	FEDERATION EUROPEENNE DE LA MANUTENTION SECTION I HEAVY LIFTING APPLIANCES	F.E.M. 1.001 3rd EDITION REVISED 1998.10.01

RULES FOR THE DESIGN OF HOISTING APPLIANCES

BOOKLET 9

SUPPLEMENTS AND COMMENTS TO BOOKLETS 1 to 8

The total 3rd Edition revised comprises booklets 1 to 5 and 7 to 9

Also available in French and German

Document prepared by the technical commission of **FEM** (European Handling Federation) **Section I** « Heavy lifting and handling equipment ».

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The third edition of the "Rules for the design of hoisting appliances" dated 1987.10.01 included 8 booklets. An addition to this edition was compiled in 1998. This addition is incorporated in booklet 9, which also replaces booklet 6.

This booklet forms part of the "Rules for the design of hoisting appliances" 3rd edition revised, consisting of 8 booklets :

Booklet 1 - Object and scope

Booklet 2 - Classification and loading on structures and mechanisms

Booklet 3 - Calculating the stresses in structures

Booklet 4 - Checking for fatigue and choice of mechanism components

Booklet 5 - Electrical equipment

~~Booklet 6 - Stability and safety against movement by the wind~~

Booklet 7 - Safety rules

Booklet 8 - Test loads and tolerances

Booklet 9 - Supplements and comments to booklets 1 to 8

NOTE: Booklet 9 must not therefore be used separately.

BOOKLET 9

SUPPLEMENTS AND COMMENTS TO BOOKLETS 1 TO 8

CONTENTS

9.1 PREFACE

The Rules for the Design of Hoisting Appliances established by the Technical Committee of Section I of the Fédération Européenne de la manutention (F.E.M), which were published in the form of 8 booklets, have been increasing widely used in many countries all over the world.

However, these rules were elaborated at the beginning of the eighties and must advance to keep pace with the improving state of knowledge and the increasingly efficient conception tool being used.

The need for a revision is based on several observations :

- The harmonized standard EN 13001 from the work of CEN/TC147/WG2 will be applied progressively only at the beginning of the 21st century.

  Thus, it is not desirable to await this date and continue to refer to the FEM rules, certain parts of which became obsolete.

- FEM 1.001 consolidates a great deal of experience and serves manufacturers and customers often as a basis for calculations.

- Development of the rules should make future application of the harmonized standards easier, based on methods with limit states, among others.

The text below shall be considered as a supplement to those texts which are the subject of booklets 2, 3, 4 and 8. Booklet 6 is deleted.

9.2 PRESENTATION

At the begining of each clause, there is a reference to the clause coming from booklet 2, 3, 4, 6 or 8 that the new text may replace.

Example :

Clause 2.2.2.1.1.of booklet 2 may be replaced by the following text:

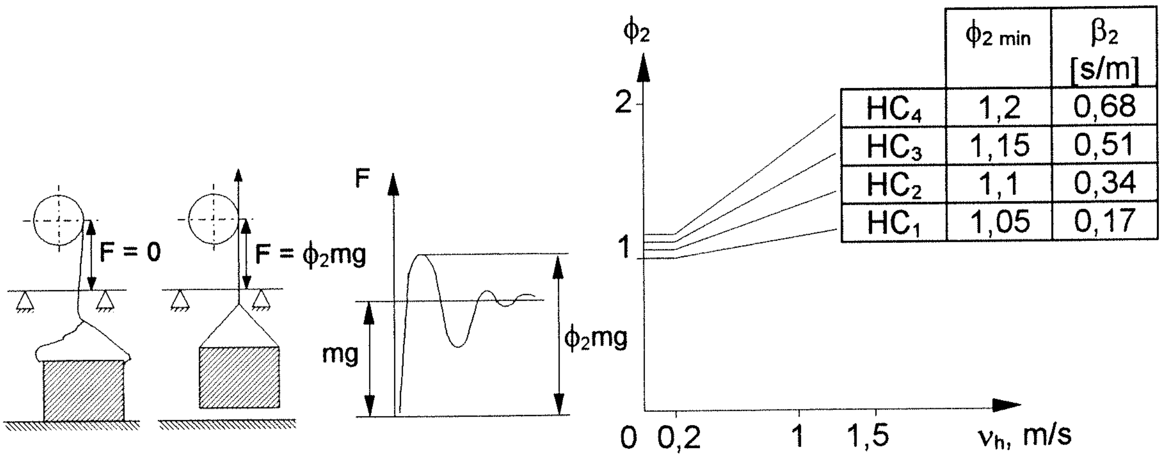
9.3 VALUES OF THE DYNAMIC COEFFICIENT ψ

Clause 2.2.2.1.1.of booklet 2 may be replaced by the following text:

For the coefficient ψ given by the clause 2.2.2.1.1 of booklet 2, we can take the value ϕ_2 given by the following text :

In the case of hoisting an unrestrained grounded load, the dynamic effects of transferring the load from the ground to the crane shall be taken into account by multiplying the gravitational force due to the mass of the hoist load by a factor ϕ_2 (see figure F.9.3).

The mass of the hoist load includes the masses of the payload, lifting attachments and a portion of the suspended hoist ropes or chains, etc.



	$\phi_{2\,min}$	β_2 [s/m]
HC_4	1,2	0,68
HC_3	1,15	0,51
HC_2	1,1	0,34
HC_1	1,05	0,17

Figure F.9.3 - Factor ϕ_2

The factor ϕ_2 shall be obtained as follows :

$$\phi_2 = \phi_{2min} + \beta_2 v_h$$

ϕ_{2min} and β_2 are given in table T.9.3.a for the appropriate hoisting class. For the purpose of this standard, cranes are assigned to hoisting classes HC1 to HC4 according to their dynamic characteristics. (The selection of hoisting classes depends on the particular types of cranes and is dealt with in the European Standards for specific crane types.) Equally, values for ϕ_2 can be determined by experiments or by analysis without reference to hoisting class.

v_h is the steady hoisting speed, related to the lifting attachment. Values of v_h are given in table T.9.3.b.

Table T.9.3.a - Values of β_2 and ϕ_{2min}

Hoisting class of the appliance	β_2 s/m	$\phi_{2\ min.}$
HC1	0,17	1,05
HC2	0,34	1,10
HC3	0,51	1,15
HC4	0,68	1,20

Table T.9.3.b - Values of v_h for estimation of ϕ_2

Load combination	Type of hoist drive and its operation method of operation				
	HD1	HD2	HD3	HD4	HD5
Case I, Case II	$v_{h,max}$	$v_{h,CS}$	$v_{h,CS}$	$0,5 \cdot v_{h,max}$	$v_{h.} = 0$
Case III	-	$v_{h,max}$	-	$v_{h,max}$	$0,5 \cdot v_{h,max}$

Where :

HD1 hoist drive cannot be operated with creep speed;

HD2 a steady creep speed for the hoist drive can be selected by the crane driver;

HD3 hoist drive control system ensures the use of a steady creep speed until the load is lifted from the ground;

HD4 a stepless variable speed control can be operated by the crane driver;

HD5 after prestressing the hoist medium, a stepless variable speed control is provided by the drive control system independent of the crane driver;

$v_{h,max.}$ is the maximum steady hoisting speed;

$v_{h,CS}$ is the steady hoisting creep speed.

9.4 FORCE DUE TO HORIZONTAL MOTIONS S_H.

Clause 2.2.3.of booklet 2 may be replaced by the following text:

9.4.1 Transverse action due to rolling action

Example of a method for analysing forces due to skewing.

9.4.1.1 Model of appliance

To enable an estimation to be made of the tangential forces between wheels and rails and also of the forces between the acting guide means that are caused by skewing of the lifting appliance, a simple travel-mechanics model is necessary. The lifting appliance is considered to be travelling at a constant speed without anti-skewing control.

The model consists of n pairs of wheels in line, of which p pairs are coupled. An individual (i) pair of wheels can be defined, either as coupled (C) mechanically or electrically, or mounted independently (I) of each other. The latter condition is also valid in the case of independent single drives.

The wheels are arranged in ideal geometric positions in a rigid crane structure which is travelling on a rigid track. Differences in wheel diameters are neglected in this model. They are either fixed (F) or movable (M) in respect of lateral movement. The lateral degree of freedom can, for example, be provided by a hinged leg.

The different combinations of transversally in-line wheel pairs that are possible are shown in figure F.9.4.a.

	Coupled (C)	Independent (I)
Fixed/Fixed (F/F)	CFF	IFF
Fixed/Movable (F/M)	CFM	IFM

Figure F.9.4.a - Different combinations of wheel pairs

In figure F.9.4.b the positions of the wheel pairs relative to the position of the guide means in front of the travelling crane are defined by the distance d_i.

NOTE: Where flanged wheels are used instead of an external guide means, $d_1 = 0$.

