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SECTION II

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RULES FOR THE DESIGN OF MOBILE EQUIPMENT FOR CONTINUOUS HANDLING OF BULK MATERIALS

1997

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RULES FOR THE DESIGN OF MOBILE EQUIPMENT FOR CONTINUOUS HANDLING OF BULK MATERIALS

DOCUMENT 2 131 / 2 132

GENERAL SUMMARY

edition 1997

Chapter 1

SCOPE AND FIELD OF APPLICATION

CHAPTER 1

SCOPE AND FIELD OF APPLICATION

CONTENTS

1-1 FOREWORD AND GENERAL CONTENTS

The rules for the design of mobile equipment for continuous handling of bulk materials developed by the Technical Committee of FEM Section II have always been widely used in many countries throughout the world.

It should be mentioned that, since its January 1978 edition, the document FEM 2 131 - 01/1978 has been adopted as an ISO international standard under the reference ISO 5049/1. It shall be proposed for this ISO standard a revision to include the corresponding chapters of the present FEM edition.

The revision done in 1997 does not bring fundamental changes to the 1992 edition. The important modifications deal with the following points :
- fatigue calculation of mechanisms,
- friction resistances to define drive mechanisms and braking devices,
- tables describing cases of notch effect for welded structure.

In order to keep the history of the evolution of these rules which apply to the machines defined below in the clause "Scope", it has been indicated below what are the added values of the 1992 edition to the documents :

a) FEM 2 131, edition January 1978 "Rules for the design of mobile continuous bulk handling equipment - chapter I Structures",

b) FEM 2 132, edition June 1977 "Rules for the design of mobile continuous bulk handling equipment - chapter II Mechanisms".

FEM Section II had decided to issue the 1992 edition of these design rules with a threefold objective :

1) to make the periodical revision of the above rules to update them,

2) to add chapters in particular on safety, tests and tolerances,

3) to harmonize them, as far as possible, with the third edition of the design rules issued by FEM Section I in 1988 for the design of lifting appliances.

The following comments can be made on the three objectives which are at the origin of the 1992 edition :

1) Revision of the rules FEM 2 131 - FEM 2 132

   The 1992 periodical revision did not involve fundamental changes, but was an updating which essentially took into account the changes brought in other standards with regard for example to units, welding symbols, etc.

2) Additions to the previous edition

   The 1992 edition had been completed with two chapters covering :
   - safety requirements (chapter 5)
   - tests and tolerances (chapter 6)

   It was planned to add an "Electrical" chapter in a next edition.

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3) Harmonization with the 3rd edition of FEM Section I design rules for lifting appliances

Design departments which have to design both handling equipment (FEM Section II rules) and lifting equipment (FEM Section I rules) have sometimes met difficulties due to a certain lack of consistency between the corresponding rules.

While it should be pointed out that continuous handling equipment and lifting appliances are different with regard to the definition of loads and their combinations, it should be noted, on the other hand, that the method of classification of the machines, of their mechanisms or components, and the calculation of certain elements, should be similar if not identical.

The 1992 edition therefore tries to be in harmony with FEM Section I design rules to the greatest possible extent. Some differences however remain : it may be possible to reduce them later when the results of many studies currently in progress (calculation for fatigue, definition of rail wheels, calculation of wind effects,...) are known.

4) Major changes in the documents FEM 2 131 and 2 132 editions of 1978 and 1977 respectively

It should be stressed that the 1992 version of the design rules for continuous bulk handling equipment does not include any major changes in its content compared to the previous edition which consisted of documents FEM 2 131 and 2 132.

In particular, the definition of the loads applying on machines and the combination of loads have been maintained for the most part.

The principal changes can be summarized as follows :

- Classification of the machines, their mechanisms and components

  Groups have been created to facilitate dialogue between user and manufacturer. As far as the whole machine is concerned, these groups called A2 to A8 are directly based on the total desired duration of utilization.

  Mechanisms can be classified in eight groups called M1 to M8, each group being based on a spectrum class on one hand and a class of utilization (i.e. on a total duration of utilization) on the other hand.

  Structural or mechanism components can be classified in eight groups named E1 to E8, each of them being based, in the same way as mechanisms, on a class of load spectrum and a class of utilization.

- Loads to be taken into account in the calculation of structures

  Clarifications have been made regarding the definition of these loads, in particular on the subject of wind loads. A load case has been added for special situations which may occur for machines during erection.

- Calculating the stresses in structural components

  A method for selecting the steel grade in relation to brittled fracture has been added : the choice is to be made between four quality groups which are distinguished by the impact strength of the corresponding steels.

  The chapter on bolted joints has been reworked and completed.

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The curves giving the permissible fatigue stresses for structural components have been maintained and given in relation to the component classification group.

- Checking and choice of mechanism components

Wire ropes are chosen on the basis of a practical safety factor which depends on the mechanism classification group. The rope breaking strength takes into account the rope fill factor and spinning loss factor.

Regarding the choice of rail wheels, the factor C_2 is given in relation to the class of utilization of the mechanism and not the classification group, in order to keep the method used so far which is fully satisfactory.

To conclude the summary of the major changes introduced in 1992 to the previous edition (documents FEM 2 131 and 2 132), it is worthwhile noting that the changes made during the elaboration of the standard ISO 5049/1 published in 1994 (which reproduced the FEM rules 2 131), have of course been incorporated in the 1992 edition.

1-2 INTRODUCTION

To facilitate the use of these rules by the purchasers, manufacturers and safety organizations concerned, it is necessary to give some explanation in regard to the two following questions :

- How should these rules be applied in practice to the different types of appliance whose construction they cover ?

- How should a purchaser use these rules to define his requirements in relation to an appliance which he desires to order and what conditions should he specify in his enquiry to ensure that the manufacturers can submit a proposal in accordance with his requirements ?

1) First of all, it is necessary to recognize the great variety of appliances which are covered by the design rules. It is obvious that a bucket-wheel reclaimer used for very high duty in a stockyard is not designed in the same manner as a small stacker for infrequent duty. For the latter, it may not be necessary to make all the verifications which would appear to be required from reading through the rules, because one would clearly finish with a volume of calculations which would be totally out of proportion to the objective in view.

   The manufacturer must therefore decide in each particular case which parts of the new machine, should be analysed and which parts can be accepted without calculation. This is not because the latter would contravene the requirements of the rules but because, on the contrary, due to experience, the designer is certain in advance, that the calculations for the latter would only confirm a favourable outcome. This may be because a standard component is being used which has been verified once and for all or because it has been established that some of the verifications imposed by the rules cannot, in certain cases, have an unfavourable result and therefore serve no purpose.

   With the fatigue calculations, for example, it is very easy to see that certain verifications are unnecessary for appliances of light or moderate duty because they always lead to the conclusion that the most unfavourable cases are those resulting from checking safety in relation to the elastic limit or to the breaking stress.

   These considerations show that calculations, made in accordance with the rules, can take a very different form according to the type of appliance which is being considered, and may, in the case of

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a simple machine or a machine embodying standard components, be in the form of a brief summary without prejudicing the compliance of the machine with the principles set out by the design rules.

2) As far as the second question is concerned, some explanation is first desirable for the purchaser, who may be somewhat bewildered by the extent of the document and confused when faced with the variety of choice which it presents, a variety which is, however, necessary if one wishes to take account of the great diversity of problems to be resolved.

In fact, the only important matter for the purchaser is to define the duty which is to be expected from the appliance and if possible to give some indication of the duty of the various individual motions.

As regards the service to be performed by the appliance, only one factor must be specified, i.e. the class of utilization, as defined in 2-1.2.2. This gives the group in which the appliance must be ranged.

In order to obtain the number of hours which determines the class of utilization, the purchaser may, for instance, find the product of :
- the average number of hours which the appliance will be used each day,
- the average number of days of use per year,
- the number of years after which the appliance may be considered as having to be replaced.

In the case of mechanisms, the following should also be specified :
- the class of utilization, as defined in 2-1.3.2,
- the load spectrum, as defined in 2-1.3.3.

On the basis of the class of utilization of the appliance as a whole, it is possible to determine a total number of working hours for each mechanism according to the average duration of a working cycle and the ratio between the operating time of the mechanism and the duration of the complete cycle. An example of classification of an appliance, its mechanisms and elements is given in 2-1.5.

As a general rule, the purchaser need not supply any other information in connection with the design of the appliance, except in certain cases : e.g. the value of the out-of-service wind, where local conditions are considered to necessitate design for an out-of-service wind greater than that defined in 2-2.3.6.

1-3 SCOPE OF THE RULES

The purpose of these rules is to determine the loads and combinations of loads which must be taken into account when designing handling appliances, and also to establish the strength and stability conditions to be observed for the various load combinations.

1-4 FIELD OF APPLICATION

These rules are applicable to mobile equipment for continuous handling of bulk materials, especially to rail-mounted :

- stackers
- shiploaders
- reclaimers
- combined stackers and reclaimers
- continuous ship unloaders

(reclaimers, combined stackers and reclaimers, continuous ship unloaders) } equipment fitted with bucket -wheels or bucket chains

For other equipment, such as :

- excavators,
- scrapers,
- reclaimers with scraper chains,
- tyre or crawler-mounted stackers and/or reclaimers,

the clauses in these design rules appropriate to each type of apparatus are applicable.

It should be noted that when a mobile machine includes one or several belt conveyors as conveying elements, the clauses of these design rules, insofar as they apply to the machine in question, are applicable. The selection of the conveyors should be made in accordance with the standard ISO 5048 : "Continuous mechanical handling equipment - Belt conveyors with carrying idlers - Calculation of operating power and tensile forces".

On the other hand, belt conveyors which are not part of a mobile machine are excluded from the scope of these design rules.

1-5 LIST OF MAIN SYMBOLS AND NOTATIONS

Symbol	Dimen- sion	Designation - First mention chapter (...)
A	m^2	Front area exposed to wind (2-2.2.1)
A2 to A8	-	Handling machine groups (2-1.2)
Ae	m^2	Enveloped area of lattice (2-2.2.1)
B	m	Belt width of the conveyor (2-2.1.2)
B0 to B10	-	Classes of utilization of structural members (2-1.4.2)
b	m	Width of the flow of material on the belt (2-2.1.2)
b	mm	Useful width of rail in wheel calculation (4-2.4.1)
C	-	Coefficient used to calculate the tightening torque of bolts (3-2.3) ; selection coefficient for choice of running steel wire ropes (4-2.2.1)
C_f	-	Shape coefficient in wind load calculation (2-2.2.1)
C_1, C_{1max}	-	Rotation speed coefficients for wheel calculation (4-2.4.1)
C_2, C_{2max}	-	Utilization class coefficient for wheel calculation (4-2.4.1)
c, c'	-	Factors characterising the slope of Wöhler curves (4-1.3.5)
D	-	Symbol used in plate inspection for lamination defects (3-2.2.1)
D	mm	Rope winding diameter (4-2.3.1) ; wheel diameter (4-2.4.1)
D_t	mm	Diameter of bolt holes (3-2.3)
d	mm	Nominal diameter of rope (4-2.2.1)
d_2	mm	Bolt diameter at thread root (3-2.3)
d_t	mm	Nominal bolt diameter (3-2.3)
E	N/mm^2	Elastic modulus (3-2.1.1)
El to E8	-	Groups of components (2-1.4)
F	N	Wind force (2-2.2.1) ; compressive force on member in crippling calculation (3-3)
F_0	N	Minimum breaking load of rope (4-2.2.1)
f	-	Fill factor of rope (4-2.2.1)

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Symbol	Dimension	Designation - First mention chapter (...)
g	m/s^2	Acceleration due to gravity, according to ISO 9.80665 m/s^2
H	-	Coefficient depending on group for choice of rope drums and pulleys (4-2.3.1)
H_y	N	Horizontal force perpendicular to rail axis (2-2.2.6)
j	-	Group number in component groups E1 to E8 (4-1.3.6)
K - K' - K"	-	Safety coefficients for calculation of bolted joints (3-2.3)
K'	-	Empirical coefficient for determining minimum breaking strength of rope (4-2.2.1)
K0 to K4	-	Stress concentration classes for welded parts (3-4.5.1)
K_L	N/mm^2	Pressure of wheel on rail (4-2.4.2)
k	-	Spinning loss coefficient for ropes (4-2.2.1)
k_d	-	Size coefficient in fatigue verification of mechanism parts (4-1.3.3)
k_m	-	Spectrum factor for mechanisms (2-1.3.3)
k_s	-	Shape coefficient in fatigue verification of mechanism parts (4-1.3.3)
k_{sp}	-	Spectrum factor for components (2-1.4.3)
k_u	-	Surface finish (machining) coefficient in fatigue verification of mechanism parts (4-1.3.3)
k_{uc}	-	Corrosion coefficient in fatigue verification of mechanism parts (4-1.3.3)
L1 to L4	-	Spectrum classes for mechanisms (2-1.3.3)
l	mm	Overall width or rail head (4-2.4.1)
l_k	mm	Length of parts tightened in bolted joints (3-2.3)
M	Nm	External moment in bolted joints (3-2.3.4.4)
M1 to M8	-	Mechanism groups (2-1.3)
M_a	Nm	Torque required to tighten bolts (3-2.3)
M_s	Nm	Stabilizing moment for the machine (3-6.1)
M_k	Nm	Overturning moment (3-6.1)
m	-	Number of friction surfaces in bolted joints (3-2.3.4.2)

Symbol	Dimension	Designation - First mention chapter (...)
N	-	Number of stress cycles (2-1.4.2)
N	N	External force perpendicular to joint plane in bolted joints (3-2.3.4.3)
N_a	kN	Permissible additional tensile force for bolt (3-2.3.4.5)
n	-	Number of stress cycles (4-1.3.5)
P	N	Load on wheel (4-2.4.2)
P1 to P4	-	Spectrum classes for components (2-1.4.3)
P10, P100	-	Symbols indicating welding tests (3-2.2.1)
P_{mean} I, II, III	N	Mean load on wheel in loading cases I, II and III (4-2.4.1)
P_{min} I, II, III	N	Minimum load on wheel in loading cases I, II and III (4-2.4.1)
P_{max} I, II, III	N	Maximum load on wheel in loading cases I, II and III (4-2.4.1)
P_a	mm	Pitch of thread (3-2.3)
p_L	N/mm^2	Limiting pressure in wheel calculation (4-2.4.1)
q	-	Correction factor shape coefficient k_s (4-1.3.1)
q	N/mm^2	Aerodynamic pressure of the wind (2-2.2.1)
R_0	N/mm^2	Minimum ultimate tensile strength of the wire of a rope (4-2.2.1)
r	mm	Radius of rope groove (4-2.3.2) ; radius of rail head (4-2.4.1) ; blending radius (4-1.3.1)
S	N	Maximum tensile force in rope (4-2.2.1)
S	m^2	Area of material on the conveyor belt (2-2.1.2)
S_G	-	Center of gravity of dead loads (3-6.1)
S_{mean}	N	Mean load in bearing calculation (4-2.1.2)
S_{min}	N	Minimum load in bearing calculation (4-2.1.2)
S_{max} I, II, III	N	Maximum load in load cases I, II or III (4-2.1.2)
S_b	mm^2	Root sectional area of bolt (3-2.3.3)
s	m	Span of handling appliance (6-2.2)
T	h	Total duration of use of handling appliance and its mechanisms (2-1.2.2) - (2-1.3.2)

Symbol	Dimen-sion	Designation - First mention chapter (...)
T	°C	Ambient temperature at place of erection (3-1.3)
T	N	External force parallel to joint plane in bolted joint (3-2.3.4.2)
T0 to T9	-	Classes of utilization of mechanisms (2-1.3.2)
T_a	N	Permissible load per bolt which can be transmitted by friction (3-2.3.4.2)
T_c	°C	Test temperature for impact test (3-1.3)
t_1, t_2,... t_j, ..., t_r	s	Duration of different levels of loading (2-1.3)
V_s	m/s	Theoretical wind speed (2-2.2.1)
V_t	m/s	Nominal travel speed of handling appliance (2-2.3.7)
V_y	N	Vertical load on travelling wheel (2-2.2.6)
W_0, W_1, W_2	-	Notch cases of unwelded members (3-4.5.1.1)
Z_A	-	Assessing coefficient for influence A (3-1.1.1)
Z_B	-	Assessing coefficient for influence B (3-1.1.2)
Z_C	-	Assessing coefficient for influence C (3-1.1.3)
Z_p	-	Minimum practical factor of safety for choice of steel wire ropes (4-2.2.1)
Δl_1	mm	Shortening of joined elements under the tightening force in bolted joints (3-2.3.3.1)
Δl_2	mm	Extension of bolt under tightening force (3-2.3.3.1)
Δs	mm	Divergence in span of machine (6-2.2) ; divergence in machine rail centres (6-2.3)
η	-	Shielding coefficient in calculation of wind force (2-2.2.1)
θ	°	Surcharge angle for material on belt (2-2.1.2)
θ	°	Angle of wind relative to longitudinal axis of member (2-2.2.1)
κ	-	Ratio of the extreme stress values in fatigue calculation (3-4.4)
λ	-	Slenderness of column in crippling calculation (3-3.1)

