HEA 260, which ha a surface area of $8188mm^2$, so higher than the previous optimal. This would indicate that The optimal profile is indeed IPE 360. To compare with other IPE profiles: one profile smaller, IPE 330, has a safety factor of 1.16 and thus does not suffice. This gives the conclusion that IPE 360 is indeed the Optimal standard Profile.



(a) Standard Profile Search (b) Solution solver

Figure 8.1: Optimized geometry for a single I-profile

8.1.2 Solver

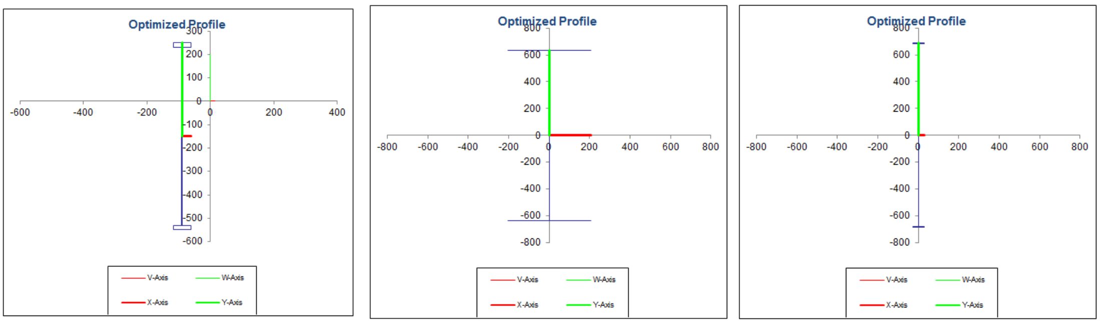
In order to compare the results of the solver with the results of the Standard Profile Search, the material for the profile is St.52-3N, the same as the results for the Standard Profile Search. The boundary and initial conditions are :

	Max	Min	Initial
h	1500	1	100
w	500	1	100
t_f	50	1	10
t_w	50	1	10

Table 8.1: Boundary- and initial conditions

The test is done with the GRG solver, both with and without multistart. First the optimziation is done without multistart. The Optimzied profile is shown in figure 8.1b. As the moment is only around the x-axis, it is expected that the optimal is a very slender profile, and this is exactly what happened: The height is at the maximum, as this adds the most inertia per added surface, furthermore the thickness of the vertical plate(t_f) barely adds to the inertia, so it is minimized. Furhtermore the position of the profile remained at the origin. The reason for this is that changing it did not directly yield a change in result.

Next Multistart is used with 10 subproblems. The solver found a better profile than it found without multistart, as shown in figure 8.2a. This means that the initial conditions yield a local optimum instead of a global optimum and proofs the usefulness of multistart at the cost of taking more time. The position of the profile is also not at the origin. The reason for this is that this solver used random initial conditions with different origins.



(a) Multistart (b) Initial conditions (c) Boundary conditions

Figure 8.2: Improved solver for I-profile

Changing the initial conditions to the expected values yields an even better results. This time the initial height is taken maximal and the other constraint are taken minimum, as the solution is expected to be. The result is shown in figure 8.2b. The thickness remained minimal. This is the reason that the surface area is even lower than before. The fact that an even better solution is found than with multistart indicates that the amount of sub problems was not large enough to find the solution as given for these initial conditions.

Lastly the boundary conditions are changed to be closer to the optimal solution. The minimal height is increased and the maximum values of the other dimensions are decreased. This forces the solver to converge to the global optimum. The resulting surface area is almost equal to the surface area for improved initial conditions, but the dimensions are quite different.

All results have been summarized in table 8.2

Method	Standard Profile	GRG	GRG multistart	GRG Initial	GRG Boundary
Safety factor(-)	1.47	1.18	1.18	1.18	1.18
Surface Area(mm^2)	6995	4047	2992	2085	2044
time(s)	6.4	15.3	163.5	4.4	5.2

Table 8.2: Results for I-Profile

Based on the surface area it is clear that the solver yields a much better result. This is mainly due to the fact that the thickness is a lot lower for the solver as buckling is not taken into account. For standard profiles the thickness is chosen such that buckling will not occur. Apart from that the standard profile is a little bit oversized, as the safety factor is quite a lot higher than the allowed safety factor, while the solver results are all exactly at the allowed safety factor. This contributes to the fact that the solver yields a better result. Based on time the results differ. If the initial values or boundary conditions are near the optimal values the solver finds the solution very fast, otherwise the solver is a bit slower than the Standard Profile search. If multistart is used the Solver is very slow. Although the difference in solving time is not a huge problem now, increasing the amount of profiles will increase solving time dramatically.

