

Sylwester MARKUSIK

Silesian University of Technology, Faculty of Transport,
Department of Logistics and Mechanical Handling
Kraśińskiego St. 8, 40-019 Katowice, Poland
Corresponding author. E-mail: sylwester.markusik@polsl.pl

**STRENGTH CONDITIONS FOR WHEEL/RAIL SYSTEM ACCORDING
WITH EUROPEAN STANDARD EN 13001-3.3**

Summary. The European Standard EN 13001 was prepared by CEN/TC-147/WG 2 “Cranes. Design - General”, where author of this article is one of the members of the international working group. The European Standard EN 13001-3.3 is one part of a large EN 13001. Transportation machines like cranes, to conform with the essential health and safety requirements of the Machinery Directive. This standard also established the interface between the user and the designer, as well as between the designer and the component manufacturer, in order to form a basic selecting wheel/rail systems of transportation machines like: cranes, baggers or removable conveyor belts.

**WARUNKI WYTRZYMAŁOŚCIOWE DLA UKŁADU KOŁO/SZYNA NA
PODSTAWIE NORMY EUROPEJSKIEJ EN 13001-3.3**

Streszczenie. Norma Europejska EN 13001-3.3 „Warunki wytrzymałościowe układu koło/szyrna”, opracowana przez Komisję Techniczną CEN TC 147 „Dźwignice”, jest częścią obszernej normy EN 13001 „Dźwignice. Ogólne zasady projektowania”. Maszyny transportowe, takie jak dźwignice, muszą być projektowane pod względem bezpieczeństwa zgodnie z Dyrektywą Maszynową UE. Norma ta stanowi informację dla projektantów, użytkowników oraz wykonawców elementów i zespołów tych maszyn, w jaki sposób należy dobierać postać konstrukcyjną układów koło/szyrna dla maszyn transportowych typu: dźwignice, koparki lub przejezdne przenośniki taśmowe.

1. INTRODUCTION

The European Standard EN 13001-3.3 has been prepared to be a harmonized standard to provide one means for the mechanical design and technical verification of the requirements and methods to prevent mechanical hazards of wheel/rail contacts of cranes or baggers, by design and theoretical verification. This standard covers steel and cast iron wheels.

Transportation machines like cranes, to conform with the essential health and safety requirements of the Machinery Directive. This standard also established interface between the user and the designer, as well as between the designer and the component manufacturer, in order to form a basic selecting wheel/rail systems of transportation machines and theirs components.

The European Standard EN 13001 was prepared by CEN/TC-147/WG 2 “Cranes. Design - General” where the author of this article is one of the members of the international working group.

The European Standard EN 13001-3.3 is a part of a large EN 13001. The other parts of this standard are as follows:

- Part 1: General principles and requirements,
- Part 2: Load actions,
- Part 3.1: Limit states and proof competence of steel structures,
- Part 3.2: Limit states and proof competence of rope/ reeving systems,
- Part 3.3: Limit states and proof competence of wheel/rail contacts,
- Part 3.4: Limit states and proof competence of machinery,
- Part 3.5: Hooks design.

Parts of EN 13001: 1, 2, 3.1 and 3.2 are just settled up in the 2006 year and introducing into the national standards of the EU countries, now. The next standards will be prepare in the near future. It shows in this article the method to determine the strength conditions for the wheel/rail contacts systems of cranes, baggers and other mechanical handling machines are useful for a better way to choose wheel units and details (shaft, bearing, etc.) of that mechanisms and can be treating like comments for EN 13 001-3.3.

2. GENERAL PRINCIPLES FOR PROOF CALCULATION

The objective of this calculation is to prove theoretically that transportation machines, like cranes, taking into account the service conditions agreed between the user, designer and/or manufacturer, as well as the states during erection, dismantling and transport, has been designed in conformance to the safety requirements to prevent mechanical hazards. The proof of competence according to EN 13001 shall be carried out by using general principles and methods appropriate for this purpose and corresponding with the recognized state of the art in machines design.