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Symbol	Dimension	Designation - First mention chapter (...)
μ	-	Friction coefficient in calculation of loads due to motion (2-2.2.5) ; coefficient of friction in threads (3-2.3)
ν_E	-	Safety coefficient for calculation of structural members depending on case of loading (3-2.1.1)
ν_R	-	Safety coefficient for calculation of mechanism parts depending on case of loading (4-1.1.2)
ν_K	-	Safety coefficient for verification of stability (3-6.1)
ρ	kg/m^3	Air density
σ	N/mm^2	Calculated stress in structures in general
σ_E	N/mm^2	Apparent elastic limit (3-2.1.1)
σ_R	N/mm^2	Ultimate tensile strength (3-2.1.1)
σ_R^E	N/mm^2	The Euler stress (3-3.3)
σ_a	N/mm^2	Permissible tensile stress for structural members (3-2.1.1)
σ_{aw}	N/mm^2	Maximum permissible stress in welds (3-2.2.2)
σ_{cg}	N/mm^2	Compression stress in wheel and rail (4-2.4.2)
σ_{cp}	N/mm^2	Equivalent stress used in calculating structural members (3-2.1.3)
τ	N/mm^2	Shear stress in general
τ_a	N/mm^2	Permissible shear stress when calculating structural members (3-2.1.2)
τ_{aw}	N/mm^2	Maximum permissible shear stress in welds (3-2.2.2)
Φ	-	Coefficient of elongation for calculation of bolts (3-2.3.4.5)
φ, φ'	-	Slope of Wöhler curve (4-1.3.5)
ψ	-	Ratio of stresses at plate edges in buckling calculation (3-3.3)
ω	-	Crippling coefficient (3-3.1.1)

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Chapter 2

CLASSIFICATION AND LOADING OF STRUCTURES AND MECHANISMS

CHAPTER 2

CLASSIFICATION AND LOADING OF STRUCTURES AND MECHANISMS

CONTENTS

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2-1 GROUP CLASSIFICATION OF MOBILE EQUIPMENT AND THEIR COMPONENTS

2-1.1 GENERAL PLAN OF CLASSIFICATION

When designing a mobile appliance and its components for the continuous handling of bulk materials, the service which they are to provide and their utilization should be taken into consideration. To that end, a group classification is employed for :

- the complete handling machine,
- the complete individual mechanisms,
- the components of structure and mechanisms.

This classification has been established of the base of two criteria :

- the duration of use,
- the load or stress spectrum.

2-1.2 CLASSIFICATION OF THE COMPLETE HANDLING MACHINE

2-1.2.1 CLASSIFICATION SYSTEM

Complete handling machines are classified in seven groups respectively designated by the symbols A2, A3, ... A8 as defined in the table T.2-1.2.2 on the basis of seven classes of utilization.

Important note : The load spectrum which characterizes all the loads handled by the machine during its lifetime is not taken into account for the classification of the complete machine (see 2-1.2.3).

2-1.2.2 CLASSES OF UTILIZATION = GROUPS

The utilization class of a machine depends on its total duration of use.

The total duration of use of a machine is defined as the number of hours during which the machine is actually in operation during its lifetime.

The total duration of use is a calculated duration of use, considered as a guide value, commencing when the appliance is put into service and ending when it is finally taken out of service.

This duration, measured by a number of hours of use T, depends on the desired service time in years, the average actual number of service days per year and the average actual number of service hours per day.

On the base of the total duration of use, we have seven groups of handling machines, designed by the symbols A2, A3, ... A8.

They are defined in table T.2-1.2.2.

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Complete mechanical handling machines are most commonly classified in groups A4 to A8.

Table T.2-1.2.2
GROUPS FOR HANDLING MACHINES

Total duration of use T (h)					GROUP Symbol :
		T	≤	1 600	A2
1 600	<	T	≤	3 200	A3
3 200	<	*T*	≤	*6 300*	*A 4*
6 300	<	*T*	≤	*12 500*	*A 5*
12 500	<	*T*	≤	*25 000*	*A 6*
25 000	<	*T*	≤	*50 000*	*A 7*
50 000	<	*T*			*A 8*

2-1.2.3 LOAD SPECTRUM

Handling machines in operation are usually loaded close to the rated capacities which have been taken into account in their dimensioning.

Additionally, the dead weight of the machines is usually important compared to the handled loads. Thus, a handling machine will normally be loaded close to the maximum design value throughout its life.

The load spectrum is therefore not taken into account for the classification of complete handling machines.

2-1.3 CLASSIFICATION OF COMPLETE INDIVIDUAL MECHANISMS

2-1.3.1 CLASSIFICATION SYSTEM

Individual complete mechanisms are classified in eight groups, designated respectively by the symbols M1, M2, ..., M8, on the basis of ten classes of utilization (T0 to T9) and four classes of loading spectrum (L1 to L4). See 2-1.3.4.

2-1.3.2 CLASS OF UTILIZATION

The class of utilization of a mechanism depends on its total duration of use. This is defined as the time during which the mechanism is actually in motion during the life time of the machine.

The total duration of use of a mechanism is a calculated duration considered as a guide value and takes into account the fact that the mechanism may be designed to be replaced a number of times in the total life of the machine.

The total duration of use of a mechanism is expressed in terms of hours, T, calculated on the basis of the average number of service hours/day, the number of actual service days/year and the required duration in terms of years before replacement.

On this basis, we have ten classes of utilization, T0, T1, T2, ..., T9, the most common classes being T3 to T9. They are defined in table T.2-1.3.2.

Table T.2-1.3.2
CLASSES OF UTILIZATION

Utilization class symbol	Total duration of use T (h)
T0	$T \leq 200$
T1	$200 < T \leq 400$
T2	$400 < T \leq 800$
T3	$800 < T \leq 1\,600$
T4	$1\,600 < T \leq 3\,200$
T5	$3\,200 < T \leq 6\,300$
T6	$6\,300 < T \leq 12\,500$
T7	$12\,500 < T \leq 25\,000$
T8	$25\,000 < T \leq 50\,000$
T9	$50\,000 < T$

2-1.3.3 LOADING SPECTRUM FACTOR

The loading spectrum factor characterizes the magnitude of the loads acting on a mechanism during its total duration of use. There is a distribution function, expressing the fraction of the total duration of use (see table T.2-1.3.2) for which the mechanism is subjected to a load attaining a fraction of the maximum rated load.

An example of a loading spectrum is given in figures 2-1.3.3.1 a and b.

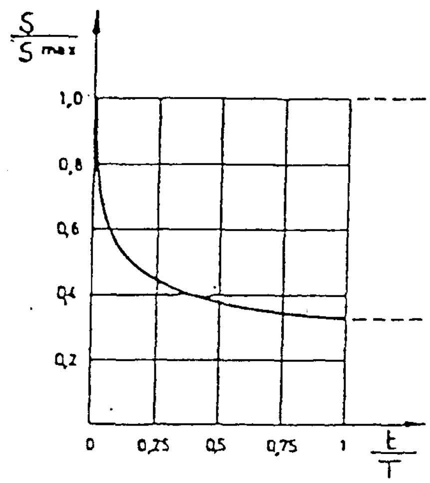


Figure 2-1.3.3.1 a.

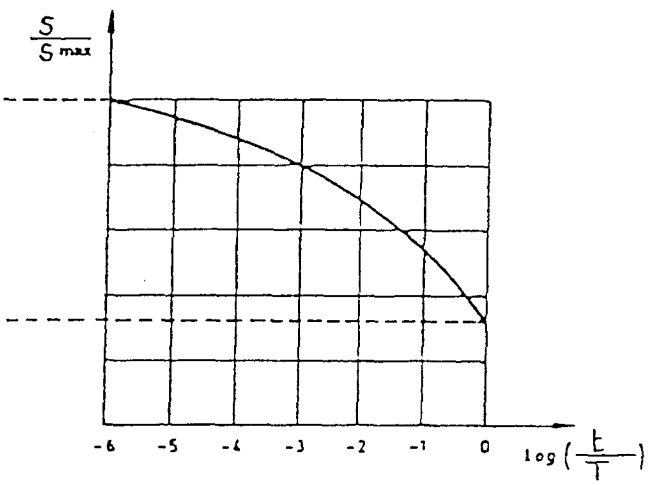


Figure 2-1.3.3.1 b.

S_i	=	loading
S_{max}	=	maximum admissible load
t_i	=	duration for which the loading is at least equal to S_i
T	=	total duration of use

Each spectrum is assigned a spectrum factor k_m, defined by :

$$k_m = \int_0^T \left(\frac{S(t)}{S_{max}}\right)^d \cdot \frac{dt}{T}$$

For the purposes of group classification, exponent d is taken by convention as equal to 3.

In many applications the function S(t) may be approximated by a function consisting of a certain number of steps r (see fig. 2-1.3.3.2), of respective durations $t_1, t_2, ..., t_r$, for which the loadings may be considered as practically constant and equal to S_i during the duration t_i. If T represents the total duration of use and S_{max} the greatest of the loadings $S_1, S_2, ..., S_r$, there exists a relation :

$$t_1 + t_2 + ... + t_r = \sum_{i=1}^{r} t_i = T$$

.../

and in approximated form :

$$k_m = \left(\frac{S_1}{S_{max}}\right)^3 \cdot \frac{t_1}{T} + \left(\frac{S_2}{S_{max}}\right)^3 \cdot \frac{t_2}{T} + \ldots + \left(\frac{S_r}{S_{max}}\right)^3 \cdot \frac{t_r}{T} = \sum_{i=1}^{r} \left(\frac{S_i}{S_{max}}\right)^3 \cdot \frac{t_i}{T}$$

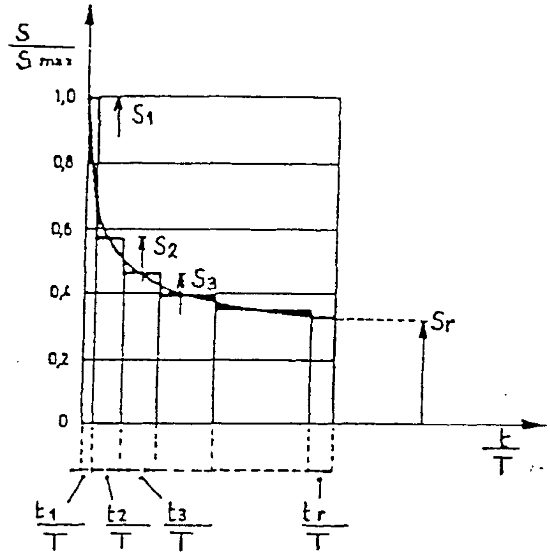


Fig. 2-1.3.3.2

Depending on its loading spectrum, a mechanism is placed in one of the four spectrum classes L1, L2, L3, L4, defined in table T.2-1.3.3, the most common classes generally being L3 and L4 for the main mechanisms.

Table T.2-1.3.3
SPECTRUM CLASSES

Spectrum class symbol	Spectrum factor k_m				
L1			k_m	≤	0.125
L2	0.125	<	k_m	≤	0.250
L3	***0.250***	<	k_m	≤	***0.500***
L4	***0.500***	<	k_m	≤	***1.000***

2-1.3.4 GROUP CLASSIFICATION OF COMPLETE INDIVIDUAL MECHANISMS

Depending on their class of utilization and spectrum class, complete individual mechanisms are classified in one of the eight groups M1, M2, ..., M8, defined in table T.2-1.3.4. The groups generally chosen for the main mechanisms are M4 to M8, a result of the commonly selected classes of utilization and spectrum factors.

Table T.2-1.3.4
MECHANISM GROUPS

Class of load spectrum	Class of utilization									
	T0	T1	T2	*T3*	*T4*	*T5*	*T6*	*T7*	*T8*	*T9*
L1	M1	M1	M1	M2	M3	M4	M5	M6	M7	M8
L2	M1	M1	M2	M3	M4	M5	M6	M7	M8	M8
L3	M1	M2	M3	*M4*	*M5*	*M6*	*M7*	*M8*	*M8*	*M8*
L4	M2	M3	M4	*M5*	*M6*	*M7*	*M8*	*M8*	*M8*	*M8*

2-1.3.5 GUIDANCE FOR GROUP CLASSIFICATION OF COMPLETE INDIVIDUAL MECHANISMS

Since appliances of the same type may be used in a wide variety of ways, according to the method of working for instance, it is not really possible to pre-determine the group of a mechanism exactly.

The classification possibilities are particularly wide for mechanisms which can be either working mechanisms or only positioning mechanisms.

Further discussion relating to the harmonization of classes and groups is given in chapter 2-1.5, along with a typical example of classification of a machine and its components.

2-1.4 CLASSIFICATION OF COMPONENTS

2-1.4.1 CLASSIFICATION SYSTEM

Components, both structural and mechanical, are classified in eight groups, designed respectively by the symbols E1, E2, ..., E8, on the basis of eleven classes of utilization B0 to B10 and four classes of stress spectrum P1 to P4.

2-1.4.2 CLASSES OF UTILIZATION FOR COMPONENTS

The class of utilization of a component depends on its total duration of use, which is defined as the number of stress cycles to which the component is subjected during the lifetime of the machine.

A stress cycle is a complete set of successive stresses, commencing at the moment when the stress under consideration exceeds the stress σ_m defined in fig. 2-1.4.3 and ending at the moment when this stress is, for the first time, about to exceed again σ_m in the same direction. Fig. 2-1.4.3 therefore represents the fluctuations of the stress σ over a duration of use equal to five stress cycles.

<u>The total duration of use</u> of a component is a calculated duration, considered as a guide value and taking into account the fact that the component may be designed to be replaced a number of times in the total life of the machine.

In the case of structural components the number of stress cycles is proportional to the number of typical handling sequences. Certain components may be subjected to several stress cycles during a typical sequence depending on their position in the structure. Hense the ratio in question may differ from one component to another. Once this ratio is known, the total duration of use for the component is derived from the total duration of use determined by the class of utilization of the appliance.

For mechanical components, the total duration of use is derived from the total duration of use of the mechanism to which that particular component belongs taking into account its speed of rotation and/or other circumstances affecting its operation.

Based on this total duration of use, we have eleven classes of utilization, designated respectively by the symbols B0, B1, ..., B10. They are defined in table T.2-1.4.2. The most usual classes are B5 to B10 for the main components.

Table T.2-1.4.2
CLASSES OF UTILIZATION

Utilization class symbol	Total duration of use measured by number N of stress cycles				
B0			N	≤	16 000
B1	16 000	<	N	≤	32 000
B2	32 000	<	N	≤	63 000
B3	63 000	<	N	≤	125 000
B4	125 000	<	N	≤	250 000
B5	250 000	<	N	≤	500 000
B6	***500 000***	<	*N*	≤	***1 000 000***
B7	***1 000 000***	<	*N*	≤	***2 000 000***
B8	***2 000 000***	<	*N*	≤	***4 000 000***
B9	***4 000 000***	<	*N*	≤	***8 000 000***
B10	***8 000 000***	<	*N*		

2-1.4.3 STRESS SPECTRUM

The stress spectrum characterizes the magnitude of the loads acting on the component during its total duration of use. There is a distribution function, expressing the fraction of the total duration of use (see 2-1.4.2), during which the component is subjected to a stress attaining at least a fraction of the maximum stress.

Each stress spectrum is assigned a spectrum factor k_{sp}, defined by :

.../

$$k_{sp} = \int_0^N \left(\frac{\sigma(n)}{\sigma_{max}}\right)^c \cdot \frac{dn}{N}$$

Wherein the exponent c is dependant on the properties of the material concerned, the shape and size of the component, its surface roughness and its degree of exposure to corrosion.

- For structural components : unless otherwise specified, the value of c should normally be taken as 3 for welded parts. Higher values for some configurations may be used in some circumstances, but these should be used with care.

- For mechanical components : the calculation of c values is given in chapter 4-1.3.5 and Annex A.4-1.3.