The fact that different settings yield such a vastly different result makes the solver's results very questionable. As it is so dependant on the initial conditions one can never be sure if the results can be trusted. There are a few ways this problem could be tackled:

- Choose initial conditions that are near the optimal solution This requires approximate knowledge of the optimal solution.
- Decrease the search area i.e. decrease the difference between the max an minimum allowed values. This also requires knowledge of the approximate location of the optimum.
- Increase the amount of subproblems to make sure the initial conditions that yield the best solution are found. This increases the calculation time a lot

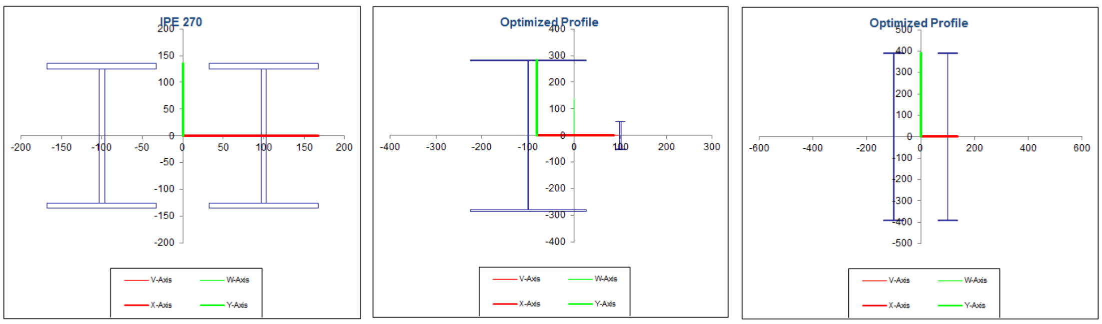
the conclusion is that the Solver yields the most optimized solution. But the solver is not robust at all. Next to this the solution is not checked for buckling. If the Solver can be made more robust and if a buckling check can be added, the solver will yield a better result, but for now the better solution is the one found by the Standard Profile Search.

8.2 Multiple profiles

To illustrate the problems that occur when optimizing multiple profiles, two I-profiles are placed horizontally at a distance of 100mm. The profile boundary conditions and load cases are equal to those used for a single profile above.

8.2.1 Standard Profile Search

As the distance between the profiles is limited to 200mm, profiles with a width larger than 200mm are excluded from the search. The search does find a solution: IPE 270, as shown in figure 8.3a. Because the validity of the search is already checked for one profile and still holds here, this check is not repeated.



(a) Standard Profile search (b) Solver (c) Solver with size ratio

Figure 8.3: Optimization for multiple I-profiles

Because this search excluded some profiles that might yield a better solution. The test is repeated with an distance of 60mm between the edges of the profiles. This yields the same profile.

8.2.2 Solver

Next the solver is used to optimize two I-profiles. First two I-profiles with no relation whatsoever are used. This should result in the best possible combination of the profiles. The Intersection check is done to prevent profile intersection. The result is shown in figure 8.3b. The result shows one profile much smaller than the other. That is why the constant size constraint is added.

With the size ratio constant, the solver found the relation as shown in figure 8.3c. The two profiles are now the same size. As the amount of variables is reduced, the solving time is also reduced, but the surface are is increased.

Next a line contact with a distance between profiles of 60mm is used to ensure the distance between profiles and prevent intersection. However as intersection was not an issue the results are almost the same. The optimization time is very high, even though the amount of variables is reduced. This might be due to the increased complexity of the problem.

The results are shown in table 8.3

Method	Standard Profile	GRG	GRG size ratio	GRG size ratio and distance
Safety factor(-)	1.41	1.18	1.18	1.18
Surface Area(mm^2)	8803	2494	2904	2750
time(s)	11.36	29.3	24.8	37.07

Table 8.3: Results for two I-Profiles

8.2.3 Conclusion

The standard profile search performed just like it did for one profile. The difference being that it was a little bit slower now because the sheet has to load in more plates than for one profile. The solver on the other hand performed a lot worse than for one profile: The increase of variables caused an increase in Local optima and solving time. Adding constraints to the profiles did reduce the amount of variables and yielded a more useful geometry. The conclusion from the test with one profile can be extended to multiple profiles. With the added argument for using Standard profile search that the optimization time increases less for multiple profiles.

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[2] FEM 1.001: Rules for the design of hoisting appliances, booklet 4: Checking for Fatigue and choice of mechanism components, *FEM*, 1998

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A|Employer and Reflection

A.1 Employer description

Huisman is a large international company, specialized in large steel constructions. The core business has been the design and development of off-shore crane installations for a wide range of application. Next to cranes Huisman also produces Pipe laying installations and drilling equipment for the offshore oil industry. However since the devaluation of the oil price Huisman has opted a shift towards other area's both on- as well as offshore including Geothermal drilling, offshore wind energy installation and amusement rides. With this shift Huisman tries to survive in a sector that is in though weather, with varying success.

A.2 Reflection of Internship period

As Huisman is a large company, it has a large R&D department. There is a special team dedicated to IMEP consisting of 5 people. I have been part of this team for 4 months. Each member of this team has its own tasks and thus there is limited direct collaboration in the team. There are, however, weekly meetings where every member presents his progress during that week to the team leader. This meeting gives the opportunity for consultation and deciding how to carry on with the tasks. The meetings were very useful for those reasons.

A.2.1 Planning

The IMEP team works with the sprint planning technique. At the start of each 5 week sprint the team discusses what each member will have done at the end of the sprint. This requires a certain degree of prior knowledge on the time certain tasks take. This was something that I misjudged largely. The work on IMEP wheel took a lot longer than i initially expected: It could not be done in one sprint. This showed me the difficulty in planning without experience in the task to do. The assignment took longer partly due to a shift in demands from the client during the process, which resulted in me having to rewrite parts of the program. Furthermore the time spend on troubleshooting was something I underestimated.