It is assumed that the gravitational forces due to the masses of the loaded appliance (mg) are acting at a distance μl from rail 1 and are distributed equally to the n wheels at each side of the crane runway.

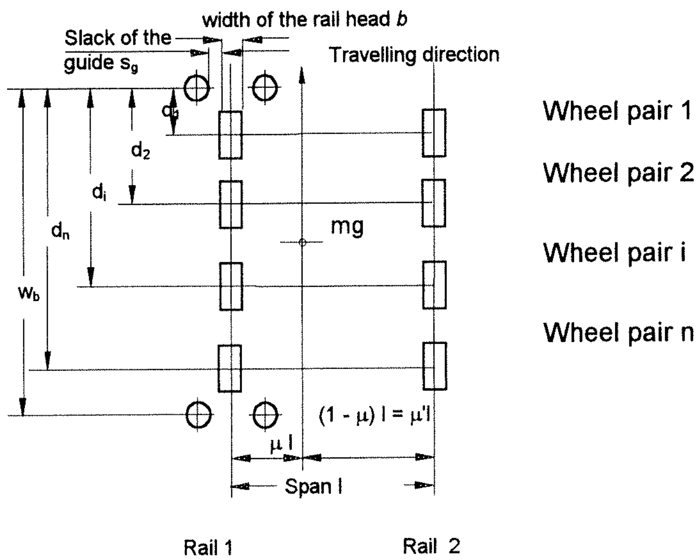


Figure F.9.4.b - Positions of wheel pairs

9.4.1.2 Relationship between tangential forces and displacements

It is at first necessary to assume a relationship between the tangential forces and the corresponding displacements occurring between wheel and rail. Since the wheel has to transfer drive moments (M_y) to the rail and its movement is restricted by the system (crane and runway), it slides in the longitudinal and lateral directions [$u(u_x, u_y)$]; corresponding tangential forces (F_x, F_y) react on the crane (see figure F.9.4.c).

Geometry:

Forces:

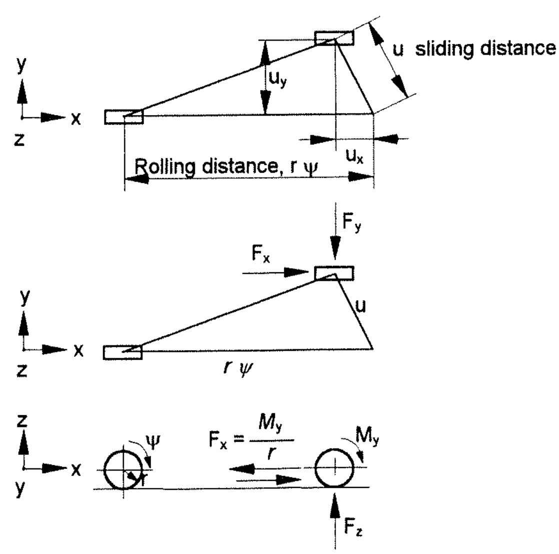


Figure F.9.4.c - Tangential forces and displacements

In general, a relationship exists between the sliding distances (u_x,u_y), the free-rolling distance $r\psi$, the wheel load F_z and the tangential forces (F_x, F_y), as follows :

$$F_x = f_x(s_x,\ s_y,\ p_c,\ \text{surface conditions}) \cdot F_z$$

$$F_y = f_y(s_x,\ s_y,\ p_c,\ \text{surface conditions}) \cdot F_z$$

The friction coefficients of the rolling wheel (f_x, f_y) depend on the slip, i.e. the relation between slide and free-rolling distances ($s_x = u_x / r\psi$, $s_y = u_Y / r\psi$), on the contact pressure between wheel and rail (p_c) and on the surface conditions of the rail. To simplify the calculation, the following empirical relationships may be used :

$$f_x = 0{,}3 \cdot \left[1 - e^{\left(-250.\ s_x\right)} \right],\ \text{for } s_x \leq 0{,}015$$

$$f_y = 0{,}3 \cdot \left[1 - e^{\left(-250.\ s_y\right)} \right],\ \text{for } s_y \leq 0{,}015$$

9.4.1.3 Forces due to skewing

The crane model is assumed to be travelling in steady motion and to have skewed to an angle α, as shown in figure F.9.4.d. The appliance may be guided horizontally by external means or by wheel flanges.

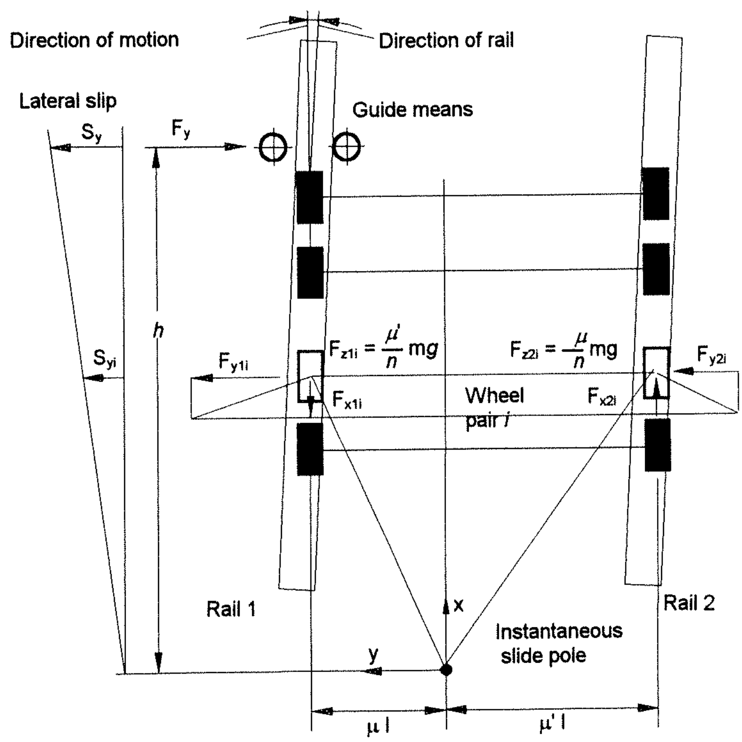


Figure F.9.4.d - Forces acting on crane in skewed position

A guide force F_y is in balance with the tangential wheel forces F_{x1i}, F_{y1i}, F_{x2i}, F_{y2i}, which are caused by rotation of the appliance about the instantaneous slide pole. With the maximum lateral slip $s_y = \alpha$ at the guide means and a linear distribution of the lateral slip s_{yi} between the guide means and the instantaneous slide pole, the corresponding skewing forces can be calculated as follows :

a) Distance between instantaneous slide pole and guide means h

For systems F/F, $h = (p\mu\mu' l^2 + \Sigma d^2_i) / \Sigma d_i$
For systems F/M, $h = (p\mu l^2 + \Sigma d^2_i) / \Sigma d_i$

where :

p	is the number of pairs of coupled wheels;
μ	is the distance of the instantaneous slide pole from rail 1;
μ'	is the distance of the instantaneous slide pole from rail 2;
l	is the span of the appliance;
d_i	is the distance of wheel pair i from the close-fitting guide means.

b) guide force F_y

$$F_y = \nu f mg$$

where :

$\nu = 1 - \Sigma d_i /nh$,	for systems F/F
$= \mu' (1 - \Sigma d_i /nh)$,	for systems F/M
$f = 0{,}3 \cdot (1 - e^{-250\,\alpha})$	where $\alpha < 0{,}015$ rad;
mg	is the gravitational force due to the mass of the loaded appliance;
n	is the number of wheels at each side of the crane runway.

9.4.1.4 Tangential forces, F_x and F_y

$$F_{x1i} = \xi_{1i}\, fmg$$
$$F_{x2i} = \xi_{2i}\, fmg$$
$$F_{Y1i} = \nu_{1i}\, fmg$$
$$F_{Y2i} = \nu_{2i}\, fmg$$

Where :

f and mg are as given in clause 9.4.1.3.b)

ξ_{1i}, ξ_{2i}, ν_{1i} and ν_{2i} are as given in table T.9.4.

Table T.9.4 - Values of ξ_{1i}, ξ_{2i}, ν_{1i} and ν_{2i}

<table>
<tr><th>Combinations of wheel pairs (see figure F.9.4.a)</th><th>$\xi_{1i} = \xi_{2i}$</th><th>ν_{1i}</th><th>ν_{2i}</th></tr>
<tr><td>CFF</td><td>$\mu\mu'l/nh$</td><td rowspan="4">$\frac{\mu'}{n}(1-\frac{d_i}{h})$</td><td rowspan="2">$\frac{\mu}{n}(1-\frac{d_i}{h})$</td></tr>
<tr><td>IFF</td><td>0</td></tr>
<tr><td>CFM</td><td>$\mu\mu'l/nh$</td><td rowspan="2">0</td></tr>
<tr><td>IFM</td><td>0</td></tr>
</table>

9.4.1.5 Skewing angle α

The skewing angle α , which should not exceed 0,015 radians, shall be chosen taking into account the space between the guide means and the rail as well as reasonable dimensional variation and wear of the appliance wheels and the rails as follows :

$$\alpha = \alpha_g + \alpha_w + \alpha_t$$

Where :

$\alpha_g = s_g/w_b$	is the part of the skewing angle due to the slack of the guide;
s_g	is the slack of the guide;
w_b	is the distance between the guide means;
$\alpha_w = 0{,}1\ (b/w_b)$	is the part of the skewing angle due to wear;
b	is the width of the rail head;
$\alpha_t = 0{,}001$ rad	is the part of the skewing angle due to tolerances.