In many applications the function $\sigma(n)$ 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 $\sigma_1, \sigma_2, ..., \sigma_r$, there exists a relation :

$$\text{with } n_1 + n_2 + ... + n_r = \sum_{i=1}^{r} n_i = N$$

$$\text{and : } \sigma_1 > \sigma_2 > .. > \sigma_r$$

we get an approximated form :

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

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

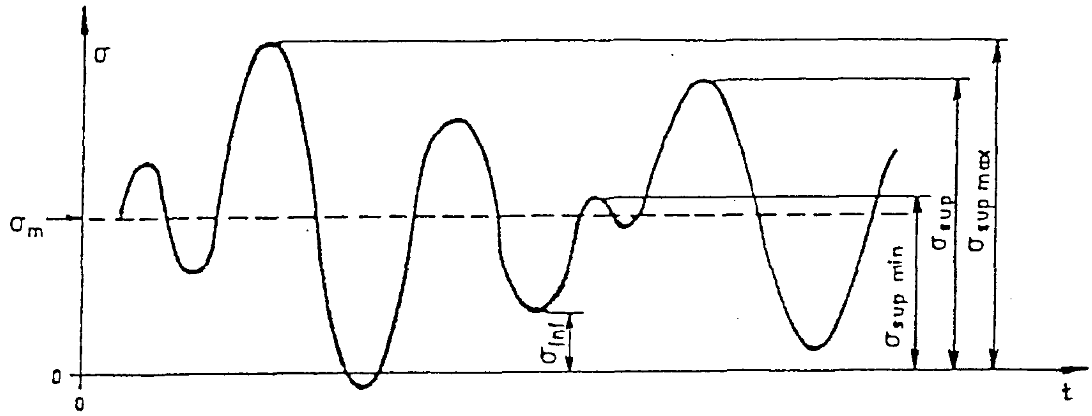
Depending on its stress spectrum, a component is placed in one of the spectrum classes P1, P2, P3, P4, defined in table T.2-1.4.3, the most usual classes being P3 and P4 for main components.

Table T.2-1.4.3
SPECTRUM CLASSES

Spectrum class symbol	Spectrum factor k_{sp}				
P1			k_{sp}	$\leq$	0.125
P2	0.125	<	k_{sp}	$\leq$	0.250
P3	***0.250***	<	k_{sp}	$\leq$	***0.500***
P4	***0.500***	<	k_{sp}	$\leq$	***1.000***

- For structural components, the stresses to be taken into consideration for determination of the spectrum factor are the differences $\sigma_{sup} - \sigma_m$ between the upper stresses σ_{sup} and the average stress σ_m, these concepts being defined by fig. 2-1.4.3 representing the variation of the stress over time during five stress cycles.

Fig. 2-1.4.3 - Variation of stress as a function of time during five stress cycles



σ_{sup}	=	upper stress
$\sigma_{sup\ max}$	=	maximum upper stress
$\sigma_{sup\ min}$	=	minimum upper stress
σ_{inf}	=	lower stress
σ_m	=	arithmetic mean of all upper and lower stresses during the total duration of use

- In the case of mechanical components, σ_m can be assumed to be zero and the stresses introduced into the calculation of the spectrum factor are then the total stresses occurring in the relevant section of the component.

Note : Stress changes with values less than 10 % of the maximum stress are not to be considered for the calculation of the spectrum factor for structural or mechanical components.

These small stress changes, as proved by experience, have no noticeable effect on the working life.

2-1.4.4 GROUP CLASSIFICATION OF COMPONENTS

On the basis of their class of utilization and their stress spectrum class, components are classified in one of the eight groups E1, E2, ..., E8, defined in table T.2-1.4.4.

Considering the most common classes of utilization and stress spectrum classes, the groups generally used for main components are E5 to E8.

Table T.2-1.4.4
COMPONENT GROUPS

Stress spectrum class	Class of utilization										
	B0	B1	B2	B3	B4	***B5***	***B6***	***B7***	***B8***	***B9***	***B10***
P1	E1	E1	E1	E1	E2	E3	E4	E5	E6	E7	E8
P2	E1	E1	E1	E2	E3	E4	E5	E6	E7	E8	E8
P3	E1	E1	E2	E3	E4	***E5***	***E6***	***E7***	***E8***	***E8***	***E8***
P4	E1	E2	E3	E4	E5	***E6***	***E7***	***E8***	***E8***	***E8***	***E8***

2-1.5 HARMONIZATION OF CLASSIFICATION FOR THE COMPLETE MACHINE, COMPLETE MECHANISMS AND COMPONENTS (STRUCTURE AND MECHANISMS)

2-1.5.1 COMPLETE MACHINE

The classification of the various mechanisms or components (of structure or mechanisms) of a given handling machine is derived from the expected TOTAL DURATION OF USE (expressed in hours) of the machine during its lifetime which defines its GROUP.

The total duration of use (expressed in hours) of the complete machine is the product of the three following estimated factors :

- average number of actual service hours per day,
- average number of actual service days per year,
- number of desired service years.

2-1.5.2 COMPLETE MECHANISMS

From the total duration of use of the machine the duration of use of each complete mechanism is determined. This duration obviously depends on the operating mode of the machine and of the mechanism involved, and corresponds to a class of utilization (T0 to T9). The combination of this class of utilization with the load spectrum class (L1 to L4) defines the mechanism group (M1 to M8). The most commonly used groups will normally be M4 to M8 for main mechanisms.

2-1.5.3 COMPONENTS

The groups for structural or mechanical components are determined as follows :

2-1.5.3.1 STRUCTURAL COMPONENT GROUPS

On the basis of the frequency of operating cycles (and therefore of repetition cycles) and the total duration of use of the machine, it is possible to calculate the total number of the stress repetition cycles expected in the lifetime of the machine and therefore to classify the structural component in one of the classes of utilization (B0 to B10). This is used, along with the spectrum class (P1 to P4) to select the component group (E1 to E8).

2-1.5.3.2 MECHANICAL COMPONENT GROUPS

Dependant on the class of utilization of a complete mechanism to which a particular component belongs, the total number of stress cycles to which this component will be subjected in the duration of use of the mechanism can be determined.

It is not possible to give a general method for determining the number of stress cycles F for a mechanical component, as the number of stress cycles greatly depends upon the type of load and the function of the component in a given mechanism.

. Elements of mechanisms whose number of stress cycles depends only on the number of working cycles

This applies to mechanism elements which, for each operating cycle, undergo stress variations corresponding to a certain multiple of the operating cycle, for example, carriage wheel axles, slewing bearings for rotating parts of the unit.

In this case the number of cycles per hour is :

$$F_h = k_a \cdot S_p$$

where :

S_p = number of working cycles per operating hour

k_a = the factor by which the number of operating cycles is multiplied when the mechanism element is subjected to several stress cycles for each operating cycle.

Example : carriage wheel axle of a reclaiming unit where one working cycle includes the following movements :

1 - forward movement of the unit with wheel load
2 - slewing the boom
3 - forward movement of the unit with altered wheel load
4 - boom return.

.../

In this case : $k_a = 2$, for a fixed axle as the stress changes only when the boom is slewed.

. Mechanism components with stress cycles depending on the number of revolutions per minute

This group contains all rotating mechanism parts which are stressed by rotating-bending and shearing stresses. It also applies to the bending and Hertzian stresses on toothed gears.

In these cases :

$$F_h = k_a \cdot n_m \cdot 60$$

where :

n_m = number of revolutions per minute

For the Hertzian stress of toothed gears, whose flanks are used on both sides, e.g. under-carriages or slewing mechanisms, it shall be considered :

$k_a = 0.5$

From the number of stress cycles to which the mechanical component will be subjected, a class of utilization (B0 to B10) can be determined. This is used, along with the spectrum class (P1 to P4), to select the component group (E1 to E8).

In this manner, all components or sets of components are classified in GROUPS representing the anticipated service to be provided by those components.

2-1.5.4 AN EXAMPLE OF THE CLASSIFICATION OF A MACHINE AND ITS COMPONENTS

The classification and values given in this chapter must only be considered as an example.

The assumptions relating to particular design requirements and operating methods are specific to this example and will vary according to the detailed design of the machine, site conditions, working method, etc.

This example is based on a combined stacker/bucket-wheel reclaimer for the handling of ore and coal.

Stacking capacity :	3000 t/h of ore - 2000 t/h of coal
Reclaiming capacity :	2000 t/h of ore - 1000 t/h of coal
Ore handling :	2/3 of total tonnage
Coal handling :	1/3 of total tonnage

Desired duration of use : 50,000 hours, i.e. GROUP A7 for the machine as a whole.

From these parameters it can be calculated that the machine will be used in the following way :

- Ore stacking	=	10,500 h	approx.	(31.5 Mt)
- Coal stacking	=	7,900 h	approx.	(15.8 Mt)
- Ore reclaiming	=	15,800 h	approx.	(31.5 Mt)
- Coal reclaiming	=	15,800 h	approx.	(15.8 Mt)
	=	50,000 h		

EXAMPLE OF GROUP CLASSIFICATION OF MECHANISMS AS A WHOLE

Mechanism	Total duration of use (h)	Class of utilization	Spectrum factor k_m * (3)	Spectrum class	GROUP
Reclaiming unit	31,600	T8	i.e. 0.76	L4	M8
Boom conveyor	50,000	T8	i.e. 0.45	L3 *(4)	M8
Slewing	33,500 *(1)	T8	i.e. 0.80	L4	M8
Lifting	5,000	T5	i.e. 1	L4	M7
Travelling	12,500 * (2)	T6	i.e. 1	L4	M8

*(1) Assuming :

	31,600 h	for reclaiming operation
+	1,900 h	during 10 % of stacking operation

*(2) Assuming :

	7,900 h	continuous travel during coal stacking to ensure preliminary blending
	1,100 h	travel during 10 % of stacking operation (iron ore)
	3,200 h	travel during 10 % of reclaim operation (5-6 seconds/minute for advance)
TOTAL :	12,200 h	
rounded to	12,500 h.	

*(3) Spectrum factors must be calculated on the basis of the loads applied during the whole or relevant parts of the four main service phases, i.e. ore stacking, coal stacking, ore reclaiming, coal reclaiming.

.../

For the reclaiming unit for instance, if the loads created by coal reclaiming amount to 80 % of the loads created by ore reclaiming (corresponding to a maximum load) for equal service times for ore and coal, factor k_m will be $0.5 + 0.8^3 . 0.5 = 0.76$ approx.

It should be noted that the most common spectrum class for these appliances is class L4.

*(4) Assuming a reversible boom conveyor, the different duration of utilization in relation to the total duration of utilization of the machine will be as follows :

- storage of iron ore	=	0.21	out of a total of
- storage of coal	=	0.158	50,000 hours of
- reclaiming of iron ore	=	0.316	scheduled utilization
- reclaiming of coal	=	0.316	

If we consider that the mechanism will be loaded at the maximum value of 1 for the storage of iron ore, the loads on the mechanism for the three other operations are for instance :

- 0.74 for storage of coal and reclaiming of iron ore
- 0.53 for reclaiming coal at the capacity of 1000 t/h

we have therefore :

$k_m = 0.21 . 1^3 + 0.158 . 0.74^3 + 0.316 . 0.74^3 + 0.316 . 0.53^3$

$k_m = 0.449$ and spectrum class L3

It must be noted that, for a boom conveyor with a variable inclination it would be possible, for each of the four kinds of utilization, to calculate an average load for the driving mechanism as it is obvious that the maximum loading when staking material is supported only when the elevation of the handled product is maximum, i.e. with the boom in the highest position.

Nevertheless, the interest of this study appears only if it is possible to go from a stress factor spectrum to a smaller one, which is not the case for the example above.

When chosing some components of the mechanism such as the gear box reducer, consideration may be given to the fact that the boom conveyor is reversible, which means that the reducer will be used 18,400 hours driving in one direction and 31,600 hours in the opposite direction.

CLASSIFICATION OF STRUCTURAL AND MECHANICAL COMPONENTS

For structural components it is necessary to determine the number of stress cycles to which a given component will be subjected during the 50,000 h use of the machine.

A structural component will therefore belong to a group (generally E5 to E8) depending essentially on its position in the machine.

As regards mechanical components, it is also necessary to determine the number of stress cycles to which each part will be subjected during the use of the mechanism as a whole, e.g. the slew drive pinion can be classified as follows :

- given rotating speed 2 rpm, hence F = 1/2 . 2 . 60 = 60/h
  number of hours of utilization = 33,500
  hense expected stress cycles = 33,500 . 60 = 2.01 . 10^6
  class of utilization = B8
  spectrum class = P4
  (according to spectrum factor of approx. 0.8, similar to spectrum factor of slewing mechanism)
  hense group classification E8.

2-2 LOADS ENTERING INTO THE DESIGN OF STRUCTURES

The structural calculations shall be undertaken to determine the stresses developed in an appliance during its operation or when out of service with maximum wind conditions. These stresses shall be calculated on the basis of the loads defined below.

Depending on their frequency, the loads are divided into three different load categories : main loads, additional loads and special loads.

a) The main loads include all the permanent loads which occur when the equipment is used under normal operating conditions.

They mainly are :

- dead loads,
- material loads,
- incrustation,
- normal tangential and lateral digging forces,
- forces on the conveyor(s),
- permanent dynamic effects,
- inclination of the working level.

b) The additional loads are loads that can occur intermittently during operation of the equipment or when the equipment is not working ; these loads can either replace certain main loads or be additional to the main loads.

They are, among other :

- wind load for machine in service, } climatic effects
- snow and ice loads, } climatic effects
- temperature effects, } climatic effects
- abnormal tangential and lateral digging forces,
- bearing friction and rolling resistances,
- reactions perpendicular to the rail due to travelling (skewing effect),
- non permanent dynamic effects.

c) The special loads comprise those loads which should not occur during and outside the operation of the equipment but the occurence of which is not to be excluded.

They include :

- clogging of chutes,
- resting of the reclaiming device or the boom,
- failure of load limiting devices (see 2-2.1.2.1),
- blocking of travelling devices,
- lateral collision of the bucket-wheel with the slope,
- wind load on machines out of service,
- buffer effects,
- loads due to earthquakes,
- loads during erection (or dismantling) of the machine.

2-2.1 MAIN LOADS

2-2.1.1 DEAD LOADS

Dead loads consist of all loads of constant magnitude and position on the machine, which act permanently on the structure or member being designed.

Note : It should be noted that stairs, platforms and gangways on which equipment can be placed must be calculated to bear 3 kN of "patch load" (on an assuming area of 0.5 x 0.5 m). Access and gangways only used as passage for people are calculated to bear 1.5 kN/m, and the railings and guards to stand 0.3 kN of horizontal load.

When higher loads are to be supported temporarily by platforms, the latter must be designed and sized accordingly. Platforms of large size must be verified with a load of at least 1 kN/m^2.

Dead loads include the structure of stairs, gangways, etc, but exclude loads <u>on</u> stairs, gangways, etc, which are to be considered as local loads for the design of these stairs, gangways, etc, but <u>not</u> for the design of the structure as a whole.

2-2.1.2 MATERIAL LOADS

The material loads carried on conveyors and reclaiming devices are to be considered as follows :

2-2.1.2.1 MATERIAL LOAD CARRIED ON THE CONVEYORS

These loads are determined from the design capacity (m^3/h) and the maximum corresponding bulk density specified by the customer.

1) <u>Units with no built-in reclaiming device</u>

a) Where the belt load is limited by automatic devices, the load on the conveyor will be assumed to be that which results from the capacity thus limited.

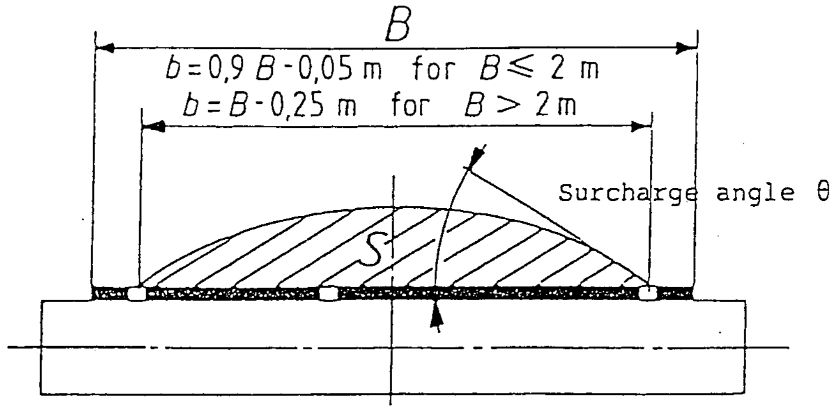
b) Where there is no capacity limiter, the design capacity is that resulting from the maximum equivalent cross sectional area of the material on the conveyor multiplied by the conveying speed.

Unless otherwise specified in the contract, this cross sectional area shall be determined assuming a surcharge angle $\theta = 20°$ and considering a surcharge area according to ISO 5048 "Continuous mechanical handling equipment - Belt conveyors with carrying idlers - Calculation of operating power and tensile forces".