A.2.2 Difficulty

The difficulty of the assignments was reasonably high. Translating the Wheel standards to a universal Fatigue criteria was rather challenging, but with the help of colleagues the solution turned out to be rather easy and straight forward. Also the implementation into excel prompted a number of challenges that seemed large at first, bu turned out to be solvable with relative ease. The fact that I had no prior experience with excel did increase the challenge, as I did not know the best way to approach some situations. E.g. I created code for actions that could be done with standard excel functions. This did cause some extra time to be spend on implementation. For the section optimization coming up with a strategy for optimization did not take much time and again most time was spend on implementation in excel. Also adding exceptions for specific situations took a lot of time. In conclusion the most challenging part of the internship lied not in solving Engineering related problems but rather programming problems.

A.2.3 Atmosphere

As mentioned I worked in a team of 5, where each member has its own tasks. This reflects in the working climate. Everyone works on his own task and does not really bother others. Sometimes there is consultation between colleagues or non work related conversations, but the most time it was quite quiet. Although not very sociable, this was a good working environment for me to get work done without interruptions.

A.2.4 Conclusion

In conclusion I think I have gotten quite a lot done in my time at Huisman. I could have done more with better communication with the client on the specification of the requirements or feedback on the work done so far. The overall appreciation of my work seemed high.

B|Validation Datasheet

To check the correctness of the Worksheet it is compared with hand calculations. This is done below. Furthermore the standards are compared with a series of wheel on rail tests done at Huisman.

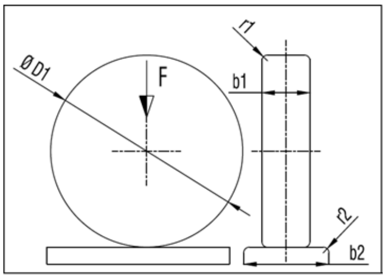
B.1 Hand calculations

To prevent errors in the excel script, a hand calculation is done for all three methods. The input for these calculations is based on The failed Basked rails and is shown in figure B.1. The load cases for this basket are shown in figure B.2.

Input	
Standard	FEM 1.001 b4 / ISO 16881-1
contact method	Wheel-on-Surface

Show all Standards

Geometry			Wheel	Rail	
Diameter	D	mm	500		
Width	b	mm	200	510	
Radius edge	r	mm	8	0	
Overrolls per Rotation	o/r	-	1	24	24.00



Material			Wheel	Rail
Material name			30CrNiMo8	Other
Elastic modulus	E	N/mm²	210000	210000
Yield stress	σy	N/mm²	723	234
Ultimate strength	σu	N/mm²	898	478
Brinell-hardness	HB	N/mm²	299	146
Add surface hardening		-	Yes	No
Hardened Brinell hardness	HB	N/mm²	409	

☑ Custom Surface Hardness

External Influences			
Wheels driven		-	Yes
Environment with abrasive particles		-	No
wheel mounting			Non aligned wheel mounting on rigid support
Wheel rotation speed	v	rot/min	0
skewing*			
Guide means			Guide rollers
Traveling method			Crane travelling
Minimum track clearance of guide means	sgmin	mm	5
Actual track clearance of guide means	sg	mm	4
wheel base(distance guide rollers/ first and last wheel)	wb	mm	2000

*based on ISO 8686

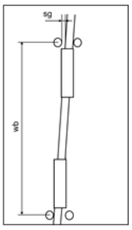


Figure B.1: Input for calculations

LoadCase [-]	F** [kN]	Wheel rotations [-]	rail overrolls [-]
static			
Fatigue 1	90	634	15216
2	170	148774	3570576
3	250	132935	3190440
4	330	117095	2810280
5	420	95975	2303400
6	500	64296	1543104
7	580	32616	782784
8	660	19416	465984
9	740	12628	303072
10	820	6688	160512
11	900	3168	76032
total		6.34E+05	1.52E+07

Figure B.2: Load cases for the basket

B.1.1 Allowed static Force

For the static force the amount of rotations used for Horrowitz depends on the safety factors of EN 13001-3-3, as described in Chapter 3.1.2. The safety factors f_{f1}, f_{f2}, f_{f3} and f_{f4} are calculated first:

f_{f1} depends on the non-contacting width and the radius of the wheel. Since the non-contacting width is large, the ratio is below the minimum and f_{f1} is minimal:

$$w = \frac{1}{2}(b_2 - b_{eff}) = \frac{1}{2}(500 - 184) = 158 \tag{B.1}$$

$$\frac{r_1}{w} = 0.05 \rightarrow f_{f1} = 0.85 \tag{B.2}$$

f_{f2} depends on the rotations from the load case in figure B.2

$$s = N\frac{D_1\pi}{1e6} = 199km \rightarrow L = 3 \tag{B.3}$$

$$f_{f2} = 0.8 \tag{B.4}$$

f_{f3} depends on the skewing angle:

$$\alpha_g = \frac{sg_{min}}{w_b} = 0.003 \tag{B.5}$$

$$\alpha_t = 0.001 \tag{B.6}$$

$$\alpha_w = \frac{0.03b_2}{w_b} = 0.008 \tag{B.7}$$

$$\alpha = 0.012 \rightarrow f_{f3} = \sqrt[3]{\frac{0.005}{\alpha}} = 0.77 \tag{B.8}$$

f_{f4} is 1 since the wheel is not driven. The static amount of rotations is now:

$$N_s = \left(0.18\frac{f_{f3}f_{f4}}{f_2}\right)^m i_D = 19,455 \tag{B.9}$$

Horrowitz

The Stribeck value for the static loadcase (N_s) is:

$$k_h = \max(0.812 - 0.07\log(N_s), 0.27) = 0.51 \tag{B.10}$$

and the allowed static force for the wheel is:

$$F_s = \frac{b_{eff}D_1}{E_m}\left(\frac{k_h * 10 * HB_1}{0.591}\right)^2 = 5495kN \tag{B.11}$$