9.4.2 Buffer effects on the structure

In clause 2.2.3.4.1 replace 0,7 m/s with 0,4 m/s

9.5 WIND ACTION

Clause 2.2.4.1.of booklet 2 may be replaced by the following text:

Other recommendations or work results can also be used provided that the same level of safety is obtained

9.6 QUALITY OF STEEL

Clause 3.1.3 of booklet 3 may be replaced by the following text:

The properties of the steel grades frequently used are provided in the following standards :

EN 10025	Hot-rolled products of non-alloy structural steels. Technical delivery conditions;
EN 10113-1	Hot-rolled products in weldable fine grain structural steels - Part 1: General delivery conditions;
EN 10137-1	Plates and wide flats made of high yield strength structural steels in the quenched and tempered or precipitation hardened conditions - Part 1: General delivery conditions;
EN 10149-1	Hot-rolled flat products made of high yield strength steels for cold forming - Part 1: General delivery conditions;
EN 10210-1	Hot finished structural hollow sections of non-alloy and fine grain structural steels - Part 1: Technical delivery requirements;
EN 10219-1	Cold formed welded structural hollow sections of non-alloy and fine grain steels - Part 1: Technical delivery requirements.

The quality of steels in these design rules refers to the property of the steel to exhibit ductile behaviour at determined temperatures.

The steels are divided into four quality groups. The group in which the steel is classified, is obtained from its notch ductility in a given test and at a given temperature.

Tables T.9.6.a, T.9.6.b, T.9.6.c and T.9.6.d comprises the notch ductility values and test temperatures for the four quality groups.

The indicated notch ductilities are minimum values, being the mean values from three tests, longitudinal test pieces are used.

The notch ductility shall be determined in accordance with V-notch impact tests according to the European Standard EN 10045-1.

Steels of different quality groups can be welded together.

T_C	is the test temperature for the V-notch impact test.
T	is the temperature at the erection site of the crane.
T_C and T	are not directly comparable as the V-notch impact test imposes a more unfavourable condition than the loading on the crane in or out of service.

Table T.9.6.a - Quality groups

Quality group	Impact energy according to EN 10045-1 J	Test temperature Tc °C	Steels, corresponding to the quality group Designation of steels: According to old standard	According to EN 10027-1 & ECISS IC 10	According to EN 10027-2	Old standard	New standard
1	-	-	ST 37-2 ST 44-2	S235JR S275JR	1.0037 1.0044	DIN 17100	EN 10025 (1993)
			50 B	S355JR	1.0045	BS 4360 (1972)	
2	27	+ 20	Fe 360-B Fe 430-B Fe 510-B	S235JR S275JR S355JR	1.0037 1.0044 1.0045	EN 10025 (1990)	
			R St 37-2 St 44-2	S235JRG2 S275JR	1.0038 1.0044	DIN 17100	
			E 24(A37)-2 E 28 - 2 E 36 (A52)-2	S235JR S275JR S355JR	1.0037 1.0044 1.0045	NF A 35-501	
			40 B 43 B	S235JRG2 S275JR	1.0038 1.0044	BS 4360 (1972)	
3	27	± 0	Fe 360-C Fe 430-C Fe 510-C	S235JO S275JO S355JO	1.0114 1.0143 1.0553	EN 10025 (1990)	
			St 37-3U St 44-3U ST 52-3U	S235JO S275JO S355JO	1.0114 1.0143 1.0553	DIN 17100	
			E 24 (A37)-3 E 28 - 3 E 36 (A52)-3	S235JO S275JO S355JO	1.0114 1.0143 1.0553	NF A 35-501	
			40 C 43 C 50 C	S235JO S275JO S355JO	1.0114 1.0143 1.0553	BS 4360 (1972)	
4	27	- 20	Fe 360-D1 Fe 360-D2 Fe 430-D1 Fe 430-D2 Fe 510-D1 Fe 510-D2	S235J2G3 S235J2G4 S275J2G3 S275J2G4 S355J2G3 S355J2G4	1.0116 1.0117 1.0144 1.0145 1.0570 1.0577	EN 10025 (1990)	
	40		Fe 510-DD1 Fe 510-DD2	S355K2G3 S355K2G4	1.0595 1.0596		
	27		St 37-3N - St 44-3N - St 52-3N -	S235J2G3 S235J2G4 S275J2G3 S275J2G4 S355J2G3 S355J2G4	1.0116 1.0117 1.0144 1.0145 1.0570 1.0577	DIN 17100	
	40		- -	S355K2G3 S355K2G4	1.0595 1.0596		
	27		E 24 (A37)-4 E 28 - 4	S235J2G3 S275J2G3	1.0116 1.0144	NF A 35-501	
	40		E 36 (A52)-4	S355K2G3	1.0595		
	27		40 D 43 D 50 D	S235J2G3 S275J2G3 S355J2G3	1.0116 1.0144 1.0570	BS 4360 (1972)	
			St 52-3N	S355J2H	1.0576	DIN 17100	EN 10210-1 (1994)
			50D	S355J2H	1.0576	BS 4360 (1972)	

Table T.9.6.b - Quality groups

Quality group	Impact energy according to EN 10 045-1 J	Test temperature Tc °C	Steels, corresponding to the quality group Designation of steels: According to old standard	According to EN 10027-1 & ECISS IC 10	According to EN 10027-2	Old standard	New standard
4	40 47	- 20		S275N S275NL	1.0490 1.0491	NF A 36-201 (1984)	EN 10113-2 (1993)
	40 47		E 355 R E 355 FP	S355N S355NL	1.0545 1.0546		
	40 47		E 420 R E 420 FR	S420N S420NL	1.8902 1.8912		
	40 47		E 460 R E 460 FP	S460N S460NL	1.8901 1.8903		
	40 47		StE285 TStE285	S275N S275NL	1.0490 1.0491	DIN 17102 (1983)	EN 10113-2 (1993)
	40 47		StE355 TStE355	S355N S355NL	1.0545 1.0546		
	40 47		StE420 TStE420	S420N S420NL	1.8902 1.8912		
	40 47		StE460 TStE460	S460N S460NL	1.8901 1.8903		
	40 47		40EE	S275N S275NL	1.0490 1.0491	(United Kingdom)	EN 10113-2 (1993)
	40 47		50EE	S355N S355NL	1.0545 1.0546		
	40 47			S420N S420NL	1.8902 1.8912		
	40 47		55EE	S460N S460NL	1.8901 1.8903		
	40 47			S275M S275ML	1.8818 1.8819		EN 10 113-3 (1993)
	40 47			S355M S355ML	1.8823 1.8834		
	40 47			S420M S420ML	1.8825 1.8836		
	40 47			S460M S460ML	1.8827 1.8838		

Table T.9.6.c - Quality groups

Quality group	Impact energy according to EN 10045-1 J	Test temperature Tc °C	Steels, corresponding to the quality group Designation of steels According to old standard	According to EN 10027-1 & ECISS IC 10	According to EN 10027-2	Old standard	New standard
4	30 40 50	- 20	S 460 T	S460Q S460QL S460QL1	1.8908 1.8906 1.8916	NFA 36-204 (1992)	EN 10137-2 (1995)
	30 40 50		S 500 T	S500Q S500QL S500QL1	1.8924 1.8909 1.8984		
	30 40 50		S 550 T	S550Q S550QL S550QL1	1.8904 1.8926 1.8986		
	30 40 50		S 620 T	S620Q S620QL S620QL1	1.8914 1.8927 1.8987		
	30 40 50		S 690 T	S690Q S690QL S690QL1	1.8931 1.8928 1.8988		
	30 40 50			S890Q S890QL S890QL1	1.8940 1.8983 1.8925		
	30 40		S 960 T	S960Q S960QL	1.8941 1.8933		
	30 40 50		TStE 460 V	S460Q S460QL S460QL1	1.8908 1.8906 1.8916	(Germany)	EN 10137-2 (1995)
	30 40 50		StE 500 V TStE 500 V EStE 500 V	S500Q S500QL S500QL1	1.8924 1.8909 1.8984		
	30 40 50		StE 550 V TStE 550 V EStE 550 V	S550Q S550QL S550QL1	1.8904 1.8926 1.8986		
	30 40 50		StE 620 V TStE 620 V EStE 620 V	S620Q S620QL S620QL1	1.8914 1.8927 1.8987		
	30 40 50		StE 690 V TStE 690 V EStE 690 V	S690Q S690QL S690QL1	1.8931 1.8928 1.8988		
	30 40 50		TStE 890 V EStE 890 V	S890Q S890QL S890QL1	1.8940 1.8983 1.8925		
	30 40		TStE 960 V	S960Q S960QL	1.8941 1.8933		
	40 50			S500A S500AL	1.8980 1.8990		EN 10137-3 (1995)
	40 50			S550A S550AL	1.8991 1.8992		
	40 50			S620A S620AL	1.8993 1.8994		
	40 50			S690A S690AL	1.8995 1.8996		