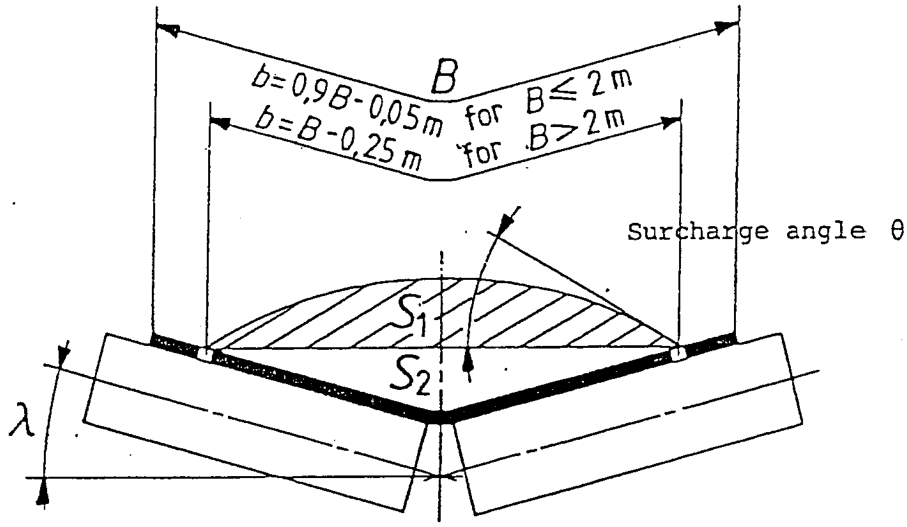
The diagrams 2-2.1.2.1 hereafter give the cross sectional area to be considered for different belt conveyor designs.

.../

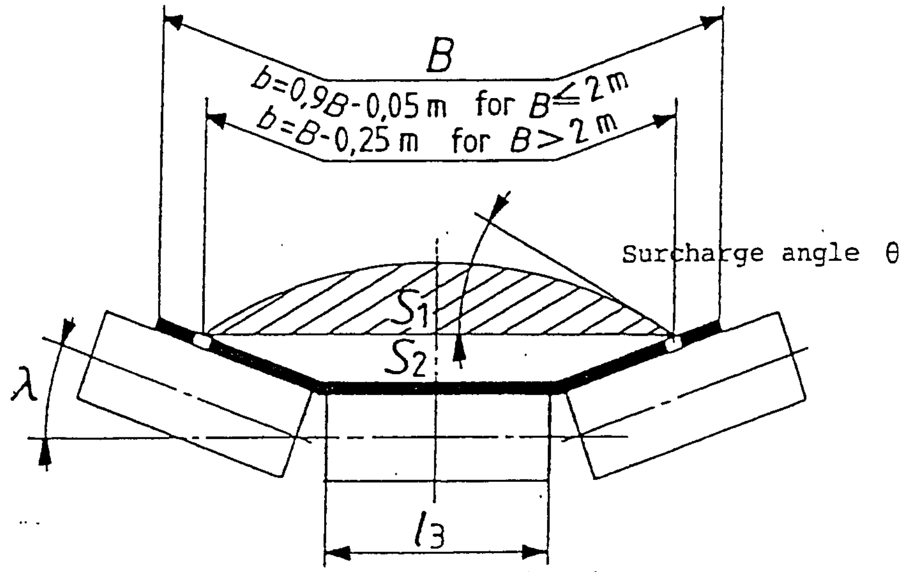
Fig. 2-2.1.2.1 - Belt conveyor cross-sections



a) With one carrying idler



b) With two carrying idlers



c) With three carrying idlers

.../

c) Where the design capacity resulting from b) on the upstream units is lower than that of downstream units, the downstream units may have the same capacity as the upstream units.

2) Units fitted with bucket-wheel or bucket chain as reclaiming device

a) Where there is no capacity limiter, the design volumetric capacity shall be taken as 1.5 times the nominal filling capacity of the buckets multiplied by the maximum number of discharges. In the case of bucket-wheels, the factor 1.5 (which takes into account the volumes which can be filled in addition to the buckets), can be replaced by taking into account the actual fill value additional to the bucket volume.

b) Where there are automatic capacity limiters, the design capacity shall be the capacity thus limited.

Where the unit is to be used to convey materials of different bulk densities (for example coal and ore) safety devices shall be provided to ensure that the calculated loads will not be exceeded with the heavier material.

3) Dynamic load factor

To account for the dynamic loads with can be applied to the conveyor in transporting the material, the material loads defined above must be multiplied by the factor 1.1.

2-2.1.2.2 LOADS IN THE RECLAIMING DEVICES

To take into account the weight of the material to be conveyed in the reclaiming devices it can be assumed that, typically :

a) for bucket-wheels :

- 1/4 of all available buckets are full at 100 %,

b) for bucket-chains :

- 1/3 of all the buckets in contact with the face are 33.3 % full,
- 1/3 of all the buckets in contact with the face are 66.7 % full,
- all other buckets up to the sprocket are 100 % full.

2-2.1.2.3 MATERIAL IN THE HOPPERS

The weight of the material in hoppers is obtained by multiplying the bulk density of the material by the volume (filled to the brim).

Where the weight of the material is limited by reliable automatic controls, deviation from the above-mentioned value is permissible.

2-2.1.3 INCRUSTATION

The degree of incrustation (dirt accumulation) depends on the specific material and operating conditions prevailing in each given case.

The data which follow are to be taken as a guide, and are generally applicable to stockyards machines.

For excavating equipment they are to be taken as minimum values.

Unless experience in particular cases or customer's requirements specify otherwise, the loads due to dirt accumulation that should be taken into account are :

a) on the conveying devices 10 % of the material load calculated according to 2-2.1.2,

b) for bucket-wheels the equivalent weight of a 5 cm thick layer of material of maximum specified bulk density on the centre of the bucket-wheel considered as a solid disc up to the cutting circle,

c) for bucket-chains 10 % of the design material load calculated according to 2-2.1.2, uniformly distributed over the total length of the ladder.

2-2.1.4 NORMAL TANGENTIAL AND LATERAL DIGGING FORCES

These forces are to be calculated as concentrated loads, i.e. on bucket-wheels as acting at the most unfavourable point of the cutting circle, on bucket-chains as acting at a point one-third of the way along the part of the ladder in contact with the face.

a) Normal tangential digging force

- For excavators and, in general, for machines for which the digging effort is largely uncertain, the normal digging force acting tangentially to the wheel cutting circle or in the direction of the bucket-chain is obtained from the nominal rating of the drive motor, the efficiency of the transmission gear, the circumferential speed of the cutting edge, and the power necessary to lift the material, and in the case of bucket-ladders from the power necessary to move the loaded bucket-chain.

  To calculate the lifting power, the figures indicated in 2-2.1.2.2 must be used.

.../

- For storage yard applications, the above method of calculation may be ignored if the digging resistance of the material is accurately known as a result of tests and if it can be guaranteed that this digging resistance will not be exceeded during normal operation.

b) Normal lateral digging force

Unless otherwise specified, the normal lateral digging force can be assumed as 0.3 times the values of the normal tangential digging force.

2-2.1.5 FORCES ON THE CONVEYOR(S)

Belt tensions, chain tensions, etc, must be taken into consideration for the calculation as far as they have an effect on the structures.

2-2.1.6 PERMANENT DYNAMIC EFFECTS

a) In general the dynamic effect of the digging resistances, the falling masses at the transfer points, the rotating parts of machinery, the vibrating feeders, etc, need only be considered as acting locally.

b) The inertia forces due to acceleration and braking of moving structural parts must be taken into account. These can be neglected for appliances working outdoors if the acceleration and deceleration are $\leq 0.2\ m/s^2$.

If possible the drive motors and brakes must be designed in such a way that this acceleration value $0.2\ m/s^2$ is not exceeded.

If the number of load cycles caused by inertia forces due to acceleration and braking is lower than 2×10^4 during the lifetime of the machine the effects should be considered as additional loads (see 2-2.2.7).

2-2.1.7 LOADS DUES TO INCLINATION OF WORKING LEVEL

In cases where the working level is inclined, dead loads acting vertically should be resolved into components acting perpendicularly and parallel to the working plane.

The slope related loads should be determined for the maximum slope contractually defined and then increased by 20 % for the calculation purposes.

2-2.2 ADDITIONAL LOADS

2-2.2.1 WIND ACTION

This clause relates to wind loads on the structure of handling machines.

It gives a simplified method of calculation and assumes that the wind can blow horizontally from any direction, that the wind blows at a constant velocity and that there is a static reaction to the loadings it applies to the cran structure.

The wind effects shall be considered for the machine in service (see 2-2.2.1.2) and for the machine out of service (2-2.3.6). Unless otherwise specified due to local conditions, the design wind speed given in these chapters should be used.

2-2.2.1.1 AERODYNAMIC WIND PRESSURE

The aerodynamic wind pressure q, in relation with the air density and the wind speed V_S, is given by the formula :

$$q = \frac{\rho V_s^2}{2}$$

q = the aerodynamic pressure N/m²
V_S = the design wind speed in m/s, used for the calculation and depending on the load case
ρ = air density in kg/m³

Under normal conditions, with $\rho = 1.225$ kg/m³, the formula becomes :

$$q = 0.613 V_s^2$$

2-2.2.1.2 IN SERVICE WIND

a) Maximum in service wind

This is the maximum wind in which the machine is designed to operate. The wind loads are assumed to be applied in the least favourable direction in combination with the appropriate service loads.

For calculation and unless specified otherwise, a maximum design wind speed V_S = 20m/s = 72 km/h (constant over the height of the machine) shall be assumed for the machine in operation.

.../

The corresponding pressure q is 250 N/m^2 (1).

b) Equivalent average in service wind

For wear or fatigue calculations on mechanisms for example, an aerodynamic wind pressure equal to a third of the maximum working aerodynamic pressure should be assumed.

This equivalent aerodynamic pressure is assumed to be applied during the whole design lifetime of the mechanism and to cause the same wear or fatigue effects as the actual effects of service winds whose aerodynamic pressures will vary between 0 and the maximum working aerodynamic pressure.

It should be noted that this equivalent aerodynamic pressure for wear calculations corresponds to a wind speed equal to approximately 60 % of the design maximum permissible speed of the service wind.

c) Start-up must always be possible against maximum in service wind, but it is assumed that the operating speeds and nominal accelerations are not necessarily reached under maximum in service wind conditions.

d) Under certain circumstances, an appliance may have to go back, unloaded, to "out of service" anchoring positions. In such cases, travelling mechanisms should be dimensioned in relation to the maximum permissible aerodynamic wind pressures defined in the specifications for this load case, and dependant on the machine surface exposed to the wind, the machine may need to be placed in a configuration designed for such transfer.

2-2.2.1.3 WIND LOAD CALCULATIONS

a) The plane of the exposed parts placed perpendicularly to the wind direction

For most complete or part structures, and individual members used in structures, the wind load is calculated from :

$$F = A \cdot q \cdot C_f$$

for a wind blowing perpendicularly to the exposed plane of the components.

---

(1) Where a wind speed measuring device is to be attached to an appliance, it shall normally be placed at the maximum height. In cases where the wind speed at a different level is more significant to the safety of the appliance, the manufacturer shall state the heigt at which the device shall be placed.

where :

F is the wind load in N
A is the effective frontal area of the part under consideration in m^2
q is the aerodynamic wind pressure corresponding to the appropriate design condition in N/m^2
C_f is the shape coefficient for the part under consideration.

Note : shape coefficient is differently named in some documents.

The total wind load on the structure is taken as the sum of the loads on its component parts.

In determining strength of the appliance and safety requirements against overturning and drifting the total wind load shall be considered (see 3-6 and 3-7).

b) The plane of the exposed parts placed non perpendicularly to the wind direction (inclined boom for instance)

. on individual components with solid area

Where the wind blows at an angle θ to the longitudinal axis of a member (see fig. 2-2.2.1.3) :

- The load in the direction of wind is :

$$F = A \,.\, q \,.\, C_f \,.\, \sin^2 \theta$$

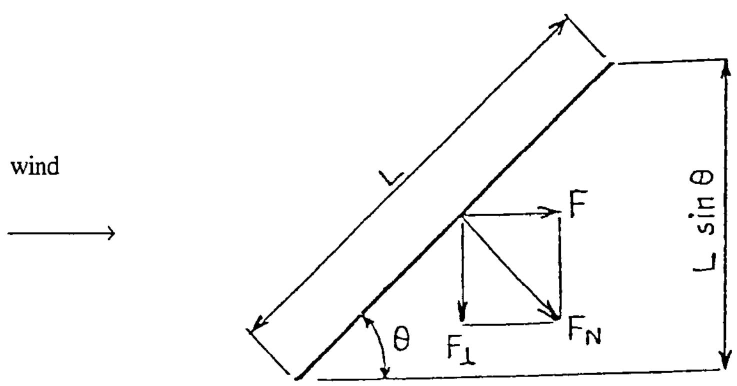
- The load in the direction perpendicular to the wind direction is :

$$F_\perp = A \,.\, q \,.\, C_f \,.\, \sin \theta \,.\, \cos \theta$$

- The load in the direction perpendicular to the longitudinal axis of the member is the resultant of F and $F_\perp$ and is :

$$F_N = A \,.\, q \,.\, C_f \,.\, \sin \theta$$

fig. 2-2.2.1.3



A' = L.b or L.D.
L = lenght
b = width
D = diameter

.../

A = area of the projected member on to a plane perpendicular to the wind direction = A'sin θ,
C_f = is in relation with the ratio L/b or L/D according to 2-2.2.1.4,
q = is given in 2-2.2.1.1.

In the case of very small values of the angle θ (< 20°), the calculation must be made with a minimum value of θ equal to 20° (to take into account unavoidable aerodynamic effects).

. on lattice girders and towers

Where the wind blows at an angle to the longitudinal axis of a lattice girder or tower, the wind load in the direction of the wind is obtained from :

$$F = A \cdot q \cdot C_f \cdot K_2 \qquad \text{in N}$$

where F, A, q, and C_f are as defined in 2-2.2.1.3a and

$$K_2 = \frac{\theta}{50\left(1.7 - \frac{Sp}{S}\right)}$$

which cannot be less than 0.35 or greater than 1

Where θ is the angle of the wind in degrees to the longitudinal axis of the girder or tower (θ ≤ 90°).

Sp is the area in m² of the bracing members of the girder or tower projected on to its windward plane.

S is the area in m² of all (bracing and main) members of the girder or tower projected on to its windward plane.

The value of K_2 is assumed to have lower and upper limits of 0.35 and 1.0 respectively. It is taken 0.35 whenever the calculated value < 0.35 and as 1.0 whenever the calculated value > 1.0.

2-2.2.1.4 SHAPE AND SHIELDING COEFFICIENTS

a) Shape coefficients for individual members, girders, etc.

Shape coefficients for individual members, single lattice girders and machinery houses are given in table T.2-2.2.1.4.1. The values for individual members vary according to the aerodynamic slenderness and, in the case of large box sections, with the section ratio.

Aerodynamic slenderness and section ratio are defined in figure 2-2.2.1.4.2.

The wind load on single lattice girders may be calculated on the basis of the coefficients for the individual members given in the top part of table T.2-2.2.1.4.1. In this case the aerodynamic slenderness of each member shall be taken into account. Alternatively the overall coefficients for lattice girders constructed of flat sided and circular sections given in the middle part of the table may be used.

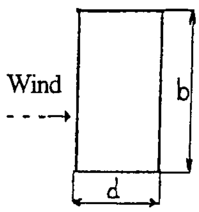
Where a lattice girder is made up of flat-sided or circular sections, or of circular sections in both flow regimes ($D \cdot V_S < 6\ m^2/s$ and $D \cdot V_S \geq 6 m^2/s$) the appropriate shape coefficients are applied to the corresponding frontal areas.

Where gusset plates of normal size are used in welded lattice construction no allowance for the additional area presented by the plates is necessary, provided the lengths of individual members are taken between the centres of node points.

Shape coefficients obtained from wind-tunnel or full-scale tests may also be used.