The maximum force from the load cases is 900 kN, so the static safety factor is:

$$s_f = \frac{5491}{900} = 6.11 \tag{B.12}$$

As the safety factor is above 1, the wheel will not fail statically. and for the rail:

$$F_s = \frac{b_{eff}D_1}{E_m}\left(\frac{k_h * 10 * HB_2}{0.591}\right)^2 = 705kN \tag{B.13}$$

With static safety factor:

$$s_f = \frac{705}{900} = 0.78 \tag{B.14}$$

So the rail will fail statically

EN 13001-3-3

The allowed static force is:

$$F_s = \frac{A^2}{\gamma_{cf}} \frac{\pi * D_1 * b_{eff}(1-\nu^2)}{e_m} f_{f1} f_{f2} \tag{B.15}$$

for the wheel the surface hardness is 409 N/mm^2, which is higher than 0.6 times the yield stress (723 N/mm^2), so A becomes:

$$A = 4.2\sigma_y = 2863N/mm^2 \tag{B.16}$$

so the static force is:

$$F_s = \frac{2863^2}{1.1} \frac{\pi * 500 * 184(1-0.3^2)}{210,000} 0.85 * 0.80 = 6346kN \tag{B.17}$$

With static safety factor

$$s_f = \frac{7142}{900} = 7.05 \tag{B.18}$$

So the Wheel will not fail statically. For the rail there is no surface hardening. The surface hardness is $149N/mm^2$, so A becomes:

$$A = 7HB_2 = 1022N/mm^2 \tag{B.19}$$

so the static force is:

$$F_s = \frac{1042^2}{1.1} \frac{\pi * 500 * 184(1-0.3^2)}{210,000} 0.85 * 0.80 = 809kN \tag{B.20}$$

With static safety factor

$$s_f = \frac{809}{900} = 0.90 \tag{B.21}$$

The rail will fail statically.

ISO/FEM

With an ultimate strength of 898 N/mm^2 the limiting pressure is 7.2. With this the allowed static force is:

$$F_s = 1.9P_L D_1(b_2 - 2 * r_2) = 7.2 * 500 * 510 = 3488kN \tag{B.22}$$

$$s_f = \frac{3762}{900} = 3.88 \tag{B.23}$$

So the wheel will not fail statically

B.1.2 Fatigue Force

The fatigue force is calculated only for the last load case shown in figure B.2, so F = 900kN and n=3168 for EN 13001-3-3 and Horrowitz. For FEM/ISO the entire load case spectrum is used.

Horrowitz

for the wheel

$$N_i = 10^{\left(-0.591\sqrt{\frac{F_i Em}{0.49HB_1{}^2 b_{eff} D_1}}+\frac{0.812}{0.07}\right)} = 10^{\left(-0.591\sqrt{\frac{900e3*210e3}{0.49*450^2*184*500}}+\frac{0.812}{0.07}\right)} = 10^{8.9} = 8.1e8 \quad \text{(B.24)}$$

Miner's factor C is:

$$C = \frac{3168}{8.1e8} = 3.9e-6 \quad \text{(B.25)}$$

So the Wheel survives this load case. For the rail:

$$N_i = 10^{\left(-0.591\sqrt{\frac{900e3*210e3}{0.49*146^2*184*500}}+\frac{0.812}{0.07}\right)} = 10^{3.3} = 2.0e3 \quad \text{(B.26)}$$

$$C = \frac{76,031}{2.0e3} = 37.11 \quad \text{(B.27)}$$

So the rail has failed for this load case

EN13001-3-3

The total safety factor is:

$$f_f = 0.85 * 0.75 * 1 = 0.64 \quad \text{(B.28)}$$

For the wheel

$$N_i = \left(\frac{A^2 \pi D_1 b_{eff}(1-v^2) ff}{E_m \gamma_{cf} F_i}\right)^m i_D = \frac{(1302)^2 \pi * 500 * 184 * (1-0.3^2) * 0.64}{900e3 * 210e3 * 1.1} = 1.83e7 \quad \text{(B.29)}$$

Miner's factor C is:

$$C = \frac{3168}{1.83e7} = 1.73e-4 \quad \text{(B.30)}$$

So the wheel survives this load case For Rail

$$N_i = \frac{(438)^2 \pi * 500 * 184 * (1-0.3^2) * 0.64}{900e3 * 210e3 * 1.1} = 1.28e4 \quad \text{(B.31)}$$