Table T.9.6.d - Quality groups

Quality group	Impact energy according to EN 10 045-1 J	Test temperature Tc °C	Steels, corresponding to the quality group Designation of steels			Old standard	New standard
			According to old standard	According to EN 10027-1 & ECISS IC 10	According to EN 10027-2		
			E 315 D	S315MC	1.0972		
			E 355 D	S355MC	1.0976		
			E 420 D	S420MC	1.0980		
				S460MC	1.0982		
				S500MC	1.0984	NF A 36-231 (1992)	
			E 560 D	S550MC	1.0986		
				S600MC	1.8969		
				S650MC	1.8976		
			E 690 D	S700MC	1.8974		
			QStE 300 TM	S315MC	1.0972		
			QStE 360 TM	S355MC	1.0976		
			QStE 420 TM	S420MC	1.0980		
			QStE 460 TM	S460MC	1.0982		
			QStE 500 TM	S500MC	1.0984	SEW 092	EN 10149-2 (1995)
			QStE 550 TM	S550MC	1.0986		
			QStE 600 TM	S600MC	1.8969		
			QStE 650 TM	S650MC	1.8976		
4	40	- 20	QStE 690 TM	S700MC	1.8974		
			43F35	S315MC	1.0972		
			46F40	S355MC	1.0976		
			50F45	S420MC	1.0980		
				S460MC	1.0982	(United Kingdom)	
				S500MC	1.0984		
			60F55	S550MC	1.0986		
				S600MC	1.8969		
				S650MC	1.8976		
			75F70	S700MC	1.8974		
			QStE 260 N	S260NC	1.0971		
			QStE 300 N	S315NC	1.0973	SEW 92-75	
			QStE 360 N	S355NC	1.0977		
			QStE 420 N	S420NC	1.0981		EN 10149-3 (1995)
				S260NC	1.0971		
			40/30	S315NC	1.0973	(United Kingdom)	
			43/35	S355NC	1.0977		
				S420NC	1.0981		

9.7 STRUCTURAL MEMBERS OTHER THAN JOINTS - PERMISSIBLE STRESSES

Clause 3.2.1.1.of booklet 3 may be replaced by the following text:

Table T.9.7.a - Values for f_y, f_u, and σ_a for non alloy, fine grain structural steels and such in the quenched and tempered conditions

Standard	Steel	Thickness t mm	Yield stress f_y N/mm^2	Ultimate stress f_u N/mm^2	Permissible stresses: σ_a Case I N/mm^2	Case II N/mm^2	Case III N/mm^2
	S235	≤ 16	235	340	157	177	214
	(Fe360)	≤ 40	225	340	150	169	205
EN 10025		≤ 100	215	340	143	162	195
		≤ 150	195	340	130	147	177
		≤ 200	185	320	123	139	168
		≤ 250	175	320	117	132	159
	S275	≤ 16	275	410	183	207	250
	(Fe440)	≤ 40	265	410	177	199	241
		≤ 63	255	410	170	192	232
		≤ 80	245	410	163	184	223
		≤ 100	235	410	157	177	214
		≤ 150	225	400	150	169	205
		≤ 200	215	380	143	162	195
		≤ 250	205	380	137	154	186
EN 10025	S355	≤ 16	355	490	237	267	323
and	(Fe510)	≤ 40	345	490	230	259	314
EN 10113	S355N and	≤ 63	335	490	223	252	305
	S355NL	≤ 80	325	490	217	244	295
	steels up	≤ 100	315	490	210	237	286
	to t ≤150	≤ 150	295	470	197	222	268
		≤ 200	285	450	190	214	259
		≤ 250	275	450	183	207	250
EN 10113	S460	≤ 16	460	550	307	346	418
		≤ 40	440	550	293	331	400
		≤ 63	430	550	287	323	391
		≤ 80	410	550	273	308	373
		≤ 100	400	550	267	301	364
EN 10137	S460	≤ 50	460	550	307	346	418
		≤ 100	440	550	293	331	400
		≤ 150	400	500	267	301	364
	S690	≤ 50	690	770	460	519	627
		≤ 100	650	760	433	489	591
		≤ 150	630	710	420	474	573
	S890	≤ 50	890	940	593	669	809
		≤ 100	830	880	553	624	755
	S960	≤ 50	960	980	640	722	873

NOTE 1: The yield stress f_y and the permissible stress σ_a of the hot finished structural hollow sections according to EN 10210-1 comply with those in Table T.9.7, $t \leq 65$ mm, for grades 235 to 460.

NOTE 2: The yield stress f_y and the permissible stress σ_a of the cold formed welded structural hollow sections according to EN 10219-1 comply with those in Table T.9.7, $t \leq 40$ mm, for grades 235 to 460.

Table T.9.7.b - Values for f_y, f_u, and σ_a for high yield strength steels for cold forming and hollow sections

Standard	Steel	Thickness t mm	Yield stress f_y N/mm^2	Ultimate stress f_u N/mm^2	Permissible stresses: σ_a Case I N/mm^2	Case II N/mm^2	Case III N/mm^2
EN 10149	S315	all t	315	390	210	237	286
	S355		355	430	237	267	323
	S420		420	480	280	316	382
	S460		460	520	307	346	418
	S500		500	550	333	376	455
	S550		550	600	367	414	500
	S600		600	650	400	451	545
	S650	≤ 8	650	700	433	489	591
		> 8	630	700	420	474	573
	S700	≤ 8	700	750	467	526	636
		> 8	680	750	453	511	618
EN 10219-1	S420MH and MLH	≤ 16	420	500	280	315	382
		≤ 40	400	500	267	300	363

9.8 JOINTS MADE WITH TENSION BOLTS WITH CONTROLLED TIGHTENING

Clause 3.2.2.2.1.of booklet 3 may be replaced by the following text:

For the calculation developed in clause 3.2.2.2.1, other recommendations or standards (for example : VDI 2230, FDE 25030, or the work of CEN/TC 185/WG 7) can be used.

However, different methods may not be mixed.

Tests (for example : extensometric) can complete and/or replace the calculations.

9.9 CHECKING MEMBERS SUBJECT TO CRIPPLING

Clause 3.3.of booklet 3 may be replaced by the following text:

The method presented in ENV 1993-1 :1992 Eurocode 3 :
Design of steel structures Part 1.1 may be used.

9.10 CHECKING MEMBERS SUBJECT TO BUCKLING

Clause 3.4 of booklet 3 may be replaced by the following text:

In determining the buckling safety coefficients, stated below, it was considered that flat plates under compressive stresses equally distributed over the plate width, are exposed to a greater danger of buckling than plates under stresses changing from compression to tension over the plate width.

In consequence, safety against buckling was made dependent on the ratio ψ of stresses at the plate edges (appendix A-3.4. of booklet 3).

It shall be verified that the calculated stress is not higher than the critical buckling stress divided by the following coefficients η_V given by table T.9.10:

Table T.9.10

	Case	Buckling safety η_V
Buckling of plane members	I	$1{,}70 + 0{,}175\ (\psi - 1)$
	II	$1{,}50 + 0{,}125\ (\psi - 1)$
	III	$1{,}35 + 0{,}075\ (\psi - 1)$
Buckling of curved members : Circular cylinders (e.g. tubes)	I	1,70
	II	1,50
	III	1,35

The edge-stresses ratio ψ varies between + 1 and - 1.

Appendix A.3.4.of booklet 3 gives the procedure for determining the critical buckling stress.

Checking members subjected to buckling can be carried out according to other recommendations , for example ENV 1993-1.

ENV 1993-1 is based on limit state analysis : partial safety factors γ_F and γ_M are used.

9.11 CASE OF STRUCTURES SUBJECTED TO SIGNIFICANT DEFORMATION

Clause 3.5.of booklet 3 may be replaced by the following text:

9.11.1 Non-proportional effect on the structure due to the forces

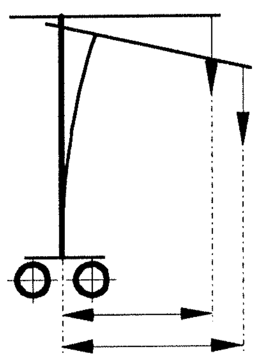


Figure F.9.11.a

In this case the stresses in the members cannot be proportional to the forces which cause them due to the deformation of the structure as a result of the application of these forces.