T.2-2.2.1.4.1

SHAPE COEFFICIENTS

Type	Description	b/d	Aerodynamic slenderness l/b or l/D *						
			< 5	10	20	30	40	50	> 50
Individual members	Rolled sections J]		1.15	1.15	1.3	1.4	1.45	1.5	1.60
	Rectangular hollow sections up to 356 mm square		1.4	1.45	1.5	1.55	1.55	1.55	1.6
	and 254 x 457 mm rectangular		1.05	1.05	1.2	1.3	1.4	1.5	1.6
	Other sections		1.30	1.35	1.60	1.65	1.70	1.80	1.80
	Circular sections where $D.V_S < 6\ m^2/s$		0.60	0.70	0.80	0.85	0.90	0.90	0.90
	$D.V_S \geq 6\ m^2/s$		0.60	0.65	0.70	0.70	0.75	0.80	0.80
	Rectangular hollow sections over 356 mm square and 254 x 457 mm rectangular 	1	1.55	1.75	1.95	2.10	2.20		
		2	1.40	1.55	1.75	1.85	1.90		
		0.5	1.0	1.20	1.30	1.35	1.40		
		0.25	0.80	0.90	0.90	1.0	1.0		
Single lattice girders	Flat-sided sections		1.60						
	Circular sections where $D.V_S < 6\ m^2/s$		1.10						
	$D.V_S \geq 6\ m^2/s$		0.80						
Machinery houses etc.	Closed rectangular structures		1.30						

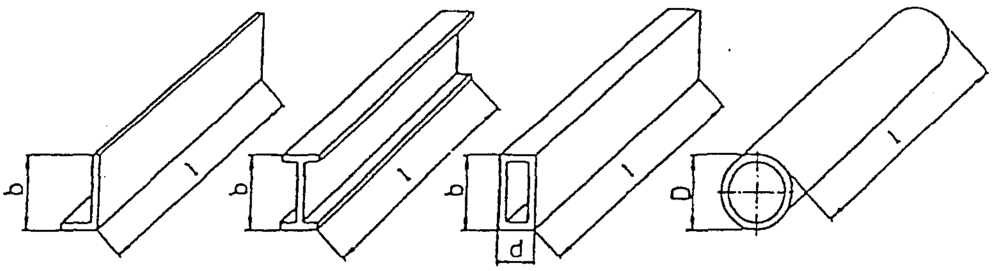
* see figure 2-2.2.1.4.2

Figure 2-2.2.1.4.2

DEFINITIONS : AERODYNAMIC SLENDERNESS, SOLIDITY RATIO, SPACING RATIO AND SECTION RATIO

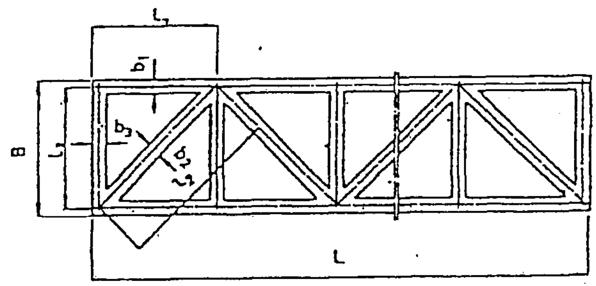
(I) Aerodynamic slenderness =

$$\frac{\text{length of member}}{\text{breadth of section across wind front}} = \frac{l}{b}{*} \quad \text{or} \quad \frac{l}{D}{*}$$

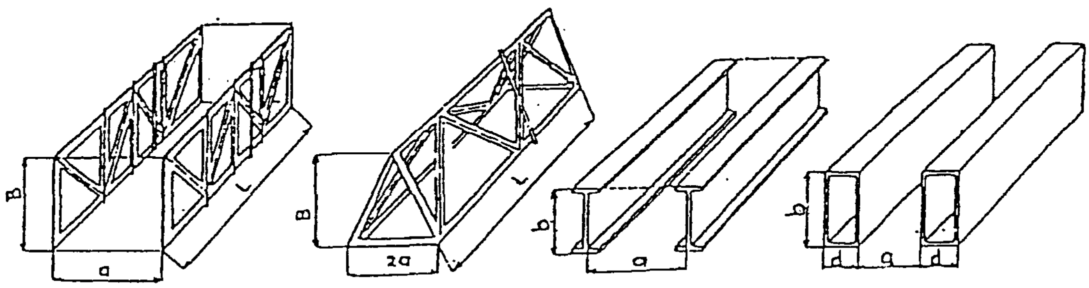


* In lattice construction the lengths of individual members are taken between the centres of adjacent node points. See diagram below.

$$\text{(II) Solidity ratio} = \frac{\text{area of solid parts}}{\text{enclosed area}} = \frac{A}{A_e} = \sum_{1}^{n} \frac{li \cdot bi}{L \cdot B}$$



$$\text{(III) Spacing ratio} = \frac{\text{distance between facing sides}}{\text{breadth of members across wind front}} = \frac{a}{b} \quad \text{or} \quad \frac{a}{B}$$



for "a" take the smallest possible value in the geometry of the exposed face.

$$\text{(IV) Section ratio} = \frac{\text{breadth of section across wind front}}{\text{depth of section parallel to wind flow}} = \frac{b}{d}$$

b) Shielding factors for multiple girders or members

Where parallel girders or members are positioned so that shielding takes place, the wind loads on the windward girder or member, and on the unsheltered parts of those behind it, are calculated using the appropriate force coefficients. The wind load on the sheltered parts is multiplied by a shielding factor η given in table T.2-2.2.1.4.3. Values of η vary with the solidity and spacing ratios as defined in figures 2-2.2.1.4.2.

Table T.2-2.2.1.4.3
SHIELDING COEFFICIENTS η

Spacing ratio a/b	Solidity ratio A/Ae					
	0.1	0.2	0.3	0.4	0.5	≥ 0.6
0.5	0.75	0.4	0.32	0.21	0.15	0.1
1.0	0.92	0.75	0.59	0.43	0.25	0.1
2.0	0.95	0.8	0.63	0.5	0.33	0.2
4.0	1	0.88	0.76	0.66	0.55	0.45
5.0	1	0.95	0.88	0.81	0.75	0.68
6.0	1	1	1	1	1	1

Where a number of identical girders or members are spaced equidistantly behind each other in such a way that each girder shields those behind it, the shiedling effect is assumed to increase up to the ninth girder and to remain constant thereafter. The wind loads are calculated as follows :

On the 1st. girder $F_1 = A.q.C_f$ in N

On the 2nd. girder $F_2 = \eta.A.q.C_f$ in N

On the nth girder (where n is from 3 to 8) $F_n = \eta^{(n-1)}.A.q.C_f$ in N

On the 9th and subsequent girders $F_9 = \eta^8.A.q.C_f$ in N

The total wind load is thus :

Where there are up to 9 girders $F_{total} = [1 + \eta + \eta^2 + \eta^3 + ... + \eta^{(n-1)}]\ A.q.C_f$

$= A.q.C_f \left(\frac{1 - \eta^n}{1 - \eta}\right)$ in N

.../

Where there are more than 9 girders $F_{total} = [1 + \eta + \eta^2 + \eta^3 + ... + \eta^8 + (n - 9)\eta^8]\ A.q.C_f$

$$= A.q.C_f \left[\left(\frac{1 - \eta^9}{1 - \eta}\right) + (n - 9)\ \eta^8 \right] \quad \text{in N}$$

Note :

The term η^x used in the above formula is assumed to have a lower limit of 0.10. It is taken as 0.10 when ever

$$\eta^x < 0.10$$

Lattice towers

For calculating the "face-on" wind load on square towers in the absence of a detailed calculation, the solid area of the windward face is multiplied by the following overall force coefficient :

For towers composed of flat sided sections $1.7.(1+\eta)$

For towers composed of circular sections

where $D.V_S < 6\ m^2/s$ $1.1.(1+\eta)$
where $D.V_S \geq 6\ m^2/s$ 1.4

The value of η is taken from table T.2-2.2.1.4.3 for $a/b = 1$ according to the solidity ratio of the windward face.

The maximum wind load on a square tower occurs when the wind blows on to a corner. In the absence of a detailed calculation, this load can be considered as 1.2 times that developed with "face-on" wind on one side.

2-2.2.2 SNOW AND ICE LOADS

For temperate areas, normal snow and ice loads can be assumed to have been included in the allowances for incrustation calculated in section 2-2.1.3.

For machines in areas with severe climatic conditions, or where specified by the purchaser, consideration should be given to the additional loads arising from snow and ice.

2-2.2.3 TEMPERATURE

Temperature effects need only by considered in special cases, e.g. for areas with extreme climatic conditions or when using materials with different expansion coefficients within the same component that are not free to expand or contract separately.

2-2.2.4 ABNORMAL TANGENTIAL AND LATERAL DIGGING FORCES

The abnormal digging force acting tangentially to the bucket-wheel or in the direction of the bucket-chain, is calculated from the starting torque of the drive motor or from the cut-off torque of the built in safety coupling taking into account the more unfavourable of the two cases listed below :

a) If the wheel or chain is not loaded :

In this case no account is taken of the power necessary to lift and/or move the material and the load due to the starting or cut-off torque of the motor is considered as a digging load.

b) If the wheel or chain is loaded according to 2-2.1.2.2, in this case the digging power can be reduced by the power required to lift and/or move the material.

The abnormal lateral digging force is calculated as in 2-2.1.4.2 (b) thereby considering a load of 0.3 times the abnormal tangential digging force.

If appropriate, this load can be calculated from the working torque of an existing cut-out device of the slewing or travelling mechanism and which should be at least equal to 1.1 times the sum of the torques due to the inclination of the machine (see 2-2.1.7) and to wind load for machines in operation (see 2-2.2.1.2).

Where a torque limit device is installed on the slewing mechanism, the mechanism must be equipped with a locking device preventing the slewing part rotating when out of service (due to wind force) if this rotation is dangerous.

2-2.2.5 BEARING FRICTION AND ROLLING RESISTANCES

a) Frictional forces need only to be calculated where they influence the size of structural components.

Following friction coefficient can be used in default of more precise values based on suppliers' specifications :

- for pivots and ball bearings	$\mu = 0.10$
- for structural parts with sliding friction	$\mu = 0.25$
- on wheels of rail-mounted machines	$\mu = 0.03$
- on wheels of crawler-mounted machines	$\mu = 0.10$
- between plate base and ground (crawler, shiftable conveyors)	$\mu = 0.60$

2-2.2.6 REACTION PERPENDICULAR TO THE RAIL DUE TO TRAVELLING OF THE APPLIANCE (LOAD CAUSED BY SKEWING)

For appliances on rails account must be taken of the reactions resulting from the travelling movement of the unit under a skewing angle, giving rise to a horizontal guide force Hy directed perpendicularly to the rail.

The force Hy can act on any guiding device (flange of wheel or separate guiding wheel) and is in equilibrium with the horizontal friction forces acting between wheels and rails.

To calculate the force Hy the relevant system properties shall be taken into account (see ISO 8686 clause 6.2.2 and annex F).

If no accurate calculation can be effected the force Hy should be taken as :

$$Hy = 0.2 \cdot Vy$$

where Vy is the maximum vertical load of the wheel or bogie.

2-2.2.7 NON-PERMANENT DYNAMIC EFFECTS

Inertia forces (due to the acceleration and braking of moving structural parts) which occur less than $2 \cdot 10^4$ times during the lifetime of the appliance should be checked as additional loads. They may be disregarded if their effect is less than the wind force during operation as defined in 2-2.2.1.

If the inertia forces are such that they have to be taken into account, the wind effect can be disregarded.

Note : As these forces are, in practice, present at the same time as the wind effects, it must be noted that actual accelerating and braking times are not the same as when still conditions prevail.

2-2.3 SPECIAL LOADS

2-2.3.1 CLOGGING OF CHUTES

The weight of material due to clogging should be calculated using a load which is equivalent to the capacity of the chute in question, with due reference to the angle of repose. The actual bulk density must be taken for calculation.

2-2.3.2 RESTING OF THE RECLAIMING DEVICE OR THE BOOM

Where safety devices are installed such as a slack rope detector for rope suspensions or pressure switches for hydraulic hoists, which prevent the full weight of the reclaiming device or the boom from coming to rest, the permissible resting force is to be calculated as a special load at 1.10 times the value set by the safety device.

Where such safety devices are not provided, the special load is to be calculated with the full resting weight.

2-2.3.3 FAILURE OF LOAD LIMITING DEVICES AS IN PARAGRAPH 2-2.1.2.1

In the case of failure on the part of an automatic load limiting to limit the useful loads on the conveyors, the output can be calculated as follows :

a) in the case of appliances without reclaiming device according to paragraph 1) (b), of item 2-2.1.2.1,

b) in the case of appliances with built-in reclaiming device according to paragraph 2) (a), of item 2-2.1.2.1.

For this purpose account need not be taken of the dynamic factor 1.1.

2-2.3.4 BLOCKING OF TRAVELLING DEVICES

The design of rail-mounted equipment must take into account the likelihood of bogies being blocked, e.g. by derailment or rail fracture. For the loads occurring under such conditions, the coefficient of friction between driven wheels and rails should be calculated as $\mu = 0.20$ provided that the drive motors can generate sufficient torque.

For equipment mounted on fixed rails, a wheel can be considered as blocked which cannot rotate but slides on the rail.

For equipment mounted on relocatable rails, blocking of a carrying wheel or bogie should be assumed as resulting from derailment or rail fracture. The maximum motive force is then determined by the non blocked wheels and it cannot exceed the force transferable by rail/wheel friction.

In the case of gantry with large rails center distance, equipped with special devices to check the respective displacement on both sides of the travelling mechanism, the calculation will be made taking into account the maximum possible difference on the position of bogies allowed by these special devices.

2-2.3.5 LATERAL COLLISION WITH THE SLOPE IN CASE OF BUCKET-WHEEL MACHINES

The maximum lateral resistance in bumping against the slope is determined by the safety coupling in the slewing gear or the kinetic energy of the superstructure. This load is to be applied in accordance with 2-2.1.4. In calculating the lateral resistance from the kinetic energy, a theoretical bracking distance of 30 cm and a constant braking deceleration are to be assumed.

2-2.3.6 WIND LOAD ON MACHINES OUT OF SERVICE

This is a maximum (storm) wind that the handling machine is designed to withstand in out of service conditions, as indicated, by the manufacturer.

It must be noted that, in this out of service position, the machine eventually need to be placed in a special parking area, in a defined position (boom orientation for instance) and could require the use of special devices such as rail clamps, anchors,...

The wind speed varies with the height above the surrounding level, and the geographical location. The wind load on the apparatus depends also on the degree of exposure to the prevailing winds.

For calculation and unless specified otherwise for handling appliances used in the open air the aerodynamic wind pressures to be used and the corresponding speeds, for "out of service" conditions are indicated in the table T.2-2.3.6.

Table T.2-2.3.6

OUT OF SERVICE AERODYNAMIC WIND PRESSURE

Height above surrounding level	Out of service aerodynamic wind pressure q	Approximate equivalent out of service design wind speed	
m	N/m²	m/s	km/h
0 to 20	800	36	130
20 to 100	1,100	42	150
more than 100	1,300	46	165

When calculating wind loads for out of service conditions the aerodynamic wind pressure shall be taken as constant over the vertical height intervals in table T.2-2.3.6.

The calculation procedure is detailed in chapter 2-2.2.1.1 (in service wind).

Where machines are to be permanently installed or used for extended periods in areas where wind conditions are exceptionally severe, the above figures may be modified by agreement between the manufacturer and purchaser in the light of local meteorological data.

2-2.3.7 BUFFER EFFECTS

For horizontal speeds below 0.7 m/s no account shall be taken of buffer effects. For speeds in excess of 0.7 m/s account must be taken of the reaction on the structure by collisions with buffers, when buffering is not made impossible by special devices.

It shall be assumed that the buffers are capable of absorbing the kinetic energy of the machine with operating load up to a certain fraction of the rated travelling speed V_t ; this fraction is fixed at minimum 0.7 V_t.

The resulting loads on the structure shall be calculated in terms of the deceleration imparted to the machine by the buffer in use.

2-2.3.8 LOADS DUE TO EARTHQUAKES

In general the structures of handling appliances do not have to be checked for seismic effects.

However, if local regulations or particular specifications so prescribe, special rules or recommendations must be applied in areas subject to earthquakes.

The supplier shall be advised of this requirement by the user of the installation who shall also provide the corresponding seismic spectra.

2-2.3.9 LOADS DURING ERECTION (OR DISMANTLING) OF THE MACHINE

In general, the loads during erection are less than they will be for the machine in operation.

Nevertheless, in some particular situations, according to the erection process, it can be necessary to check some structural (or mechanical) parts for a certain stage of the erection.

The checking is made by taking into account the actual supporting conditions and the eventual anchorages of the given structural part.

The permissible stresses are :

- stress allowed in case I for no wind condition
- stress allowed in case III for out of service wind condition.

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2-3 LOAD CASES FOR STRUCTURAL DESIGN

The main, additional and special loads mentioned in chapter 2-2 must be combined in load cases I, II and III according to table T.2-3.1 hereafter.