Miner's factor C is:

$$C = \frac{76,031}{1.5e4} = 5.92 \quad \text{(B.32)}$$

So the rail has failed for this load case

FEM/ISO

The speed is 9.5 RPM, so c_1 is 1.13. The mechanism group class is:

$$t = \frac{N}{v} = \frac{6.34e5}{9.5 * 60} = 1113h \rightarrow T = 3 \quad \text{(B.33)}$$

The spectrum group is:

$$k_c = \frac{1}{i_{tot}} \sum_{i=1}^{i_{tot}} \left(\frac{F_i}{F_{max}}\right)^m = 0.098 \rightarrow L = 1 \quad \text{(B.34)}$$

making c_2 1.25. The mean force is 630 kN. For the maximum amount of rotations(T=9), the allowed fatigue force is found to be:

$$F = P_L D_1 (b_2 - r_2) c_1 c_2 = 7.2 * 500 * 550 * 0.82 * 0.8 = 1299kN \quad \text{(B.35)}$$

As the allowed force for the maximum mechanism class is below the mean force the allowed rotations is infinite and the Miner's factor is 0.

For the rail only the minimum required Ultimate strength is given based on the limiting pressure. This is equal to 510 N/mm^2, which is above the ultimate strength of the rail.

B.1.3 Results drivesheet

The results from the drivesheet are shown in figure B.3. These results match the results from the calculations above. The results are plotted in graph B.4

Horrowitz						Wheel			rail		
LoadCase [-]	F** [kN]	Wheel rotations [-]	rail overrolls [-]	σ-H [N/mm²]	Hardening Depth [mm]	Static Safety Factor [-]	Max Rotations Allowed [-]	Lifetime consumed [-]	Static [-]	Max [-]	Lifetime consumed [-]
static											
Fatigue 1	90	634	15216	268	1.81	74.43	5.6.E+10	0.00	7.83	9.5.E+08	0.00
2	170	148774	3570576	368	2.49	39.40	2.7.E+10	0.00	4.15	9.9.E+07	0.04
3	250	132935	3190440	446	3.02	26.79	1.5.E+10	0.00	2.82	1.7.E+07	0.19
4	330	117095	2810280	513	3.47	20.30	9.4.E+09	0.00	2.14	3.8.E+06	0.74
5	420	95975	2303400	579	3.91	15.95	5.8.E+09	0.00	1.68	8.7.E+05	2.66
6	500	64296	1543104	631	4.27	13.40	3.9.E+09	0.00	1.41	2.6.E+05	5.84
7	580	32616	782784	680	4.60	11.55	2.8.E+09	0.00	1.22	8.8.E+04	8.86
8	660	19416	465984	725	4.90	10.15	2.0.E+09	0.00	1.07	3.2.E+04	14.66
9	740	12628	303072	768	5.19	9.05	1.5.E+09	0.00	0.95	1.2.E+04	24.96
10	820	6688	160512	809	5.47	8.17	1.1.E+09	0.00	0.86	4.9.E+03	32.89
11	900	3168	76032	847	5.73	7.44	8.1.E+08	0.00	0.78	2.0.E+03	37.11
total		6.34E+05	1.52E+07			7.44		0.00	0.78		127.93
						ok		ok	not ok		not ok

(a) Horrowitz

EN 13001-3-3						Wheel			rail		
LoadCase [-]	F** [kN]	Wheel rotations [-]	rail overrolls [-]	σ-H [N/mm²]	Hardening Depth [mm]	Static Safety Factor [-]	Max Rotations Allowed [-]	Lifetime consumed [-]	Static [-]	Max [-]	Lifetime consumed [-]
static											
Fatigue 1	90	634	15216	268	1.82	79.35	3.94E+10	0.00	8.99	2.77E+07	0.00
2	170	148774	3570576	368	2.51	42.01	4.73E+09	0.00	4.76	3.32E+06	1.07
3	250	132935	3190440	446	3.04	28.57	1.31E+09	0.00	3.23	9.19E+05	3.47
4	330	117095	2810280	513	3.49	21.64	5.18E+08	0.00	2.45	3.64E+05	7.72
5	420	95975	2303400	579	3.94	17.00	2.32E+08	0.00	1.93	1.63E+05	14.13
6	500	64296	1543104	631	4.30	14.28	1.30E+08	0.00	1.62	9.12E+04	16.93
7	580	32616	782784	680	4.63	12.31	7.91E+07	0.00	1.39	5.56E+04	14.08
8	660	19416	465984	725	4.94	10.82	5.14E+07	0.00	1.23	3.61E+04	12.90
9	740	12628	303072	768	5.23	9.65	3.51E+07	0.00	1.09	2.47E+04	12.28
10	820	6688	160512	809	5.51	8.71	2.49E+07	0.00	0.99	1.75E+04	9.16
11	900	3168	76032	847	5.77	7.94	1.83E+07	0.00	0.90	1.28E+04	5.92
total		6.34E+05	1.52E+07			7.94		0.00	0.90		97.67
						ok		ok	not ok		not ok