Hazards can occur if extreme values of load effects or their histories exceed the corresponding limit states. To prevent these hazards with a margin of safety, it shall be shown that the calculated extreme values of load effects from all loads acting simultaneously on a machine and multiplied with an adequate partial safety coefficient, as well as the estimated histories of load effects, do not exceed their corresponding limit states at any critical point of the machine. For this purpose the limit state method, and where applicable the allowable stress method, is used in accordance with international and European design codes.

The analysis of load actions from individual events or representative use of machines, like crane (representative load histories) is required to reflect realistic unfavorable operational conditions and sequences of actions of the crane. Figure 1 illustrates the general layout of a proof calculation for cranes.

3. PROOF OF STATIC STRENGTH

3.1. Design rope force

For the proof of static strength it shall be proven for all relevant load combinations [2]:

$$F_{Sd,s} \leq F_{Rd,s} \quad (1)$$

where: $F_{Sd,s}$ - is the design contact force, $F_{Rd,s}$ - is the limit design contact force.

The design contact force $F_{Sd,s}$ of all wheel/rail contacts shall be calculated for all relevant load combinations of EN 13001-2, taking into account the respective dynamic factors ϕ , partial safety factors γ_p and where required the risk coefficient γ_n [3].

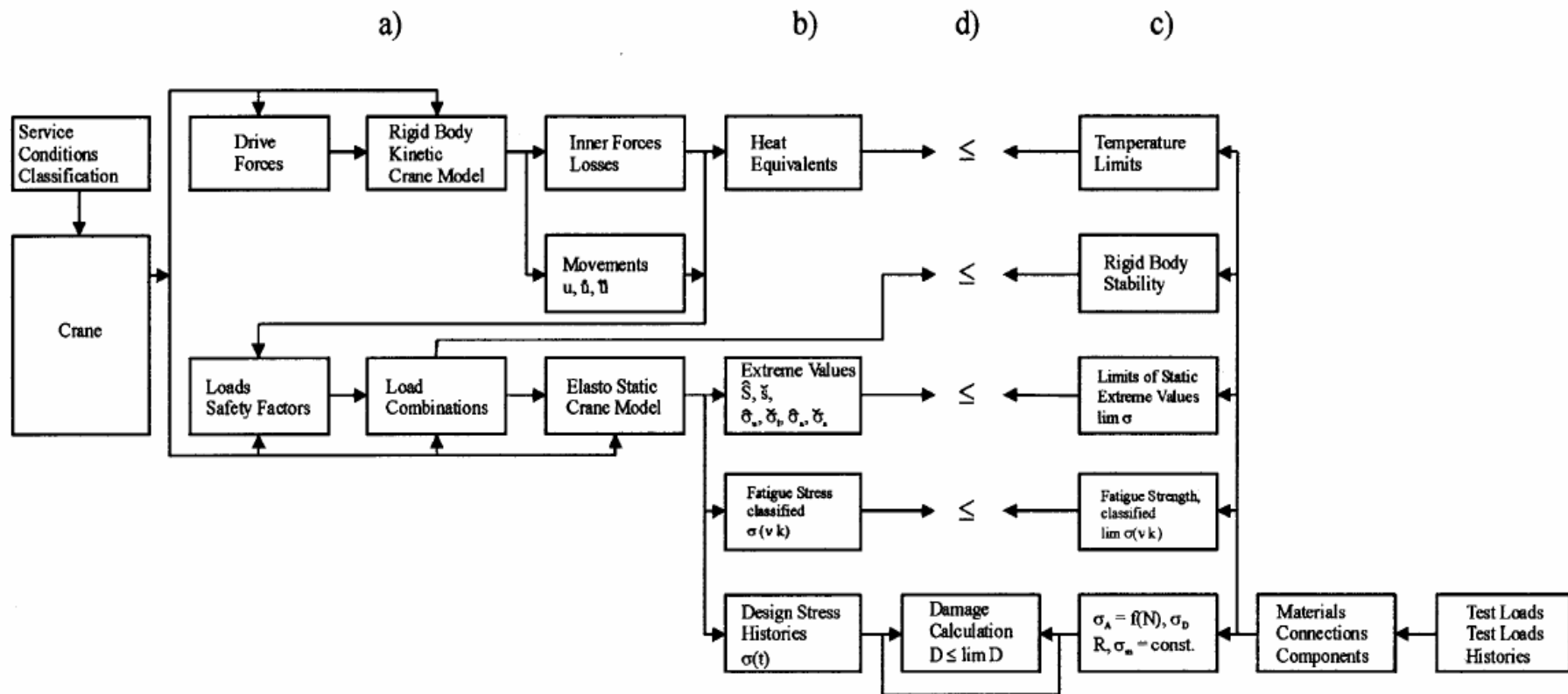


Fig. 1. Layout of the proof calculation where: a) models of crane and loads; b) load actions; c) limit states; d) proof