Loads should only be combined that can occur simultaneously to provide the highest resultant stresses.

We shall retain for case III the most unfavourable combination.

2-3.1 TABLE OF LOAD CASES

(see table on following page).

Table T.2-3.1
<u>LOAD COMBINATIONS</u>

Items		Main loads case I	Main and additional loads case II	Main, additional and special loads case III								Loads during erection
				1	2	3	4	5	6	7	8	
2-2.1.1	Dead loads	*	*	*	*	*	*	*	*	*	*	*
2-2.1.2	Material loads on conveyors, reclaiming devices and hoppers	*	*	*	*	*	*	*	......	*	*	
2-2.1.3	Incrustation	*	*	*	*	*	*	*	*	*	*	
2-2.1.4	Normal tangential and lateral digging forces	*	......	*	......	*	......	*				
2-2.1.5	Forces on the conveying elements	*	*	*	*	*	*	*	*	*	*	
2-2.1.6	Permanent dynamic effects	*	* (1)	*	*	*	*	*	......	*	*	
2-2.1.7	Loads due to inclination of working level	*	*	*	*	*	*	*	*	*	*	
2-2.2.1	Wind force during service	......	*	*	*	*	*	*	......	*	*	
2-2.2.2	Snow and ice (possibly)											
2-2.2.3	Temperature (possibly)											
2-2.2.4	Abnormal tangential and lateral digging forces	......	*									
2-2.2.5	Resistances due to friction and travel	......	*									
2-2.2.6	Reactions perpendicular to the rail	......	*									
2-2.2.7	Non-permanent dynamic effects	......	* (1)									
2-2.3.1	Chute blocking	......	......	*								
2-2.3.2	Bucket-wheel resting	......	......	......	*							
2-2.3.3	Failure of load limiting devices (see 2-2.1.2.1)	......	......	......	......	*						
2-2.3.4	Travelling device blocking	......	......	......	......	......	*					
2-2.3.5	Lateral collision with the slope (bucket-wheel)	......	......	......	......	......	......	*				
2-2.3.6	Wind force out of service	......	......	......	......	......	......	......	* (2)	......	......	*
2-2.3.7	Buffer effects	......	......	......	......	......	......	......	......	*		
2-2.3.8	Earthquake loads	......	......	......	......	......	......	......	......	......	*	
2-2.3.9	Loads during erection	......	......	......	......	......	......	......	......	......	......	*

(1) the most unfavourable should be considered for design.
(2) the mechanism must be equipped with a locking device preventing the slewing part from rotating (due to wind force), when out of service, if this rotation is dangerous.

.../

2-4 LOADS ENTERING INTO THE DESIGN OF MECHANISMS

2-4.1 GENERAL INFORMATION

A "mechanism" is understood to be a combination of elements intended to transform the power of the drive motor, usually electric motor, into work or movement of the unit's moving parts or the whole of the unit.

Included in this group are :

- reclaiming component and cutting component mechanisms (for example bucket-wheels, bucket-chains),
- belt conveyor mechanisms,
- slewing mechanisms (slewing connections and slewing drives),
- mechanisms for moving the whole unit (mobile frames, steering and traversing movement mechanisms),
- raising and lowering mechanisms (cable controlled mechanisms, hydraulic raising mechanisms).

In order to design a mechanism, two loads categories are examined :

- the average loads, taking into account the load fluctuations and the operating severity as a function of the operating time (calculation of fatigue or wearing),

- the maximum loads the mechanism must bear for short times with sufficient safety (calculation of strength).

2-4.2 LOADS DEFINITION

In general the loads dealt with in articles 2-2.1, 2-2.2 and 2-2.3 in the "structure" chapter as well as certain loads defined further on shall be taken into account in the various load combinations.

2-4.2.1 FRICTION RESISTANCES

Resistances induces by friction should be calculated by taking into account the maximum friction coefficients when defining a drive mechanism and the minimum friction coefficient when defining braking devices.

The following friction coefficients can be used :

- for (roller or ball) slewing bearings	$0.005 \leq \mu \leq 0.01$
- for pivots plain bearings	$\mu = 0.1$
- for rail wheels	
. with ball or roller bearings	$0.005 \leq \mu \leq 0.01$
. with plain bearings	$0.01 \leq \mu \leq 0.03$
- on wheels of crawler machines	$\mu = 0.1$
- between plate base and ground (crawler, shiftable conveyors)	$\mu = 0.6$

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2-5 LOAD CASES FOR THE DESIGN OF MECHANISMS

In order to take the load fluctuations into account, we consider the two following load cases :

Case I :

Load combinations which may appear simultaneously during normal operation.

Case II :

Load combinations in which additional or special loads, which may arise occasionally, are taken into account (in service or out of service).

2-5.1 TABLE OF LOAD CASES T.2-5.1 (*)

The load cases to be considered in cases I and II for the following mechanisms, are found in table T.2-5.1 :

. reclaiming component mechanism,
. belt conveyor mechanism,
. slewing mechanism,
. travelling mechanism (on rails or crawlers),
. lifting or luffing mechanism.

* The numbers of the corresponding loads refer to chapter 2-2 "structures".

Table T.2-5.1.1

<table>
<tr><th>Mechanism or mechanism parts</th><th colspan="2">Loads</th><th>Case I</th><th colspan="3">Case II</th></tr>
<tr><td rowspan="9">reclaiming unit mechanism</td><td>2-2.1.1</td><td>dead loads</td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
<tr><td>2-2.1.2.2</td><td>material loads in reclaiming device</td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
<tr><td>2-2.1.3</td><td>incrustation</td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
<tr><td>2-2.1.4</td><td>normal tangential and lateral digging forces</td><td>x</td><td></td><td>x</td><td>x</td></tr>
<tr><td>2-2.1.5</td><td>tension of bucket-chain, belt,...</td><td>x</td><td>x</td><td>x</td><td>x</td></tr>
<tr><td>-</td><td>friction resistance of the material between the bucket-wheel or bucket-chain and the chute</td><td>x</td><td>x</td><td></td><td></td></tr>
<tr><td>2-2.2.4</td><td>abnormal tangential and lateral digging forces</td><td></td><td>x (*)</td><td></td><td></td></tr>
<tr><td>2-2.3.2</td><td>resting of bucket-wheel or chain on ground or face</td><td></td><td></td><td>x</td><td></td></tr>
<tr><td>2-2.3.5</td><td>lateral collision of bucket-wheel with the slope</td><td></td><td></td><td></td><td>x</td></tr>
<tr><td rowspan="6">belt conveyor mechanism</td><td>2-2.1.1</td><td>dead loads</td><td>x</td><td colspan="3">x</td></tr>
<tr><td>2-2.1.2.1</td><td>material loads on conveyor (without dynamic coefficient)</td><td>x</td><td colspan="3"></td></tr>
<tr><td>-</td><td>belt conveyor motion resistances (according to ISO 5048)</td><td>x</td><td colspan="3"></td></tr>
<tr><td>2-2.1.5</td><td>belt tension (according to ISO 5048)</td><td>x</td><td colspan="3">x</td></tr>
<tr><td>2-2.3.3</td><td>abnormal material loads on conveyors (without dynamic coefficient) arising from failure of load limiting devices as in 2-2.1.2.1)</td><td></td><td colspan="3">x (*)</td></tr>
<tr><td colspan="2">if the output and elevation vary, take this into account when calculating the fatigue strength</td><td></td><td colspan="3"></td></tr>
</table>

(*) = referred to the maximum allowable torque

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Table T.2-5.1.2
<u>SLEWING MECHANISM</u>

Part of the mechanism		Loads	Case I	Case II working		Case II not working
<u>supporting components</u> (tracks with ball or roller bearing)	2-2.1.1	dead loads	x	x	x	x
	2-2.1.2	material loads (without dynamic factor)	x	x	x	
	2-2.1.3	incrustation (without dynamic factor)	x	x	x	x
	2-2.1.4	normal tangential and lateral digging forces	x		x	
	2-2.1.7	loads due to inclination of the working level	x	x	x	x
	2-2.1.6 / 2-2.2.1	permanent dynamic effects / in service wind : q in N/m² (1) (3) } * — * take the least favourable load	x	x	x	
	2-2.2.4	abnormal tangential and lateral digging forces		x		
	2-2.3.6	wind load on machine out of service (1) (3)				x
	2-2.3	special loads (when working) : 2-2.3.1, 2-2.3.2, 2-2.3.3, 2-2.3.4, 2-2.3.5, 2-2.3.7, 2-2.3.8 (take the load giving the least favourable combination)			x	
<u>drive components</u> (driving unit and ring gear)			(2)	(2)		
	2-2.1.4	normal tangential and lateral digging forces	x	x		
	2-2.1.6	permanent dynamic effects (forces due to acceleration and braking - slewing motion)	x			
	2-2.1.7	loads due to inclination of the working level	x	x		x
	2-2.2.1	in service wind : q/3 N/m²	x			
	2-2.2.1	in service wind : q in N/m² (1) (3)		x		
	2-2.3.6	wind load on machine out of service (1) (3)				x
	2-4.2.1	friction resistances	x	x		

Note (1) : unless otherwise specified, in service wind pression value q given in section 2-2.2.1.2 and out of service wind pression value q given in section 2-2.3.6 shall be used.

Note (2) : where a torque limit device is installed the loads when working are limited to the adjustment of the torque limiter, which will be at a minimum equal to the following values :

- if there is a separate locking device for the slewing part :
  1.1 (2-2.1.7 + 2-2.2.1)
- if there is no separate locking device and where a lock is to be effected by the brake of the driving unit :
  1.1 (2-2.1.7 + 2-2.3.6)

Note (3) : to take into account the non uniform aerodynamic pressure of the wind on the total area of the machine, wind force will be decreased by 50 % on one side of the slewing axis when calculation is least favourable.

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Table T.2-5.1.3

TRAVELLING MECHANISM (ON RAILS OR CRAWLERS)

Part of the mechanism	Loads		Case I	Case II		
				working		not working
supporting components (wheels, pins equalizers,...)	2-2.1.1	dead loads	x	x		x
	2-2.1.2	material loads (without dynamic factor)	x	x		
	2-2.1.3	incrustation (without dynamic factor)	x	x		x
	2-2.1.4	normal tangential and lateral digging forces	x	x		
	2-2.1.7	loads due to inclination of the working level	x	x		x
	2-2.2.1	in service wind : q in N/m² (1)	x	x		
	2-2.2.5	bearing friction and rolling resistances	x	x		
	2-2.2.6	reactions perpendicular to the rail due to movement of appliance	x	x		
	2-2.3.6	wind load on machine out of service (1)				x
	2-2.3	special loads (when working) : 2-2.3.1, 2-2.3.2, 2-2.3.3, 2-2.3.4, 2-2.3.5, 2-2.3.7, 2-2.3.8 (take the load giving the least favourable combination)		x		
moving components (driving unit)						(3)
	2-2.1.4	normal tangential and lateral digging forces	x	x		
	2-2.1.6	permanent dynamic effects (forces due to acceleration and braking - travelling motion)	x			
	2-2.1.7	loads due to inclination of the working level	x	x	x	x
	2-2.2.1	in service wind : q/3 N/m²	x			
	2-2.2.1	max. in service wind : q in N/m² (1)		x		
	2-2.2.6	reactions perpendicular to the rail due to movement of appliance	x	x		
	2-2.3.4	blocking of travelling devices (on rails)			x (2)	
	2-2.3.6	wind load on machine out of service (1)				x
	2-4.2.1	friction resistances	x	x		x

Note (1) : unless otherwise specified, in service wind pression value q given in section 2-2.2.1.2 and out of service wind pression value q given in section 2-2.3.6 shall be used.

Note (2) : loads corresponding to slipping of rail wheels in case 2-2.3.4 with friction coefficient between wheel and rail $\mu = 0.2$.

Note (3) : in the out of service situation, the brake of the driving unit and (eventually) the locking device (rail clamp for instance) have to prevent the machine from drifting (see 3-7).

Table 2-5.1.4
LUFFING MECHANISM

Mechanism	Loads		Case I	Case II		
				working *		not working
(cable winding mechanisms, hydraulic lifting mechanisms, screw lifting mechanisms)	2-2.1.1	dead loads	x	x	x	x
	2-2.1.2	material loads (without dynamic coefficient)	x	x	x	
	2-2.1.3	incrustation	x	x	x	x
	2-2.1.4	normal tangential and lateral digging forces	x			
	2-2.1.6	permanent dynamic effets	x	x	x	
	2-2.1.7	inclination of the working level	x	x	x	x
	2-2.2.1	wind in operation : q in N/m² (1)	x	x	x	
	2-2.2.4	abnormal tangential and lateral digging forces		x		
	2-2.3.6	wind out of service (1)				x
	2-2.3	special loads (when working) : 2-2.3.1, 2-2.3.2, 2-2.3.3 (take the load giving the least favourable combination)			x	
	2-4.2.1	friction resistances	x	x	x	

Note (1) : unless otherwise specified, in service wind pression value q given in section 2-2.2.1.2 and out of service wind pression value q given in section 2-2.3.6 shall be used.

* In some circumstances, the case II working load combination may be restricted by load limiting devices (to be noted that machine <u>must</u> stay stable, but, under overload, boom may descend after the limiting device has taken action).

Where there is no such device, as when the machine is out of service, the least favourable load combination must be considered.

<u>Note</u> : Precautions to be taken in the event of the failure of load limiting devices shall be specified by the manufacturer (see chapter 5 : "Safety").

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Chapter 3

CALCULATING THE STRESSES IN STRUCTURES

CHAPTER 3

CALCULATING THE STRESSES IN STRUCTURES

CONTENTS

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INTRODUCTION

The stresses set up in the various structural members are determined for the three load cases defined in section 2-3, and a check is made to ensure that, when compared with the critical stresses, the factor of safety v is adequate for the following three possible causes of failure :

- exceeding the elastic limit,
- exceeding the critical crippling or buckling load,
- and, eventually, exceeding the limit of endurance to fatigue.

The grade of the steel used must be stated and the physical properties, chemical composition and weldability must be guaranteed by the manufacturer of the material.

The permissible stresses for the materials used should be determined as stipulated in clauses 3-2, 3-3, 3-4 and 3-5 hereunder, based on the critical stresses for the material.

The critical stresses are those which correspond either to the elastic limit or the stress corresponding to the critical limit for elongation as appropriate, or to the critical stress for crippling or buckling, or, in the case of fatigue, to the stress for which the probability of survival, under tests, is 90 %.

The suitability of the selected material to resist brittle fracture should be assessed as outlined in section 3-1.

The stresses in the structural members should be calculated on the basis of the different load cases as defined in section 2-3 by applying conventional strength of materials calculation procedures.

The sections of metal to be considered shall be the gross sections (i.e. without deducting the areas of holes) for all parts which are subjected to compression loads (1), and the net sections (i.e. with the areas of holes deducted) for all parts subjected to tensile loads.

In the case of a member subjected to bending, a half-net section should be assumed, taking the net section in parts under tension and the gross section in parts under compression. To simplify the calculations, however, one may use either the section modulus of the net section or the section modulus computed for the half-net section, using as centre of gravity of the section that of the gross section.

(1) The area of the holes shall be included in the cross-sectional area only when they are filled by a rivet or a bolt.

.../

3-1 SELECTION OF STEEL TO RESIST BRITTLE FRACTURE

The usual calculations required by the design rules to assess the safety of the structure against yielding, crippling, buckling and fatigue failure do not guarantee safety against brittle fracture.

In order to obtain sufficient safety against brittle fracture, a steel grade has to be chosen to suit on the conditions influencing brittle fracture.

3-1.1 ASSESSMENT OF THE FACTORS WHICH INFLUENCE BRITTLE FRACTURE

The most important influences on the sensitivity to brittle fracture in steel structures are :

A. Combined effect of longitudinal residual tensile stresses with stresses from dead load.
B. Thickness of the member.
C. Influence of cold.

Influences A., B. and C. are evaluated with coefficients Z_A, Z_B and Z_C respectively. The required steel quality is then determined from on the sum of these coefficients.

3-1.1.1 INFLUENCE A : COMBINED EFFECT OF LONGITUDINAL RESIDUAL TENSILE STRESSES WITH STRESSES FROM DEAD LOAD

Equations for lines I, II, III in figure 3-1.1.1.