This is the case, for example, with the stresses produced in the column of a crane (illustrated in figure F.9.11.a) where it is clear that the moment in the column is not proportional to the forces applied because of deformations that increase their moment arm.

In this case the calculation can be carried out :
- either by using the limit states method;
- or by using the method described in clause 3.5 of booklet 3.

Limit states method

The figure F.9.11.b shows the limit states method :

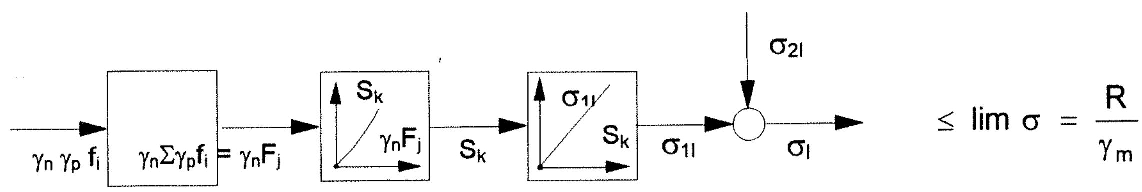


Figure F.9.11.b - Typical flow chart of the limit state method

- f_i is the load i on the element or component;
- F_j is the load combination j from loads f_i, multiplied by partial load coefficients and risk coefficient, when applicable;
- S_k are the load effects in section k of members or supporting parts, such as inner forces and moments, resulting from load combination F_j;
- σ_{1l} are the stresses in the particular element l as a result of load effects S_k;
- σ_{2l} are the stresses in the particular element l arising from local effects;
- σ_l is the resulting design stress in the particular element l;
- R is the specified strength or characteristic resistance of the material, particular element or connection, such as the stress corresponding to the yield point, limit of elastic stability or fatigue strength (limit states);

lim σ is the limit design stress;

γ_P are the partial load coefficients applied to individual loads according to the load combination under consideration;

γ_n is the risk coefficient, where applicable;

γ_m is the resistance coefficient.

NOTE 1: Instead of a comparison of stresses, as mentioned above, a comparison of forces, moments, deflections, ets. may be made.

NOTE 2: A general description of the limit state, method of design is given in ISO 2394 : 1986, *General principle on reliability for structures.*

The individual specific f_i loads are calculated according to the data in booklet 2. They are multiplied by the appropriate partial coefficients of load. Then they are combined according to the combinations given in clause 2.3 of booklet 2.

Table T.9.11 - Partial coefficients γ_p

Loads		Clause	Loading condition (see 2.3)		
			Case I	Case II	Case III
Dead loads	unfavourable effect	2.2.1	1,22	1,16	1,10
	Favourable effect - estimated weight		0,90	0,95	1,00
	Favourable effect - measured weight		1,00	1,00	1,00
Working loads		2.2.2.1	1,34	1,22	
Acceleration from drives		2.2.3	1,34	1,22	1,10
Effects of climate		2.2.4		1,16	1,10

with $\gamma_m = 1{,}10$

For checking erection or dismanling, the case III is appropriate.

NOTE: The values given in table 10 of the document pr EN 13001-2 may be accepted.

9.11.2 Non linear structures and favourable effects of own weight

Clause 3.5 of Booklet 3 describes a corrective method for proof of competence calculations in cases of structures subjected to significant deformation. However, the significant deformations are not the only cases where the designer shall consider the use of similar correction.

By combining clauses 2.3.1 and 3.2.1 or 3.4 the conditions for calculated stress in load combination Case I can be expressed as follows:

$$\sigma \{\gamma_c (S_G + \psi\, S_L + S_H)\} \leq \sigma_{cr} / \nu$$

where

σ_{cr} is the yield stress, crippling stress or buckling stress, whichever is the most critical one,

ν is the relevant coefficient ν_E or ν_V.

The above formula can also be presented as:

$$\nu\sigma \{\gamma_c (S_G + \psi S_L + S_H)\} \leq \sigma_{cr}$$

for any structural system.

If the structural system is behaving nearly linearly, the above formula can be modified as follows:

$$\sigma = k_g\, \nu\, \gamma_c\, S_G + k_L\, \nu\, \gamma_c\, \psi\, S_L + k_h\, \nu\, \gamma_c\, S_H \leq \sigma_{cr}$$

where coefficients k_g, k_L, and k_h represent the linear relationships between the load effects (S_G, S_L, S_H) and the calculated stress. Those coefficients depend on the configuration and type of loading of the crane.

<u>A simplified example:</u> *A simply supported beam,with span l and section modulus W, loaded by its own weight mg (S_G) and in the middle by force F (produced by the load effects S_L) . The bending stress is calculated by the formula:*

$$\sigma = \frac{\nu \cdot \gamma_C \cdot mg \cdot l}{8 \cdot W} + \frac{\nu \cdot \gamma_C \cdot \psi \cdot F \cdot l}{4 \cdot W}$$

where it is seen that $k_g = \dfrac{l}{8 \cdot W}$ and $k_L = \dfrac{l}{4 \cdot W}$

In order to check the most critical effect of all the loads for a particular design detail, it is evident that the signs for the variable loads shall be selected so that they lead to the maximum combined stress (if such a combination is physically possible). Furthermore, the loads are multiplied by coefficients taking into account dynamic effects and an adequate margin for failure.

However, in a case where the dead weight S_G decreases the absolute value of the stress due to the variable loads (S_G having an opposite, favourable stress effect) the multiplication of the dead weight by its coefficients would lead to an situation where the actual margin for the critical stress might be dangerously reduced. To maintain the intended margin for failure, the calculation method described in clause 3.5 shall be applied in the following cases:

1. - When the dead weight has an effect in the opposite direction to the effect of the variable loads, i.e. the dead weight has a balancing effect. Examples: towers and lower structures of slewing jib cranes and tower cranes.

2. - Especially for structures where the dead weight has an effect in the opposite direction to the effect of the variable loads and the tension and compression forces are carried by different structural elements.

   Example 1: Tie-downs for storm anchoring take the difference between upwards and downwards loads while the wheels carry all the downwards loads in the case of opposite wind action.

   Example 2: Bogie pins and the ties of pin joints (such as end cups). The pins carry all the compression while the ties carry the difference between upwards and downwards loads.

3. - Pre-stressed structures.

   Example: Bolted flange joints.

In this case the variable loads shall be multiplied by the coefficient ν, but the pre-stress loads shall be taken as the lowest estimated nominal values. The coefficient for the dead weight shall be selected between 1 and ν depending on whether its effect is favourable or unfavourable for the bolts.

9.12 CHOICE OF RAIL WHEELS

Clause 4.2.4.of booklet 4 may be replaced by the following text:

The method proposed in clause 4.2.4.1 can be used with the values of P_L and c_2 given below tables T.9.12.a and T.9.12.b:

Table T.9.12.a - Values of P_L

Ultimate strength of metal used for rail wheel N/mm²	P_L N/mm²	Minimum strength for the rail N/mm²
$f_u > 500$	5,00	350
$f_u > 600$	5,60	350
$f_u > 700$	6,50	510
$f_u > 800$	7,20	510
$f_u > 900$	7,80	600
$f_u > 1000$	8,50	700

Table T.9.12.b - Values of c_2

Group classification of mechanism	c_2
M1 and M2	1,25
M3 and M4	1,12
M5	1,00
M6	0,90
M7 and M8	0,80

The hardening of the running surface at the depth of 0,01D may be taken into account when selecting the value of P_L

When using the tables above, it is not necessary to consider the 5 last paragraphs of clause 4.2.4.1.3 in booklet 4.

9.13 DESIGN OF GEARS

Clause 4.2.5.of booklet 4 may be replaced by the following text:

Standards or calculation methods such as follows may be used, for example:

- NF E 23015, Henriot method;
- DIN 3990;
- ISO 6336.

For the design calculation for gears, the coefficient γ_m is not cumulative with the service factor (k_a). However, it shall be at least equal to γ_m.