Rys. 1. Przebieg obliczeń wytrzymałościowych: a) model dźwignicy i obciążeń; b) działanie obciążeń; c) stan graniczny; d) sprawdzenie

A contact force of the magnitude of the static limit design contact force $F_{Rd,S}$ causes permanent radial deformation of 0.02% of the wheel radius.

The static limit design contact force $F_{Rd,S}$ depends on:

- materials properties (modulus of elasticity and hardness) of wheel and rail,
- contact case (point contact or line contact),
- geometry (radius of wheel and rail),
- decreasing effects (stiffness, edge effects).

The static limit design contact force $F_{Rd,S}$ for cases of point contact are describing by formulas where:

$$F_{Rd,S} = \left(\frac{HB^*}{\gamma_{Rcp}} \right)^3 \cdot \left(\frac{\pi}{1,5} \right) \cdot \left[\frac{3 \cdot (1 - \nu^2)}{E_m \cdot \left(\frac{2}{D_w} + \frac{1}{r_k} \right)} \right]^2 \quad (2)$$

$$\gamma_{Rcp} = \gamma_m \cdot \gamma_{scp}$$

E_m is the equivalent elasticity modulus, ν is the radial strain coefficient ($\nu = 0.3$ for steel), D_w is the wheel diameter, r_k is the radius of the rail surface or the second wheel radius (see Figure 2), HB^* - the point of maximum shear, γ_{Rcp} is the resulting resistance coefficient for point contact, γ_m is the general resistance coefficient ($\gamma_m=1.1$), γ_{scp} is the specific resistance coefficient for point contact ($\gamma_{scp}=0.1$).

The point of maximum shear is situated at depth z_{mp} below the surface [5].

$$z_{mp} = 0.47 \cdot \frac{HB^*}{\gamma_{Rcp}} \cdot \frac{2 \cdot \pi \cdot (1 - \nu^2)}{E_m \cdot \left(\frac{2}{D_w} + \frac{1}{r_k} \right)} \quad (3)$$

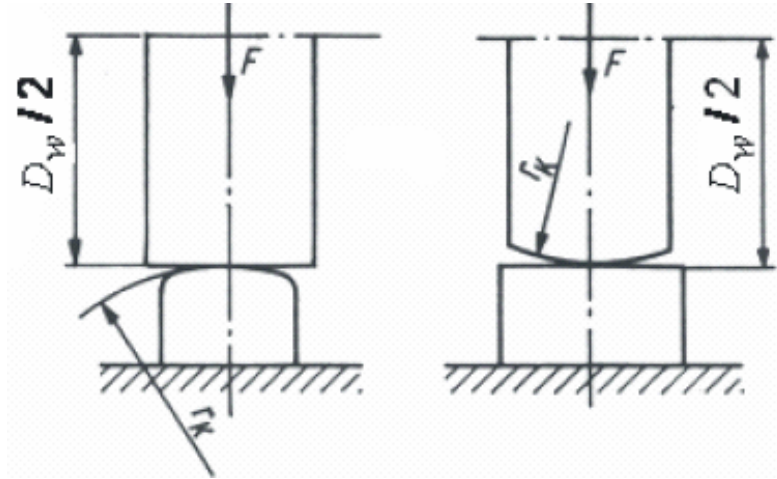


Fig. 2. Point contact
Rys. 2. Kontakt punktowy

The static limit design contact force $F_{Rd,S}$ for cases of line contact is describe by formula 3 [6].

$$F_{Rd,S} = \left(\frac{HB^*}{\gamma_{Rcl}} \right)^2 \cdot \frac{\pi \cdot D_w \cdot l \cdot (1 - \nu^2)}{E_m} \cdot f_1 \cdot f_2 \quad (4)$$

$$z_{ml} = 1.56 \cdot \frac{HB^*}{\gamma_{Rcl}} \cdot \frac{D_w \cdot (1 - \nu^2)}{E_m}$$

where: f_1 is the decreasing factor for edge pressure, f_2 is the decreasing factor for non-uniform pressure distribution.