Line I : no welds, or only transverse welds

$$Z_A = \frac{\sigma_G}{0.5 \cdot \sigma_a} - 1$$

valid only for $\sigma_G \geq 0.5 \cdot \sigma_a$

Line II : longitudinal welds

$$Z_A = \frac{\sigma_G}{0.5 \cdot \sigma_a}$$

Line III : accumulation of welds

$$Z_A = \frac{\sigma_G}{0.5 \cdot \sigma_a} + 1$$

where :

σ_a = permissible tensile stress with respect to the elastic limit, loading case I

σ_G = tensile stress from permanent load

The likehood of brittle fracture is increased by high stress concentrations, in particular by 3-axial tensile stresses, as is the case with an accumulation of welds.

If members are stress relieved after welding (approx 600 - 650°C) line I can be used for all types of welds.

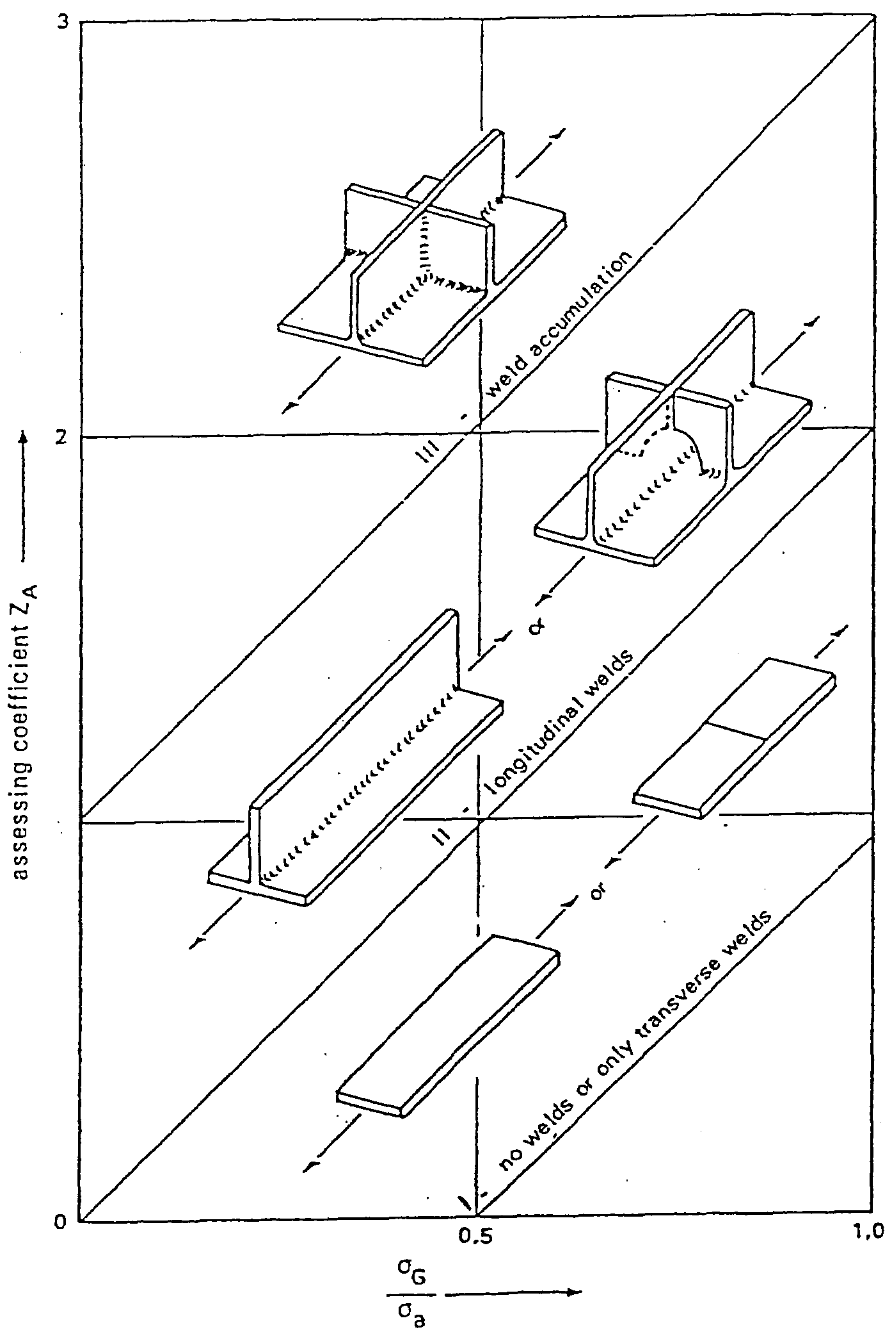


Figure 3-1.1.1
Z_A IN TERMS OF STRESSES AND WELDS

3-1.1.2 INFLUENCE B : THICKNESS OF MEMBER

from t = 5 to t = 20 mm :

$$Z_B = \frac{9}{2500} \cdot t^2$$

from t = 20 to t = 100 mm :

$$Z_B = 0.65 \cdot \sqrt{t - 14.81} - 0.05$$

where t = thickness of member in mm

t mm	Z_B	t mm	Z_B	t mm	Z_B
		16	0.9	60	4.3
5	0.1	20	1.45	65	4.55
6	0.15	25	2.0	70	4.8
7	0.2	30	2.5	75	5.0
8	0.25	35	2.9	80	5.2
9	0.3	40	3.2	85	5.4
10	0.4	45	3.5	90	5.6
12	0.5	50	3.8	95	5.8
15	0.8	55	4.0	100	6.0

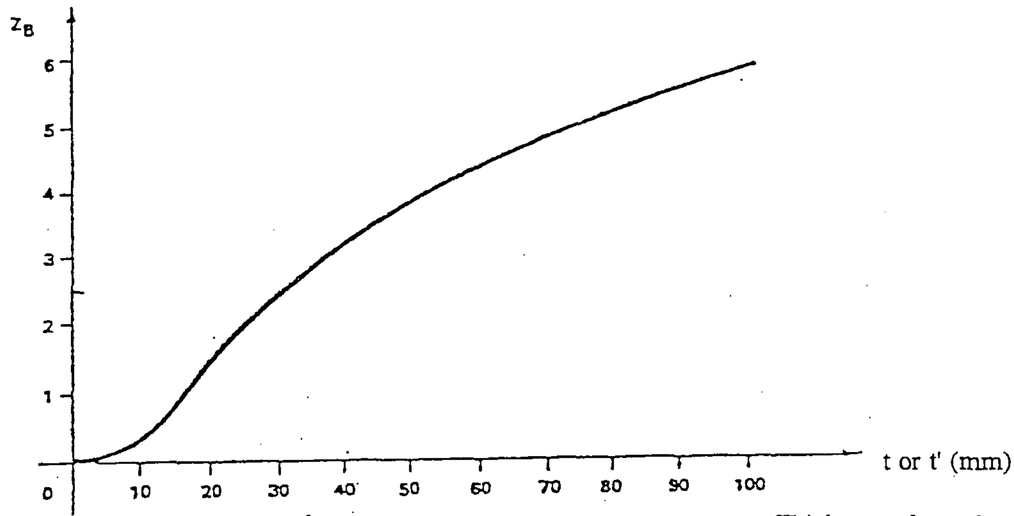


Figure 3-1.1.2
ASSESSING COEFFICIENT $Z_B = f(t)$

For round, square or rectangular solid sections an equivalent thickness t' is to be used. This is :

for round sections : $t' = \frac{d}{1.8}$

for square sections : $t' = \frac{t}{1.8}$

for rectangular sections : $t' = \frac{b}{1.8}$

.../

where b represents the larger side of the rectangle and the ratio of the side $b/t \leq 1.8$. For $b/t > 1.8$, $t' = t$.

3-1.1.3 INFLUENCE C : INFLUENCE OF COLD

The lowest temperature at the place of erection of the appliance determines the classification. This temperature is generally lower than the working temperature.

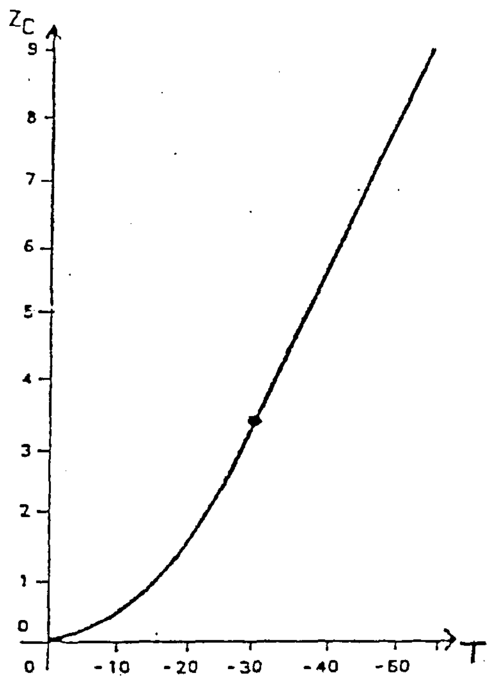
from T = 0 °C to T = - 30 °C

$$Z_C = \frac{6}{1,600} \cdot T^2$$

from T = - 30 °C to T = - 55 °C

$$Z_C = \frac{-2.25 \cdot T - 33.75}{10}$$

where T = temperature at the place of erection in °C



Temperature in °C

Figure 3-1.1.3
ASSESSING COEFFICIENT $Z_C = f(T)$

T °C	Z_C	T °C	Z_C
0	0.0	- 30	3.4
- 5	0.1	- 35	4.5
- 10	0.4	- 40	5.6
- 15	0.8	- 45	6.7
- 20	1.5	- 50	7.9
- 25	2.3	- 55	9.0

3-1.2 DETERMINATION OF THE REQUIRED STEEL QUALITY GROUP

It is the sum of assessing coefficients from paragraph 3-1.1 which determines the minimum required quality for the steel structure.

Table T.3-1.2 shows the classification of the quality groups in relation to the sum of the assessing coefficients.

If the sum of the assessing coefficients is higher than 16 or if the required steel quality cannot be obtained, special measures must be taken to obtain the necessary safety against brittle fracture.

This should be done in conjunction with the steel suppliers and metallurgical experts.

.../

Table T.3-1.2
CLASSIFICATION OF QUALITY GROUPS IN RELATION TO THE SUM OF THE ASSESSING COEFFICIENTS

Sum of the assessing coefficients from paragraph 3-1.1 $\Sigma Z = Z_A + Z_B + Z_C$	Quality group corresponding in table T.3-1.3
≤ 2	1
≤ 4	2
≤ 8	3
≤ 16	4

3-1.3 QUALITY OF STEELS

The quality of steels in these design rules is the property of steel to exhibit a 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 temperature.

Table T.3-1.3 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, where no value must be below 20 Nm/cm^2.

The notch ductility is to be determined in accordance with V-notch impact tests to ISO R 148 and Euronorm 45-63.

Steels of differents quality groups can be welded together.

T_c is the test temperature for the V-notch impact test

T is the temperature at the place of erection of the appliance

T_c and T are not directly comparable as the V-notch impact test imposes a more unfavourable condition than the loading on the appliance in or out of service.

Table T.3-1.3
QUALITY GROUPS

Quality group	Notch ductility measured in ISO sharp notch test ISO R 148 in Nm/cm²	Test tempe-rature T_C °C	Steels corresponding to the quality group designation of steels (1)	Standard
1	-	-	Fe 360 - A Fe 430 - A	ISO 630
			St 37 - 2 St 44 - 2	DIN 17100
			E 24 - 1	NF A 35-501
			43 A *	BS 4360 1986
2	35	+ 20°	Fe 360 - B Fe 430 - B Fe 510 - B	ISO 630
			R St 37 - 2 St 44-2	DIN 17100
			E 24 - 2 E 28 - 2 E 36 - 2	NF A 35-501
			40 B 43 B 50 B *	BS 4360 1986
3	35	0°	Fe 360 - C Fe 430 - C Fe 510 - C	ISO 630
			St 37 - 3N St 44 - 3N St 52 - 3N	DIN 17100
			E 24 - 3 E 28 - 3 E 36 - 3	NF A 35-501
			40 C 43 C * 50 C	BS 4360 1986
4	35	- 20°C	Fe 360 - D Fe 430 - D Fe 510 - D	ISO 630
			St 37 - 3N St 44 - 3N St 52 - 3N	DIN 17100
			E 24 - 4 E 28 - 4 E 36 - 4	NF A 35-501
			40 D 43 D * 50 D	BS 4360 1986

* The test requirements of steels to BS 4360 do not in all cases correspond with the ISO and other national standards, and the guaranteed impact test properties for steels to BS 4360 may be different to other steels in the same quality group. Impact test properties are stated in BS 4360 and where the requirements are different from those guaranteed in BS 4360, agreement must be obtained from the steel suppliers.

(1) In the present rules, steels are designed by the ISO symbols, essentially Fe 360, Fe 430, Fe 510.

3-1.4 SPECIAL RULES

In addition to the above provisions for the choice of the steel quality, the following rules are to be observed :

1) Non killed steels of group 1 shall be used for load carrying structures only in case of rolled sections and tubes not exceeding 6 mm wall thickness.

2) Members of more than 50 mm thickness, shall not be used for welded load carrying structures unless the manufacturer has a comprehensive experience in the welding of thick plates. The steel quality and its testing has in this case to be determined by specialists.

3) If parts are cold bent with a radius/plate thickness ratio < 10, the steel quality has to be suitable for folding or cold flanging.

3-2 CHECKING WITH RESPECT TO THE ELASTIC LIMIT

For this check, a distinction is made between the actual members of the structure and the welded or bolted joints.

3-2.1 STRUCTURAL MEMBERS

3-2.1.1 MEMBERS SUBJECTED TO SIMPLE TENSION OR COMPRESSION

1) For steels where the ratio between the elastic limit σ_E and the ultimate tensile strength σ_R is ≤ 0.7, the computed stress σ must not exceed the maximum permissible stress σ_a obtained by dividing the elastic limit stress σ_E by a coefficient ν_E which depends upon the load case as defined under section 2-3.

The values of ν_E and the permissible stresses are :

Load case	I	II	III
ν_E	1.5	1.33	1.2 *
permissible stresses σ_a	$\frac{\sigma_E}{1.5}$	$\frac{\sigma_E}{1.33}$	$\frac{\sigma_E}{1.2}$ *

For currently manufactured carbon steels of grades (ISO) Fe 360 - Fe 430 - Fe 510, the critical stress σ_E is conventionally taken as that which corresponds to a permanent elongation of 0.2 %.

* Note : For handling machines, $\nu_E = 1.2$ in case III, while this coefficient is equal to 1.1 for lifting appliances in the same case III (see FEM rules - Section I).

<u>Characteristic values for steels of current manufacture</u>

Steel grade	σ_E N/mm²	σ_R N/mm²	$\frac{\sigma_E}{\sigma_R}$	E N/mm²	G N/mm²	αt $\frac{mm}{mm.K}$
Fe 360	240	370	0.65	$21 \cdot 10^4$	$8.1 \cdot 10^4$	$1.2 \cdot 10^{-5}$
Fe 430	280	440	0.64	$21 \cdot 10^4$	$8.1 \cdot 10^4$	$1.2 \cdot 10^{-5}$
Fe 510	360	520	0.69	$21 \cdot 10^4$	$8.1 \cdot 10^4$	$1.2 \cdot 10^{-5}$

Table T.3-2.1.1
VALUES OF σ_E AND σ_a FOR STEELS Fe 360 - Fe 430 - Fe 510

Steel grade	Elastic limit σ_E	Maximum permissible stresses σ_a		
		Case I	Case II	Case III
	N/mm^2	N/mm^2	N/mm^2	N/mm^2
Fe 360	240	160	180	200
Fe 430	280	187	210	233
Fe 510	360	240	270	300

2) For steels with high elastic limit, where the ratio σ_E / σ_R is greater than 0.7, the use of the ν_E coefficients does not ensure a sufficient margin of safety. In this case a check has to be made that the permissible stress σ_a given by the formula below is not exceeded :

$$\sigma_a = \frac{\sigma_E + \sigma_R}{\sigma_{E(Fe\ 510)} + \sigma_{R\ (Fe\ 510)}} \cdot \sigma_{a(Fe\ 510)}$$

where

σ_E and σ_R are the elastic limit and the ultimate tensile strength of the steel considered

$\sigma_{E(Fe\ 510)}$ and $\sigma_{R\ (Fe\ 510)}$ these same stresses for steel Fe 510, i.e. 360 N/mm^2 and 520 N/mm^2

$\sigma_{a(Fe\ 510)}$ the permissible stress for steel Fe 510 in the case of loading considered

3-2.1.2 MEMBERS SUBJECTED TO SHEAR

The permissible stress in shear τ_a has the following value :

$$\tau_a = \frac{\sigma_a}{\sqrt{3}}$$

σ_a being the permissible tensile stress.