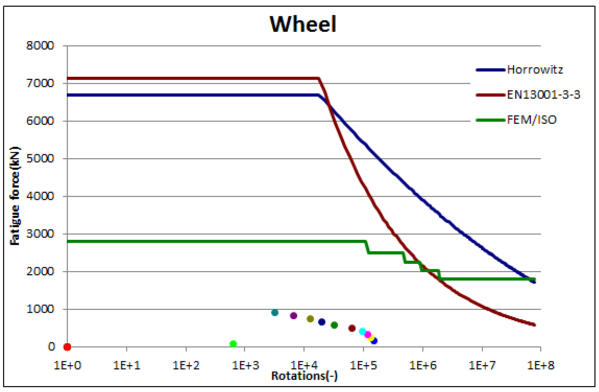
(b) EN13001-3-3

FEM/ISO								wheel			rail
LoadCase [-]	F** [kN]	Wheel rotations [-]	rail overrolls [-]	σ-H [N/mm²]	Hardening Depth [mm]	Mean Fatigue Force [kN]	Force Spectum Factor [-]	Static Safety Factor [-]	Max Rotations Allowed [-]	Lifetime consumed [-]	Minimum Ultimate strength [N/mm²]
static											510
Fatigue 1	90	634	15216	268	5	630	0.098	14.04	6.00E+09	0.000	
2	170	148774	3570576					ok		ok	
3	250	132935	3190440								
4	330	117095	2810280								
5	420	95975	2303400								
6	500	64296	1543104								
7	580	32616	782784								
8	660	19416	465984								
9	740	12628	303072								
10	820	6688	160512								
11	900	3168	76032								
total		6.34E+05	1.52E+07								

Utilization class (T) [-]	Maximum operating time [h]	Rotations at max speed [-]	c2 [-]	Allowed fatigue Force [kN]
9	10000000	6.00E+09	0.8	1790
8	50000	3.00E+07	0.8	1790
7	25000	1.50E+07	0.9	2014
6	12500	7.50E+06	1	2237
5	6300	3.78E+06	1.12	2506
4	3200	1.92E+06	1.12	2506
3	1600	9.60E+05	1.25	2797
2	800	4.80E+05	1.25	2797
1	400	2.40E+05	1.25	2797
0	200	1.20E+05	1.25	2797

(c) FEM/ISO

Figure B.3: Results Basket for the different standards



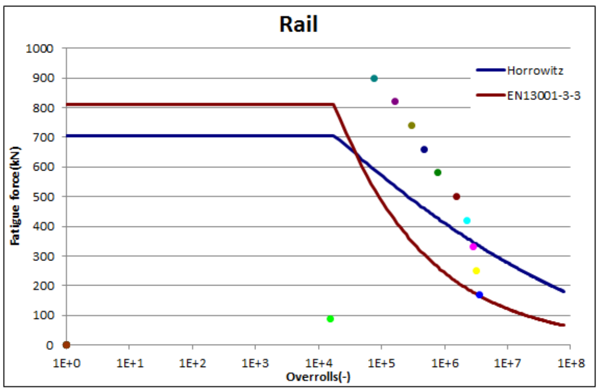


Figure B.4: Load cases Basket plotted for the different standards

B.1.4 Conclusion

The results match the results from the script, given some round of errors in the hand-calculations. From the calculations it becomes clear that for this case EN 13001 and Horrowitz yield a much higher allowed force than FEM/ISO. As EN takes the most variables into account, it could be said to be the most reliable. It seems that FEM/ISO takes these variable variables into account by assuming a worst case scenario, which could explain the large difference in results.

Failure does occur however for the rail for both EN and Horrowitz. First of all some of the forces exceed the allowed static force, which would already mean rail failure. If the rail would survive these static forces, then it will still fail due to fatigue: The Basket was designed to last 20 years, while the rail will only last around 1.8 months according to Horrowitz and 2.5 months according to EN 13001-3. When installed, this rail already showed large damage after around 2 months of service, so the lifetime expectation according to the standards turns out to be quite accurate.

B.2 Test results

The standards are now compared with measurements done at Huisman. Unfortunately not all tests were done for a high number of rotations, but the results still give some insight in which load cases should be allowed. If a standard does not allow a load case that has been proven not to fail, the standard is too conservative.

For the test, the input is as shown in figure B.5. The different tested materials and load cases are shown in table B.1. It should be noted that the hardness of the tested material differ from the prescribed hardness ($\frac{\sigma_{uts}}{3}$).