Typical line contacts are shown in figure 3.

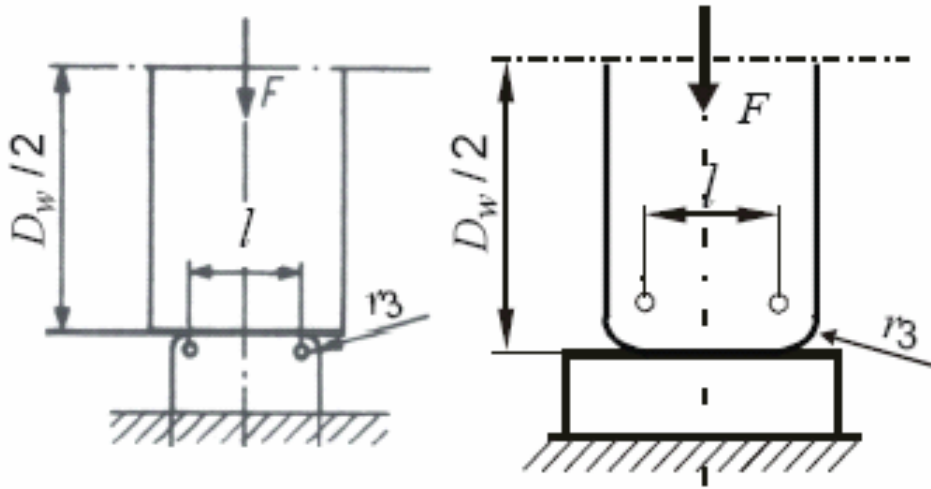


Fig. 3. Line contact
Rys. 3. Kontakt liniowy

Sharp edges at the end of the contact line of wheel or rail decrease the limit design contact force. This effect is taken into account by factor f_1 , given in table 1.

An ideal uniform distribution requires sufficient elasticity of the rail fixation or support and/or wheels on hinged legs. Otherwise deformation of the machine structure (e.g. bending of main girders) or tolerances in rail alignment result in non-uniform pressure distribution, decreasing the limit design contact force. This effect is taken into account by factor f_2 , given in table 2 (Tolerance classes according ISO 12488-1).

Table 1

Factor f_1 for edge pressure

edge r_3/l	f_1
sharp edge: $r_3/l \leq 0.1$	0.5
$0.1 < r_3/l < 0.2$	$5^*(r_3/l)$
smooth edge: $r_3/l \geq 0.2$	1.0
where l is the length of the contact line and r_3 is the radius of the edge (see Figure 3)	

Table 2

Factor f_2 for pressure distribution

	Tolerance class 1	Tolerance class 2	Tolerance class 3	Tolerance class 4
wheels in hinged leg	1,0	1,0	0,95	0,9
rail mounted on elastic layer	0,9	0,85	0,8	0,7
stiff support, rail welded on support	0,8	0,75	0,7	0,6

4. PROOF OF FATIGUE STRENGTH

For the proof of fatigue strength of all wheel/rail contacts it shall be proven that for each wheel and for all points on the rails:

$$F_{Sd,f} \leq F_{Rd,f} \quad (5)$$

where: $F_{Sd,f}$ is the design contact force for fatigue, $F_{Rd,f}$ is the limit design contact force for fatigue.

The design contact force $F_{Sd,f}$ shall be calculated for regular loads (load combinations A of EN 13001-2) only, with the respective dynamic factors ϕ , partial safety factors γ_p , and risk coefficient γ_n set to 1,0 [1].