9.14 DETERMINATION OF PERMISSIBLE STRESSES IN MECHANISM COMPONENTS SUBJECTED TO FATIGUE

Clauses 2.1.4.3, 4.1.3.5, 4.1.3.6, 4.1.3.7 and appendix A 4.1.3 of booklets 2 and 4 may be replaced by the following text:

9.14.1 Introduction

The calculation methods for determining the fatigue strength of mechanism components are similar in the documents of section I (FEM 1.001 edition 1987) and section II (FEM 2.131 and FEM 2.132 edition 1992)

In the above-mentioned editions, the Wöhler curve of a component includes a second slope (factor c') for the number of cycles n greater than 2.10^6:

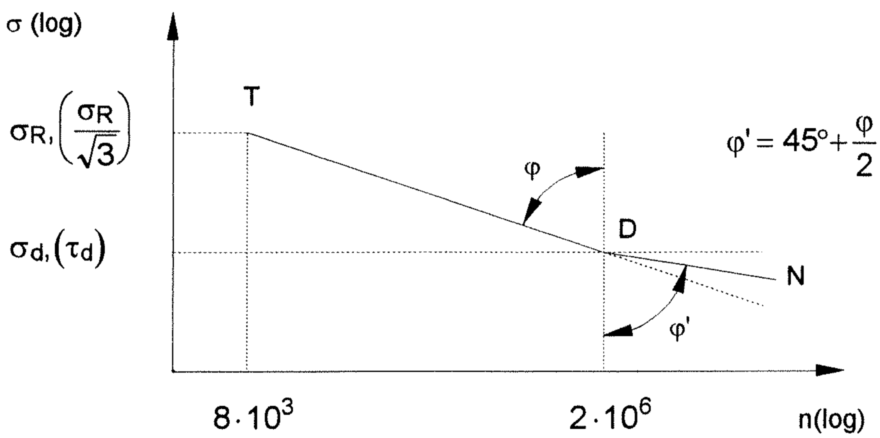


Figure F.9.14.a

The presence of this second slope results in the determination of very low values for fatigue strength for very heigh number of cycles n, and consequently the safety level is too high.

The text below proposes in particular the cancelling of the second slope of the Wöhler curve is presented below.

9.14.2 Partial modifications of booklet 2 and 4

BOOKLET 2 MODIFIED

NOTE: The modifications are written in bold face type.

2.1.4.3 STRESS SPECTRUM

....In many applications the function f(x) may be approximated by a function consisting of a certain number r of steps, comprising respectively n_1, n_2,... n_r stress cycles ; the stress σ may be considered as practically constant and equal to σ_i during n_i cycles. If n represents **the total number of cycles** and σ_{max} the greatest of the stresses σ_1, σ_2, ..., σ_r there exists a relation :

$$n_1 + n_2 + \ldots + n_r = \sum_{i=1}^{r} n_i = n$$

and

$$\sigma_1 > \sigma_2 > \ldots > \sigma_r$$

we obtain an approximated form :

$$k_{sp} = \left(\frac{\sigma_1}{\sigma_{max}}\right)^c \frac{n_1}{n} + \left(\frac{\sigma_2}{\sigma_{max}}\right)^c \frac{n_2}{n} + \ldots.. + \left(\frac{\sigma_r}{\sigma_{max}}\right)^c \frac{n_r}{n} = \sum_{i=1}^{r} \left(\frac{\sigma_i}{\sigma_{max}}\right)^c \frac{n_i}{n}$$

the summation is truncated for the first $n_i \geq 2.10^6$. This n_i is taken as n_r and replaced with $n_r = 2.10^6$ cycles.

Depending on its stress spectrum, a component is placed in one of the spectrum classes P1, P2, P3, P4, defined in table T.2.1.4.3.

BOOKLET 4 MODIFIED

NOTE: The modifications are written in bold face type.

4.1.3.5 WÖHLER CURVE

In this context, the Wöhler curve, shows the number of stress cycles n which can be withstood before fatigue failure as a function of the maximum stress σ (or τ), when all stress cycles present the same amplitude and the same ratio k between extreme values.

With regard to this WÖHLER curve, the following hypotheses are made respectively

- for $n = 8.10^3$:

$$\sigma = \sigma_R$$

or

$$\tau = \frac{\sigma_R}{\sqrt{3}}$$

- for $8 \cdot 10^3 \leq n \leq 2 \cdot 10^6$, the area of limited endurance, the function is represented by a straight line TD in a reference comprising two logarithmic scale axes (figure 4.1.3.5 modified).

  The slope of the WÖHLER curve, in the interval considered, is characterized by the factor :

$$c = \tan \varphi = \frac{\log 2 \cdot 10^6 - \log 8 \cdot 10^3}{\log \sigma_R - \log \sigma_d}$$

or

$$c = \tan \varphi = \frac{\log 2 \cdot 10^6 - \log 8 \cdot 10^3}{\log \frac{\sigma_R}{\sqrt{3}} - \log \tau_d}$$

- for $n > 2.10^6$:

$$\sigma = \sigma_d$$

or

$$\tau = \tau_d$$

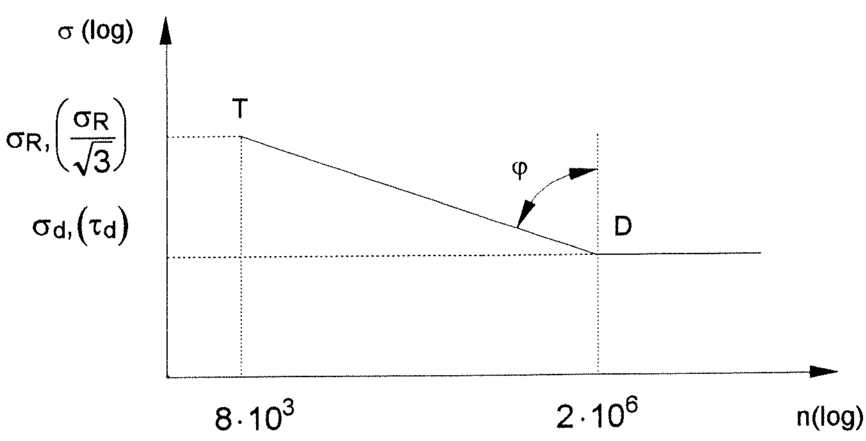


Figure 4.1.3.5 modified

The spectrum factor k_{sp} of the component is determined by means of the above mentioned value of c.

4.1.3.6 FATIGUE STRENGTH OF A MECHANICAL COMPONENT

The fatigue strength σ_k or τ_k of a given mechanical component is determined by the following expressions respectively :

$$\sigma_k = \left(2^{\frac{8-j}{c}}\right) \cdot \sigma_d$$

or

$$\tau_k = \left(2^{\frac{8-j}{c}}\right) \cdot \tau_d$$

where j is the component's group number.

The group classification of components, on the basis of their total number of cycles n and their spectrum factor k_{sp}, as well as the critical fatigue stresses associated with each group, are illustrated in figure 4.1.3.6 modified where σ_{jk} represents the stress applying to group E_j. For the critical shear stresses, the letter σ must be replaced with τ.

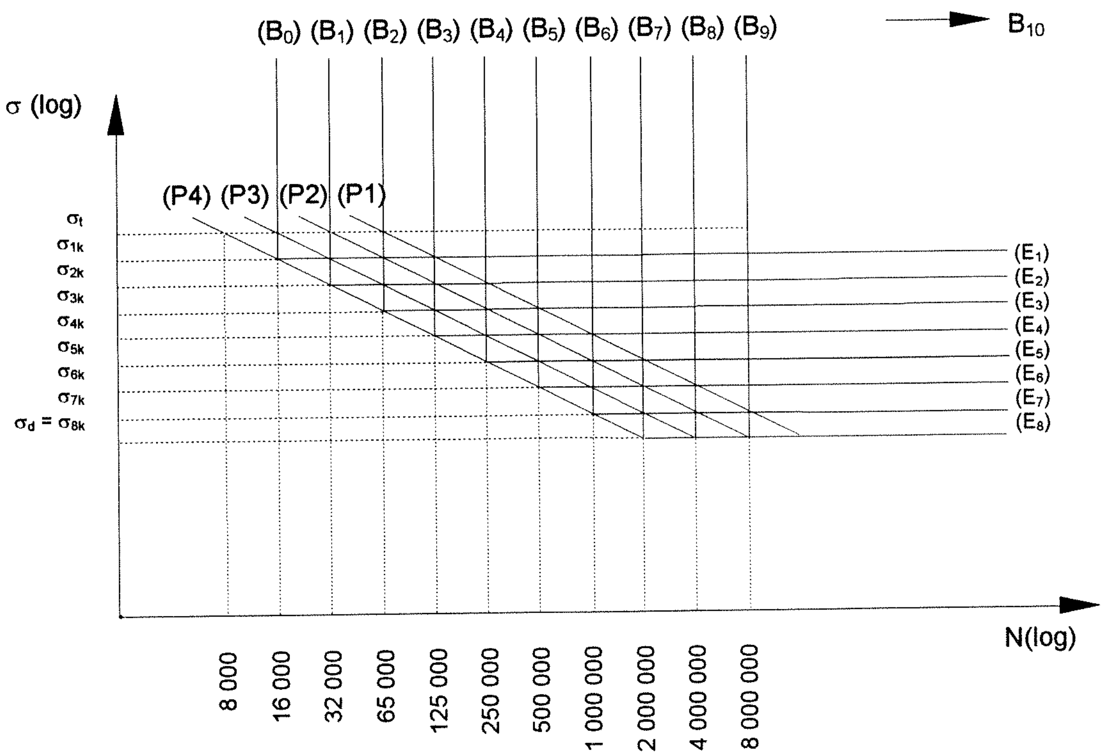


Figure 4.1.3.6 modified

Concerning the relation between spectrum classes P1 to P4 and the spectrum factor k_{sp} see table T.2.1.4.3 in booklet 2.