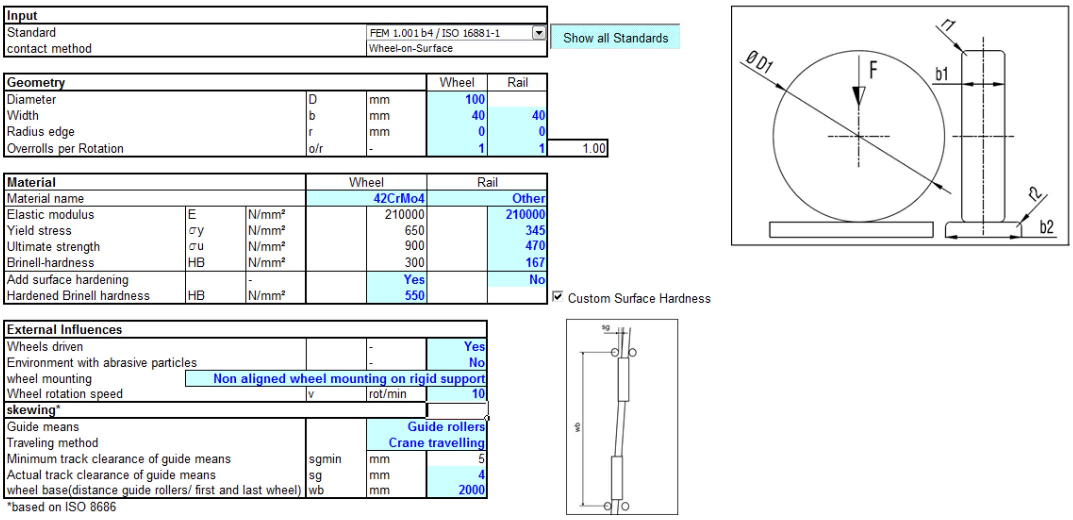


Figure B.5: Test variables

	Rail		Wheel
Material	St 52-3N	St.E690	42CrMo4
hardened	No	No	Yes
Yield stress(N/mm^2)	345	690	650
Hardness (N/mm^2)	167	248	550

Table B.1: Test materials and load cases

B.2.1 FEM allowed Force

For the test the maximum allowed wheel force is determined. The maximum allowed Hertz stress for the rail material without axial load is estimated using Finite Element Method(FEM (not to be confused with FEM standard)) analyses:

- St 52-3N : 1136MPa

- St.E690 : 1572MPa

This allowed Hertze stress can be translated in an allowed Force, which is compared to the standards yields a static factor in table B.2.

	St 52-3N	St.E690
F_{allow}(kN)	70	137
Horrowitz	2.07	1.84
EN 13001-3-3	1.48	1.31

Table B.2: Static failure ratio according to standards

The fact that static failure is predicted by the standards could indicate that the amount of static rotations is too high. As the amount of static rotations for Horrowitz is now based on EN 13001-3-3 static rotations, it would be useful to check the allowed rotations For the FEM force for Horrowitz. For St 52-3N this results in:

$$N_i = 10^{\left(\frac{-1136}{0.7*167}+\frac{0.812}{0.07}\right)} = 76 \tag{B.36}$$

For St.E690 the amount of static rotations based on FEM is:

$$N_i = 10^{\left(\frac{-1572}{0.7*248}+\frac{0.812}{0.07}\right)} = 295 \tag{B.37}$$

B.2.2 Tests

The FEM allowed forces were validated using actual tests. As an initial test (test 1), the allowed force is loaded on the wheel for 200,000 over rolls for 42CrMo4 wheels and St 52-3N is tested. The rail shows no damage after this load case.

Next the actual force at which damage occurs is investigated: a series of test above the maximum allowed force are done for 42CrMo4 wheels and St.E690 rail (test 2). It is found that at 138$\%$ load real deformation occurs.

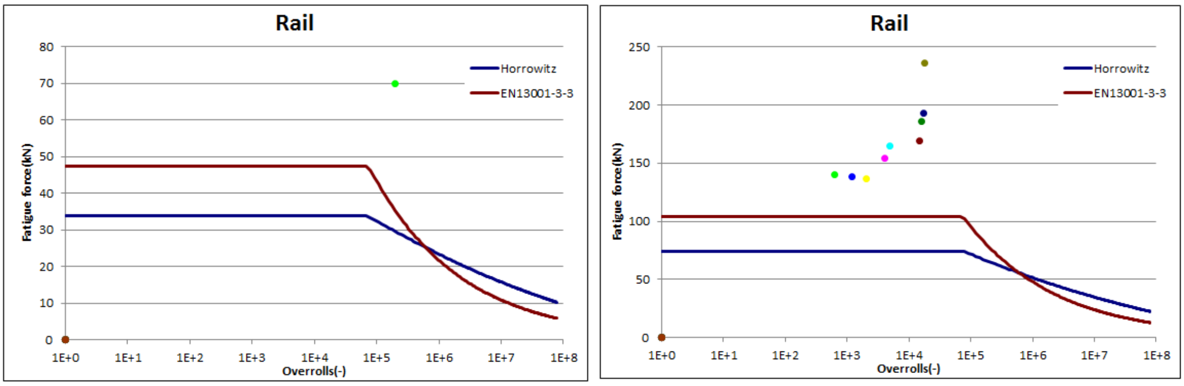
B.2.3 Results standards

The results according to the standards for both tests described above are shown in table B.3 and plotted in figure B.6. First of all the FEM/ISO shows failure for the wheel, while the other standards show barely any damage. The tests confirm the latter. The FEM/ISO results are very conservative.

As mentioned before, the allowed static force for the rail is much lower for the standards than it is for the FEM force. For for the first test the lifetime is consumed many times over. For the second test, where a lot less rotations have been performed, the life time resembles the test a lot more. Generally the EN 13001-3-3 results resemble the test results much more than Horrowitz, but they are both very conservative. Next the individual Load cases of test 2 are looked at individually to determine during which load case failure occurs.

Test	1		2	
Material	Wheel 42CrMo4	Rail St 52-3N	Wheel 42CrMo4	Rail St.E690
Horrowitz	0.0004	2470.4	0.02	48997.04
EN13001-3-3	0.03	9.94	0.42	8.65
FEM/ISO	∞	-	∞	-

Table B.3: Lifetime consumed for tests according to standards



(a) Test 1 (b) Test 2

Figure B.6: Rail Fatigue curve according to the standards

The lifetime consumed after the individual load cases for test 2 is shown in table B.4. According to the test real deformation (failure) occurs during the 138 $\%$ load case. According to Horrowitz it already occurs during the 118$\%$ load case and according to EN 13001-3-3 it occurs during the 133 $\%$ load case.