The limit design contact force $F_{Rd,f}$ shall be calculated for wheels and rails separately by formula:

$$F_{Rd,f} = \frac{F_u}{\sqrt[m]{S_c \cdot \gamma_{cf}}} \cdot f_f \quad (6)$$

where: F_u is the minimum contact force, S_c is the contact force history parameter, γ_{cf} is the minimum contact resistance factor ($\gamma_{cf} = 1.1$), f_f is the factor of further influences, m is the exponent for wheel/rail contacts: $m=3$ for cases of point contact and $m=10/3$ for cases of line contact.

In analogy to stress history parameter (see EN 13001-1 [1]), the contact force history parameter is given by:

$$s_c = k_c \cdot v_c \quad (7)$$

where: k_c is the contact force spectrum factor, v_c is the relative total number of rolling contacts.

The contact force history parameter shall be determined either by direct use of equation (7) or simplified (based on experience) by selection of a class S_c from table 3.

Table 3

Classes S_c of contact force history parameter s_c

Class	S_{c0}	S_{c1}	S_{c2}	S_{c3}	S_{c4}	S_{c5}	S_{c6}	S_{c7}	S_{c8}	S_{c9}
s_c	0,008	0,016	0,032	0,063	0,125	0,25	0,5	1,0	2,0	4,0

The contact force spectrum factor k_c is calculated by formula:

$$k_c = 1/i_{tot} \cdot \sum_{i=1}^{i_{tot}} \left(\frac{F_{Sd,f,i}}{F_{Sd,f}} \right)^m \quad (8)$$

where: i is the index of one rolling contact with $F_{Sd,fi}$, i_{tot} is the total number of rolling contacts during the useful life of wheel or rail, $F_{Sd,fi}$ is the design contact force in contact i , $F_{Sd,f}$ is the maximum design contact force.

The relative total number of rolling contacts v_c is calculated by formula:

$$v_c = \frac{i_{tot}}{i_D} \quad (9)$$

where: i_{tot} is the total number of rolling contacts during the useful life of wheel or rail, i_D is the number of rolling contacts at reference point: $i_D = 6,4 \cdot 10^6$ [2].

The factor f_f takes into account further influences on the limit design contact force:

$$f_f = f_{f1} \cdot f_{f2} \cdot f_{f3} \cdot f_{f4} \cdot f_{f5} \quad (10)$$

where: factor for edge pressure $f_{f1} = f_1$; pressure distribution factor $f_{f2} = 1,0$ [2].

A skewing wheel causes wear of wheel and rail and thus shortens its useful life. The wear is increased overproportionally in relation to the skewing angle α . This effect is taken into account by factor f_{f3} .

$$f_{f3} = 1 \quad \text{for } \alpha \leq 3 \text{ ‰} \quad (11)$$

$$f_{f3} = \sqrt[3]{\frac{3}{\alpha}} \quad \text{for } \alpha > 3 \text{ ‰}$$

where: α is the skewing angle either of the crane in ‰, calculated according to [2].

Matching materials cause equal wear of a wheel and a rail per rolling contact. Non-matching materials will increase wear of one partner and decrease wear of the other partner.

This may be taken into account by factor f_{f4} .

$$f_{f4} = f_{4w} \text{ – for wheels} \quad (12)$$

$$f_{f4} = f_{4r} \text{ – for rails}$$

where:

$$f_{4w} = \frac{1}{f_{4r}} \quad (13)$$

The factor f_{f4} may be chosen from experience in the range between 0.5 and 2,0 [2].

The mechanical drive factor f_{f5} may be chosen:

$$f_{f5} = 0,95 \text{ for driven wheels in unclean environment,}$$

$$f_{f5} = 1,0 \text{ for non- driven wheels or wheels in clean environment.}$$

CONCLUSION

- New standard EN 13001- 3.3 giving more practical information for designers and users of wheel units in different mechanisms of cranes or baggers than others standards (like: EN ISO 12100-1:2003 *Safety of machinery — Basic concepts, general principles for design* or ISO/DIS 12488-1, *Cranes — Tolerances for wheels and travel and traversing tracks*).

- For users, eg. designers of driving mechanisms, this standard is not sufficient, because in live ly applications they should have other (so thick) standards from EN 13001 series.
- Actually, standard EN 13001- 3.3 is on the last consulting way with national standardization committees and discussion between members of CEN Working Group WG 2.

Literature

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