<u>Comment</u> : The above fatigue strengths are based on the component's group number so those values are discontinuous. The above formulas may be usefully replaced with the following:

$$\sigma_k = \frac{\sigma_d}{\left(k_{sp} \cdot \frac{n}{n_d}\right)^{1/c}} = \frac{\sigma_d}{\left(k_{sp} \cdot \frac{n}{2 \cdot 10^6}\right)^{1/c}} \quad \text{or}$$

$$\tau_k = \frac{\tau_d}{\left(k_{sp} \cdot \frac{n}{n_d}\right)^{1/c}} = \frac{\tau_d}{\left(k_{sp} \cdot \frac{n}{2 \cdot 10^6}\right)^{1/c}}$$

See example in 9.14.3.

BOOKLET 4 MODIFIED

NOTE: The modifications are written in bold face type.

4.1.3.7 PERMISSIBLE STRESSES AND CALCULATIONS

The permissible stresses σ_{af} and τ_{af} are obtained by dividing the stresses σ_k and τ_k, defined in 4.1.3.6., respectively by a safety factor of ν_k.

Taking :

$$\nu_k = 3{,}2^{\frac{1}{c}}$$

σ_{af} and τ_{af} will be obtained by the relations :

$$\sigma_{af} = \frac{\sigma_k}{\nu_k}$$

$$\tau_{af} = \frac{\tau_k}{\nu_k}$$

and it is verified that :

$$\sigma \leq \sigma_{af}$$

$$\tau \leq \tau_{af}$$

with : σ maximum calculated normal stress **amplitude**,
τ maximum calculated shear stress **amplitude**.

In the case of components acted upon simultaneously by normal stresses and shear stresses with different ratios κ between extreme stresses, the following condition must be satisfied :

$$\left(\frac{\sigma_x}{\sigma_{kx}}\right)^2 + \left(\frac{\sigma_y}{\sigma_{ky}}\right)^2 - \left(\frac{\sigma_x \sigma_y}{|\sigma_{kx}| \cdot |\sigma_{ky}|}\right) + \left(\frac{\tau}{\tau_k}\right)^2 \leq \frac{1{,}1}{\nu_k^2}$$

in which :

σ_x, σ_y	= maximum normal stresses in the directions x and y respectively,
τ	= maximum shear stress,
σ_{kx}, σ_{ky}	= fatigue strength for normal stresses, in the directions x and y respectively,
τ_k	= shear fatigue strength.

If it is not possible to determine the most unfavourable case of the foregoing relation from the corresponding stresses σ_x, σ_y and τ, calculations must be performed separately for the loads $\sigma_{x\,max}$, $\sigma_{y\,max}$ and τ_{max} and the most unfavourable corresponding stresses.
It should be noted that the checks described above do not guarantee safety against brittle fracture.

Such safety can be ensured only by a suitable choice of material quality.

BOOKLET 4 MODIFIED

APPENDIX A-4.1.3 - DETERMINATION OF PERMISSIBLE STRESSES IN MECHANISM COMPONENTS SUBJECTED TO FATIGUE

NOTE: The modifications are written in bold face type.

The endurance limit for a polished specimen is a laboratory value, which is practically never attained in parts actually used. Numerous factors - shape, size, surface condition (machining quality) and possible corrosion - induce discontinuities resulting in "notch effects", which decrease the permissible stresses in the part, when these stresses are calculated by conventional elementary methods for the strength of materials. These factors are taken into account by coefficients, called k_s, k_d, k_u, k_c, respectively all greater than or equal to unity.The endurance limit fora polished specimen isdivided by the product of theses coefficients.

Designers are advised against using a skin factor taking into account the influence of surface treatments.

...

The calculation for permissible stresses for fatigue can also be constructed with the stress gradient method (or Siebel method), which takes into account plastic adaptation with notch root ..

This method is used in the following documents :

- "Handbuch für Werkstoffprüfung", E. SIEBEL, Berlin 1958,

- "Calcul des pièces à la fatigue - Méthode du gradient", A. BRAND, CETIM 1980,

- FKM Forschungskuratorium Maschinenbau e. V. (Hrsg.) : Festigkeitsnachweis. Vorhaben Nr. 154, FKM-Heft 183-1Frankfurt 1994,

- E DIN 743 : Tragfähigkeit von Wellen und Achsen. Teile 1-4, Beuth-Verlag, Berlin, April 1996

9.14.3 Checking for fatigue of a mechanical component - example

We consider a shaft whose initial stress spectrum is as given in table T.9.14:

Table T.9.14

Level	σ_i (N/mm²)	σ_i/σ_{max}	n_i (real)	n_i (effective)
1	200	1	10 000	10 000
2	160	0,8	50 000	50 000
3	125	0,625	200 000	200 000
4	90	0,45	1 500 000	1 500 000
5	80	0,4	5 000 000	2 000 000
6	71	0,355	20 000 000	0
7	63	0,315	50 000 000	0
				$n = \sum n_i = 3\,760\,000$

According to booklet 2 Classification of the component

$$n = 3{,}76.10^6$$

It belongs to the class of utilization **B8** (clause 2.1.4.2).

c = 3 (slope of the Wöhler curve for the component)

We calculate the spectrum factor k_{sp} (clause 2.1.4.3):

$$k_{sp} = 1 \cdot \frac{10^4}{3{,}76 \cdot 10^6} + 0{,}8^3 \cdot \frac{5 \cdot 10^4}{3{,}76 \cdot 10^6} + 0{,}625^3 \cdot \frac{2 \cdot 10^5}{3{,}76 \cdot 10^6} + 0{,}45^3 \cdot \frac{1{,}5 \cdot 10^6}{3{,}76 \cdot 10^6} + 0{,}4^3 \cdot \frac{2 \cdot 10^6}{3{,}76 \cdot 10^6}$$

$$= 0{,}0026 + 0{,}006809 + 0{,}012986 + 0{,}036353 + 0{,}034043$$

$$= 0{,}09285$$

It belongs to the spectrum class **P1** and, consequently, to the component group **E6** (clause 2.1.4.4).

According to booklet 4 Checking for fatigue

The endurance limit of the component is : σ_d = 100 N/mm^2 (clause 4.1.3.4).

The fatigue strength of the shaft is (clause 4.1.3.6):

$$\sigma_k = 2^{\left(\frac{8-j}{c}\right)} . \sigma_d = 2^{\left(\frac{8-6}{3}\right)} .100 = 158 \text{ N / mm}^2$$

The safety factor is (clause 4.1.3.7):

$$\nu_k = 3.2^{1/c} = 3.2^{1/3} = 1.473$$

The permissible stress of the shaft is :

$$\sigma_{af} = \frac{\sigma_k}{\nu_k} = \frac{158}{1{,}473} = 107{,}3 \text{ N/mm}^2$$

The maximum calculated stress is :

$$\sigma = 200 \text{ N/mm}^2$$

$$\sigma = 200 \text{ N/mm}^2 > \sigma_{af} = 107.3 \text{ N/mm}^2$$

The shaft is not acceptable for fatigue, because the max stress amplitude is higher than the permissible value.

NOTE: If we use the fatigue strength formula proposed in the comment of the clause 4.1.3.6, it becomes :

$$\sigma_k = \frac{\sigma_d}{\left(k_{sp} \cdot \frac{n}{2 \cdot 10^6}\right)^{1/c}} = \frac{100}{\left(0{,}09285 \cdot \frac{3760000}{2000000}\right)^{1/3}} = 178{,}9 \text{ N/mm}^2$$

The permissible stress of the shaft is :

$$\sigma_{af} = \frac{\sigma_k}{\nu_k} = \frac{178{,}9}{1{,}473} = 121{,}5 \text{ N/mm}^2$$

The maximum calculated stress is :

$$\sigma = 200 \text{ N/mm}^2$$

$$\sigma = 200 \text{ N/mm}^2 > \sigma_{af} = 121.5 \text{ N/mm}^2$$

The shaft is still not acceptable for fatigue, because the max stress amplitude is higher than the permissible value.

9.15 STABILITY AND SAFETY AGAINST MOVEMENT BY WIND

The following text replaces booklet 6

9.15.1 Scope

These requirements specifiy the conditions to be met when verifying, by calculation, the stability of cranes that are subject to tipping and drifting; it assumes that the cranes are standing on a firm, level supporting surface or track.

> NOTE: Where the crane is required to operate on an inclined surface, the manufacturer shall take the specified conditions into account.