Load case($\%$)	Test deformation	Horrowitz	EN 13001-3-3
100	No	0.09	0.02
99	No	0.16	0.04
98	No	0.26	0.07
110	No	0.98	0.17
118	No	1.55	0.23
121	No	9.99	0.90
133	No	12.64	1.01
138	Yes	16.31	1.13
168	Yes	36.03	1.27

Table B.4: Total damage after each load case compared to damage test 2

One reason for the conservativeness of these results is the fact that the rail hardness increases as a result strain hardening caused by the over rollings. The Hardness increases nearly 100 N/mm^2. If instead of the initial hardness this increased hardness is used for the standards, Horrowitz will fail during the 138$\%$ load case and EN 13001-3-3 will not fail at all.The most realistic situation would be to use the average Hardness of all load cases, which is around 295. In this case Horrowitz fails during the 121$\%$ load case and EN 13001-3-3 does not fail. It can be concluded that if the initial hardness is used EN 13001-3-3 gives the most accurate results and if the hardened hardness is used Horrowitz does. It should be noted however that it is difficult to predict the increase in hardness caused by the load cases. Furthermore this effect is reduced drastically for surface hardened material, as the Hardness will increase a lot less for this case.

B.3 Conclusion

The main conclusion from the Static comparison between the standards and the test is that the Allowed static force given by the standards is very conservative. For EN 13001-3-3 and Horrowitz, the reason for that is that it is based on a high amount of rotations. For Horrowitz the amount of static rotations is now based on the amount of static rotations of EN 13001-3-3, so it is a simple step to reduce this amount of rotations. For EN 13001-3-3 the amount of static rotations cannot be changed, as it is prescribed by the standard. For FEM the static force is the lowest of all, which confirms the conclusion from the previous comparison that ISO/FEM is very conservative

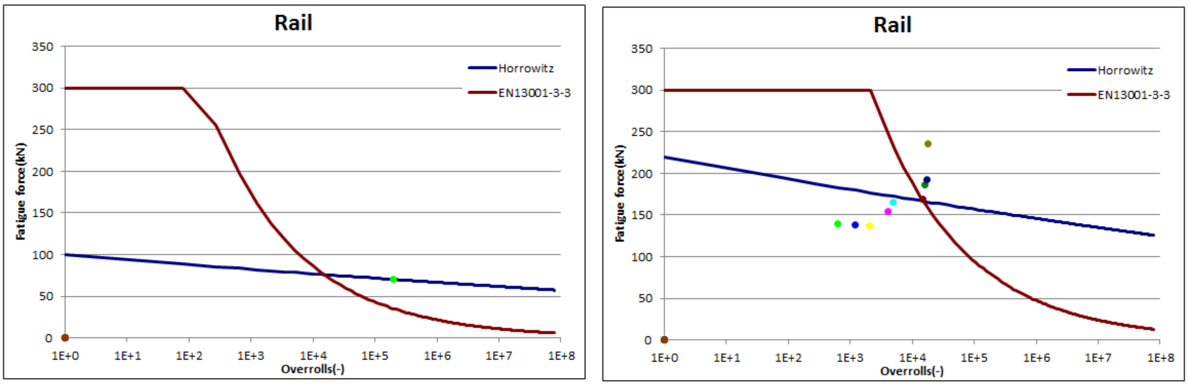
From the first fatigue test it can be concluded that for a high amount of rotations none of the standards give a proper lifetime prediction. However the prediction from EN 13001-3-3 is a lot more realistic than Horrowitz.

For a low amount of rotations EN 13001-3-3 gives a quite accurate approximation of the lifetime, while Horrowitz is again very conservative. Furthermore this test shows that the effect of strain hardening is very large. If this is taken into account Horrowitz predicts the lifetime best.

As Horrowitz is very conservative for all load cases, the conclusion from the test report is to rewrite the Stribeck value for rail without axial load, like the tested rail, to withstand the imposed load at 200,000 cycles. This yields a new Stribeck value:

$$k_h = max((0.812 - 0.025\log(n)), 0.25) \tag{B.38}$$

With this adjustment the rail in test 1 the lifetime consumed is 0.91 and test 2 fails during the 121 $\%$ load case. Because the plots are capped at the static force, the difference between the methods is unclear, so the static force has been omitted in figure B.7 and for readability the EN 13001-3-3 curve has been capped at 300 kN, while in reality it keeps increasing as the amount of rotations decreases. The plots clearly show that with the new Stribeck value the slope of the Horrowitz curve decreases. This makes that the standard now fulfills the tests. EN 13001-3-3 is now passed as the least conservative method much earlier than before.



(a) Test 1

(b) Test 2

Figure B.7: Rail Fatigue curve for the new Stribeck value without Static Force

Although this new Stribeck value represents the test results better than the old one, it cannot simply be taken over. The reason is that there is not enough data to validate this change. It is advised to perform this tests for a number of load cases geometries and material relevant to Huisman and choose a Stribeck value that complies with all these variations. For now the old, albeit somewhat conservative, Stribeck value is kept.