9.15.2 Stability - Calculations

9.15.2.1. A crane is said to be stable when the algebraic sum of the stabilizing moments is greater than or equal to the sum of the overturning moments.

9.15.2.2 Calculations shall be made to verify the stability of the crane by computing the sum of the overturning moments and the stabilizing moments using the loads multiplied by the load factor given in table T.9.15.a

In all calculations, the position of the crane and its components, and the effect of all loads and forces, shall be considered in their least favourable combination, direction and effect.

9.15.2.3 For cranes designed to travel with load, the forces induced by the maximum allowable vertical track variation as specified by the manufacturer shall be taken into account, in addition to other loads specified in condition II of table T.9.15.a.

9.15.2.4 Where required, excitation effects appropriate to the particular site or zone shall be considered as an additional loading condition.

9.15.2.5 In the calculations shown in table T.9.15.a, consideration shall be given to the loads induced by the weight of the crane and its components, including any lifting attachments which are a permanent part of the crane in its working condition.

9.15.2.6 For the case of collision (e. g. buffer impact), the stability calculations shall be based on dynamic considerations.

9.15.2.7 For tower cranes the stability case according to table T.9.15.b shall be met.

Table T.9.15.a.

Condition	Loading	Load factor to be considered
I. Basic Stability	Loads induced by the dead weight	1,0
	Applied load	1,6P
	Wind load	0
	Inertia forces*	0
II. Dynamic Stability	Loads induced by the dead weight	1,0
	Applied load	1,35 P
	Wind load	1,0 W1
	Inertia forces	1,0 D
III. Backward Stability (Sudden release of load)	Loads induced by the dead weight	1,0
	Applied load	-0,2 P
	Wind load	1,0 W1
	Inertia forces	0
IV. Extreme Wind Loading	Loads induced by the dead weight	1,0
	Applied load	1,0 P1
	Wind load	1,2 W2
	Inertia forces	0
V. Stability During Erection or dismantling	Loads induced by the dead weight	1,0
	Applied load	1,25 P2
	Wind load	1,0 W3
	Inertia forces	1,0 D

Where :

- D are the inertia forces from drives
- P is the net load
- P1 is the fixed load lifting attachment; out-of-service the fixed load lifting attachment shall be considered as part of the weight of the crane and its components
- P2 is the weight of the part being installed/removed during erection or dismantling
- W1 is the in-service wind effect
- W2 is the out-of-service wind effect - gusting effects are included
- W3 is the in-service wind effect W1 or the effect of the wind limit for erection work in accordance with the instruction handbook of the manufacturer

Table T.9.15.b.

Condition	Loading	Load factor to be considered
VI Stability During Erection or dismantling see figure F.9.15.	Loads induced by the dead weight	1,0
	Horizontal applied load	0,10 P2
	Vertical applied load	1,16 P2
	Wind load	1,0 W3
	Inertia forces	1,0 D

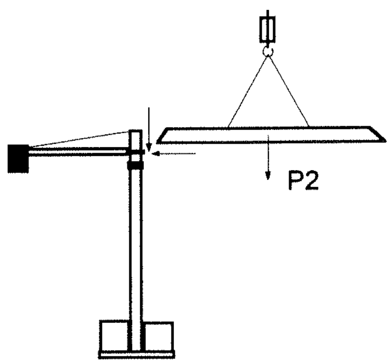


Figure F.9.15 - Example: application of a load P2 for fitting a jib

9.15.3 Backward stability in service conditions

The backward stability is covered by condition III.

9.15.4 Application of wind loads

9.15.4.1 On-service wind forces shall always be applied in the least favourable direction.

9.15.4.2 Out-of-service wind forces shall be applied in the least favourable direction for those cranes which are not free to rotate with the wind. For those cranes which are designed to rotate with the wind, the force shall be applied on the superstructure in the direction contemplated, and in the least favourable direction on the lower structure.

9.15.5 Crane base

The crane manufacturer shall specify the forces imposed by the crane on the ground or supporting structure. The information given by the manufacturer should state all applicable conditions for which the forces have been stipulated (including out-of-service wind). Where the crane base provides all or part of the stability of the crane, the manufacturer shall specify the requirement applicable to the crane base.

9.15.6 emporary additional stability devices

Tower cranes shall be stable in their operating configuration (condition I to IV in table T.9.15.a) without use of temporary additional devices.

Temporary additional devices may be used to satisfy condition V in table T.9.15.a, erection or dismantling.

Detachable ballast may be used to satisfy the case in condition IV of table T.9.15.a. However, this condition shall be met without this extra ballast, using a factor of 1,1 W2.

9.15.7 Deformation

Where it can be shown that with the most unfavourable loads for the most destabilizing configuration, and consideration given to deformation (Second Order Theory), the effect on the overturning moment does not exceed 10%.Then stability calculations may be carried out ignoring deflections (First Order Theory) for ease of calculation.

However, when this is the case, the overturning moments for each condition in table T.9.15.a shall be increased by the value obtained above according to the second order.

9.15.8 Resistance to drifting caused by wind

The resistance to drifting caused by wind shall be proven by calculation for all cranes on rails operating in the open air under the conditions in table T.9.15.c.

Table T.9.15.c - Drifting caused by wind

Condition	Loading	Load factor to be considered
1.- IN SERVICE	Loads induced by the dead weight	1,0
	Applied load	1,35 P
	Wind load	1,2 W1
	Inertia forces	1,0 D
2.- OUT OF SERVICE	Loads induced by the dead weight	1,0
	Applied load	1,0 P1
	Wind load	1,2 W2
	Inertia forces	0

Where rail clamps or similar measures are necessary to avoid out-of-service drifting, the operator's manual shall advise that they shall be applied when the in-service wind limit has been reached.

The resistance to travel due to friction and the coefficients of friction shown in table T.9.15.d shall apply.

Table T.9.15.d - Resistance to travel and coefficients of friction

Ratio : Resistance to travel / Radial load		Coefficient of friction between track and	
Plain bearings	Anti friction bearings	the braked wheel	the rail clamp
0,02	0,005	0,14	0,25

NOTE: Higher coefficients of friction may be allowed for if it can be shown that these are present in all surface conditions and qualities (e.g. oil, dirt, ice).

9.16 TESTS

Clause 8.1.of booklet 8 may be replaced by the following text:

Prior to being in service, appliances must be tested under overload conditions, as follows :

The cranes shall be tested dynamically, using the maximum nominal speed for each drive movement and overload that is not less than that obtained by multiplying the rated load by coefficient ρ given in table T.9.16:

Table T.9.16 - Values of ρ dynamic test coefficient

Last (t)	$\psi \leq 1,2$	$\psi \leq 1,4$	$\psi > 1,4$
≤ 30	1,2	1,25	1,3
≤ 100	1,15	1,2	1,25
< 100	1,10	1,15	1,2

NOTE: These values are not applicable for cranes equipped with powered series hoist mechanisms with a direct action lifting force limiter. In that case the values given in FEM 9.751 may be used.

where ψ = dynamic coefficient according clause 9.3.

For cranes this dynamic test also covers the requirements of static overload and stability testing.

9.17 TORANCES OF CRANES AND TRACKS

Clause 8.2.of booklet 8 may be replaced by the following text:

The axes of the wheel bores shall not have an angular deviation greater than α from its theoretical axis, in the horizontal plane, see figure F.9.17

The theoretical axis is the arithmetic mean value of the direction angles of all wheel axes. The values for α are given in table T.9.17 below.

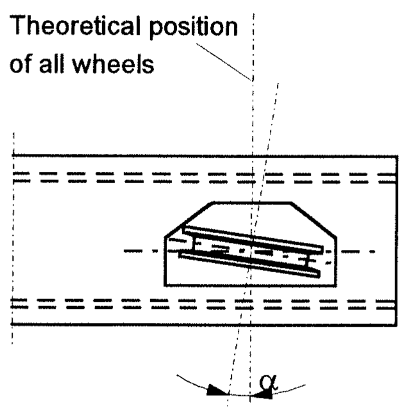


Figure F.9.17

Table T.9.17 - Wheel direction deviation angle α/rad

Class of mechanism	Travel speed v (m/min)				
	≤25	≤50	≤100	≤200	>200
M1	0.0012	0,0012	0,0012	0,0010	0,0008
M2	0,0012	0,0012	0,0010	0,0008	0,0007
M3	0,0012	0,0010	0,0008	0,0007	0,0006
M4	0,0010	0,0008	0,0007	0,0006	0,0005
M5	0,0008	0,0007	0,0006	0,0005	0,0004
M6	0,0007	0,0006	0,0005	0,0004	0,0004
M7	0,0006	0,0005	0,0004	0,0004	0,0004
M8	0,0005	0,0004	0,0004	0,0004	0,0004

NOTE: The angles α give approximately same amount of wear of the wheels and rails, when the wheels are designed according to 9.12.