3-2.1.3 MEMBERS SUBJECTED TO COMBINED STRESSES - EQUIVALENT STRESS

σ_x and σ_y being respectively the two normal stresses and τ_{xy} the shear stress at a given point, a check shall be made :

1) that each of the two stresses σ_x and σ_y is less than σ_a and that τ_{xy} is less than τ_a

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2) that the equivalent stress σ_{cp} is less than σ_a, i.e. :

$$\sigma_{cp} = \sqrt{\sigma_x{}^2 + \sigma_y{}^2 - \sigma_x.\sigma_y + 3\tau_{xy}{}^2} \leq \sigma_a$$

When using this formula, a simple method is to take the maximum values σ_x, σ_y and τ_{xy}. But, in fact, such a calculation leads to too great an equivalent stress when it is impossible for the maximum values of each of the three stresses to occur simultaneously.

Nevertheless, this simple calculation method, being conservative, is always acceptable.

If the equivalent stress is to be calculated more precisely, it is necessary to determine the most unfavourable practical combination that may occur. Three checks must then be made by calculating successively the equivalent stress resulting from the three following combinations :

$\sigma_{x\,max}$ and the corresponding stresses σ_y and τ_{xy}

$\sigma_{y\,max}$ and the corresponding stresses σ_x and τ_{xy}

$\tau_{xy\,max}$ and the corresponding stresses σ_x and σ_y

Note : It should be noted that when two out of the three stresses are approximately of the same value, and greater than half the permissible stress, the most unfavourable combination of the three values may occur in different loading cases from those corresponding to the maximum of each of the three stresses.

Special case :

- Tension (or compression) combined with shear

The following formula should be checked : $\sqrt{\sigma^2 + 3\tau^2} \leq \sigma_a$

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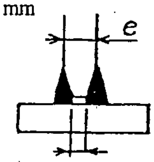
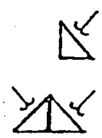
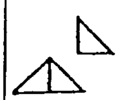
3-2.2 WELDED JOINTS

3-2.2.1 WELD QUALITIES

The types of weld most commonly used for handling appliances are butt welds, double bevel butt welds (K welds) and fillet welds, of ordinary or special quality (S.Q.) as specified below.

Weld testing is also stipulated for certain types of joint.

Table T.3-2.2.1
WELD QUALITIES

Type of weld	Weld quality	Execution of weld	Example of symbols 1)	Weld testing	Symbol
full depth butt weld	special quality (S.Q.)	root of weld scraped (or trimmed) before making sealing run. No end craters. Weld must be trimmed by grinding parallel to the direction of load until it is flush with the plate surface		check (e.g. with X-rays) over 100 % of seam length	P 100
	ordinary quality	root of weld scraped (or trimmed) before making sealing run. No end craters		as for S.Q. but solely under tensile stress when σ_{max} calculated $\geq 0.8\ \sigma_a$ (σ_a in function of κ)	P 100
				otherwise random check over at least 10 % of seam length (for example X-rays)	P 10
K-weld in angle formed by two parts with bevel on one of the parts to be joined at location of seam	special quality (S.Q.)	root of weld scraped (or trimmed) before making weld on other side. Weld edges without undercutting and ground if necessary. Full penetration welds		check that for tensile loads the plate perpendicular to the direction of the forces is free from lamination (for example ultrasonic examination)	D
	ordinary quality	width clear of weld penetration between the two welds ≤ 0.2 e with maxi 3 mm  ≤ 0.2 e <u>and</u> < 3 mm	K		
fillet welds in the angle formed by two parts	special quality (S.Q.)	welded edges without undercutting and ground if necessary		check that for tensile loads the plate perpendicular to the direction of the forces is free from lamination (for example ultrasonic examination)	D
	ordinary quality				

1) weld symbols are taken from ISO 2553, this symbol means root of weld scraped

3-2.2.2 MAXIMUM PERMISSIBLE STRESSES

In welded joints, it is assumed that the deposited metal has at least as good characteristics as the adjacent parent metal.

It must be verified that the stresses developed, in the cases of longitudinal tension and compression, and equivalent stresses do not exceed the permissible stresses σ_{aw} given in table T.3-2.2.2.

For shear in the welds, the permissible stress τ_{aw} is given by :

$$\tau_{aw} = \frac{\sigma_a}{\sqrt{2}}$$

However, for certain types of loading, particularly transverse stresses in the welds, the maximum permissible equivalent stress is reduced.

Table T.3-2.2.2 summarizes the maximum permissible values, for certain steels, according to the type of loading (and the load case).

Table T.3-2.2.2
MAXIMUM PERMISSIBLE STRESSES IN WELDS σ_{aw} (N/mm²)
STEELS Fe 360 - Fe 430 - Fe 510

Types of loading	Fe 360			Fe 430			Fe 510		
	case I	case II	case III	case I	case II	case III	case I	case II	case III
LONGITUDINAL AND EQUIVALENT STRESSES FOR ALL TYPES OF WELDS	160	180	200	187	210	233	240	270	300
TRANSVERSE TENSILE STRESSES 1) butt-welds (S.Q. or C.Q.) and special quality K-welds	160	180	200	187	210	233	240	270	300
2) ordinary quality K-welds	140	158	175	164	184	204	210	236	263
3) fillet welds (S.Q. or C.Q.)	113	127	141	132	149	165	170	191	212
TRANSVERSE COMPRESSIVE STRESSES 1) butt-welds and K-welds (S.Q. or C.Q.)	160	180	200	187	210	233	240	270	300
2) fillet welds (S.Q. or C.Q.)	130	146	163	152	171	189	195	220	244
SHEAR (S.Q. OR C.Q.)	113	127	141	132	149	165	170	191	212

Complementary information on welded joints is given hereafter.

3-2.2.3 COMPLEMENTARY INFORMATION ON STRESSES IN WELDED JOINTS

Determining the stresses in welds is a highly complex problem primarily because of the great number of possible configurations welded joints can assume.

For this reason it is not possible, as the matter stands at present, to lay down precise directives in these Rules for the Design of Handling Appliances. Indeed, both the volume and the subject matter of rules relating to welding would be difficult to fit into the general context of the present design rules. It was consequently decided to include only the following general indications :

1) All methods of calculation assume of necessity a properly executed joint, i.e. a weld with correct penetration and a good shape, so that the joint between the components to be assembled and the weld seam is free from discontinuity or sudden change of section as well as from craters or notches due to undercutting.

   The design of the weld must be adapted to the forces to be transmitted, and specialized literature on the subject should be consulted.

   It should be noted that the strength of a welded joint is significantly improved if the surface of the weld is finished by careful grinding.

2) There is no need to take into consideration stress concentrations due to the design of the joint or residual stresses.

3) The permissible stresses in welds are those determined under clause 3-2.2.2 and the equivalent stress σ_{cpw} in the case of combined stress (tensile or compressive) σ and shear stress τ is given by the formula :

$$\sigma_{cpw} = \sqrt{\sigma^2 + 2\tau^2} \leq \sigma_{aw} \quad \text{(given in 3-2.2.2)}$$

   In cases involving dual stresses σ_x and σ_y and the shearing stress τ_{xy}, the following formula is applied :

$$\sigma_{cpw} = \sqrt{\sigma_x{}^2 + \sigma_y{}^2 - \sigma_x \cdot \sigma_y + 2\tau_{xy}{}^2} \leq \sigma_{a\,w}$$

4) In a fillet weld, the thickness of the section considered is the depth of the weld to the bottom of the throat and its length is the effective length of the weld less the end craters.

   This length need not be reduced if the joint closes on to itself or if special precautions are taken to limit the effect of the craters.

5) Attention is drawn to the fact that it seems to be reliably established that fatigue failures in welded joints seldom occur in the weld seam itself but usually beside it in the parent metal.

.../

Therefore, in general the stresses σ_{min} and σ_{max} for the fatigue strength calculations for the parent metal beside the weld seam, must be computed using the classical methods for calculating the strength of materials.

In order to verify the fatigue strength of the weld itself, it is generally held that it is sufficient to confirm that the weld is capable of transmitting the same loads as the adjacent parent metal.

This rule is not obligatory however when the parts jointed are generously dimensioned in relation to the forces actually transmitted. When this is the case the weld seam need only be dimensioned in accordance with those forces, with the proviso that a fatigue check should then be performed in accordance with chapter 3-4.5.2.1.

Whatever the case it is emphasised that the size of a weld should invariably be in proportion to the thickness of the assembled parts.

3-2.3 BOLTED JOINTS

3-2.3.1 GENERAL

3-2.3.1.1 DEFINITIONS

Mobile equipment for continuous handling of bulk materials includes a large range of both sizes and types of structure which, in turn, can be subjected to a wide variety of loads.

Various structural and mechanical engineering disciplines have evolved simplified rules and codes of practice for the use of bolts with the types of structure and loadings most commonly encountered in each particular field.

Bulk material handling machines often include design details of many types, and it is the responsibility of the designing engineer to apply existing codes of practice only where appropriate.

National standards and codes of practice, though often similar, do vary considerably in detail between countries. Where National standards exist, these may be used in order to comply with local regulations and to suit the bolts, nuts, etc available in the country of manufacture and use.

In an attempt to avoid problems aristing from particular national interpretations of certain terms and phrases, some definitions are given in the following chapters.

In this section, the term "bolt" may be taken to include all ISO metric bolts, screws, studs and threaded fasteners. In every case, for each type and grade of bolt, the appropriate nut and washers should be used.

3-2.3.1.2 DESIGN

Bolted joints should be designed on the basis of a realistic assumption of the distribution of internal forces, having regard to relative stiffnesses. Such an assumption should correspond with the direct load paths through the elements of connections.

Where members are connected to the surface of the web or flange of a section, the local ability of the web or flange to transfer the applied forces should be checked and stiffening provided where necessary.

Ease of fabrication and erection should be considered in the design of joints and splices. Attention should be paid to clearances necessary for tightening fasteners, welding procedures, subsequent inspection, surface treatment and maintenance.

.../

The ductility of steel assist the distribution of forces generated within a joint. Therefore residual stresses need not usually be calculated.

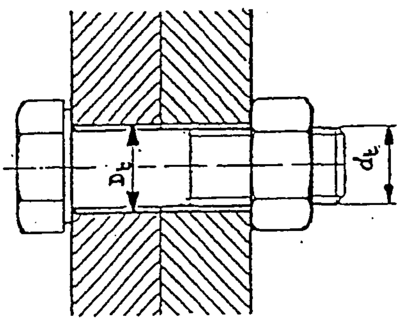
When different forms of fasteners are used, or when welding and fasteners are combined to carry a shear load, then one form of connection should normally be designed to carry the total load.

High duty bolted joints

For high duty bolted joints, especially under fatigue loading, where pretension is critical and embedding of the mating surfaces and local joint design details have a significant effect, a more rigorous analysis of the joint and bolt stresses may be required.

Clearances

The recommended clearance for bolts in holes is based on ISO 273. Values for the more common bolt diameters are tabulated below for reference.



Thread diameter d_t (mm)	Clearance hole D_t (mm)		
	fine	medium	coarse
12	13	13.5	14.5
16	17	17.5	18.5
20	21	22	24
24	25	26	28
30	31	33	35
36	37	39	42

Notes : Where necessary to avoid interference between the edge of the hole and the under-head fillet of the bolt, a chamfer is recommended.

Especially for higher grade bolts under tension in medium and coarse clearance holes, it may be necessary to check the bearing stress in the plate material under the bolt head and nut, using hardened steel washers as necessary, to transfer the load.

Tolerances for fitted bolts

Where fitted bolts are used, the holes must be drilled and reamed. The tolerance for the hole should be H11.

Note : The term "Fitted bolt" is used to refer to any bolt in a close fitting hole. In some countries special bolts are made for use in close fitting holes (e.g. DIN 7968 + DIN 7999) and these should be used if appropriate.

3-2.3.1.3 CONTROLLED TIGHTENING

Joints tightened by controlled means must not be placed under externally applied structural load until the jointing process is complete. This may include the tightening of more than one joint.

Care should be taken to ensure that bolts are correctly tightened and that the preload induced in the first bolts in a joint is not lost as the rest of the bolts are tightened.

The use of spring or bite type lock washers is not recommended for bolts tightened by controlled means. The extra friction and joint settlement involved causes unpredictable results and loss of preload.

Each system of controlled tightening has its associated tolerance range, and both the designer and user must be aware of this to ensure that the bolt is neither over-tensioned nor the joint under-tensioned and liable to slip or fatigue.

During controlled torque tightening, consideration must be given to the effects of thread and nut face friction. Under the combined effect of tension and torsional loading, the nominal tensile stress should not exceed 80 % of the elastic limit to take account of the scatter in the tightening process.

3-2.3.1.4 MAXIMUM PERMISSIBLE STRESS

The maximum permissible stress, σ_a, for the bolt material should be calculated in accordance with section 3-2.1.1.

The permissible working stresses for bolts grades 4.6/5.6/8.8 and 10.9 are summarized in table T.3-2.3.3.5.

3-2.3.2 ORDINARY BOLTS

3-2.3.2.1 DEFINITIONS

Ordinary bolts, often referred to as black bolts, are carbon steel bolts with wide geometric tolerances, normally of the lower strength grades (e.g. DIN 601, BS 4190 and ISO 4016).

Note : the term "black" no longer relates to the appearances of the product, which may look either black or bright in its finished state.

3-2.3.2.2 APPLICATION

Ordinary or black bolts are normally used in medium or coarse clearance holes in secondary joints that do not transmit heavy loads.

They must not be used in joints subject to fatigue loading.

The calculated stresses must not exceed :

$\sigma_{ab} = 0.625\ \sigma_a$ in tension

$\tau_{ab} = 0.5\ \sigma_a$ in shear

$\sigma_{an} = 1.0\ \sigma_a$ in bearing (see note)

Note : The permissible stress in bearing should be based on the yield stress of the bolt or the plate material, whichever has the lowest value.

Wherever there is a risk of nuts or bolts coming loose, they should be secured by the use of locking washers or equivalent.

3-2.3.3 HIGHER GRADE BOLTS : PRECISION BOLTS

Higher grade bolts, sometimes called precision bolts, are turned or cold finished to close tolerances (e.g. DIN 931, BS 3692 and ISO 4014).

3-2.3.3.1 PRECISION BOLTS IN TENSION

Bolts in tension can be used in fine, medium or coarse clearance holes, as required.

A) Bolts - tightened by not controlled means

This simple tightening method should only be used on secondary joints.

The maximum calculated tensile stress shall not exceed 0.625 σ_a.

Under fluctuating or reversing loads, the stress range (σ_{max} - σ_{min}) must be less than 10 % of the ultimate stress σ_R, with a mean stress level less than 15 % σ_R.

Bolts under other cyclic loading conditions should be tightened by controlled means.

Note : Spring or bite-type lock washers are not normally effective when used with bolts of grade 8.8 and above, where the hardness of the bolt is similar to or greater than that of the lock washer.

B) Bolts - tightened by controlled means

Care should be taken to ensure that the joint is not subjected to shear loading unless the conditions in section 3-2.3.4 are met.

a) Tightening without tightening torque

The maximum tensile stress σ_b shall not exceed 0.8 σ_E.

$$\sigma_b \leq \sigma_{ab} = 0.8\ \sigma_E$$

b) Tightening with twist

The maximum combined stress σ_b shall be checked as follows :

$$\sigma_b = \sqrt{\sigma_p^2 + 3\tau_b^2} \leq \sigma_{ab} = 0.8\ \sigma_E$$

$$\text{where } \tau_b = \frac{2 d_2 \sigma_p}{d_t} \cdot \left(\frac{p_a}{\pi\ d_2} + 1.155\ \mu\right)$$

where

σ_p = theoretical tensile stress under the tightening effect

τ_b = torsional stress under the tightening effect

d_2 = diameter of the root of the thread

d_t = nominal diameter of the bolt

p_a = thread pitch

μ = friction coefficient in the threads *

σ_E = elastic limit of the bolt metal

* It is necessary to get (from the manufacturer for instance) the actual value of μ. Just for information, the friction coefficient is normally between 0.10 and 0.18.

C) Maximum allowable external loads

Under maximum loading F_1 the following two checks must be satisfied :

a) The elastic limit, σ_E, of the bolt must not be exceeded

Determine :

$$\sigma_1' = \sqrt{\sigma_E^2 - 3\tau_b^2}$$

and check that :

$$F_1 \leq \frac{(\sigma_1' - \sigma_p) \cdot S_b}{K \cdot K' \cdot \delta_b}$$

where

F_1 = load on joint

S_b = bolt area at root of thread

K.K' = safety coefficients given in section b) : table T.3-2.3.3.1 below

$$\delta_b = \frac{\Delta l_1}{\Delta l_1 + \Delta l_2}$$

where

Δl_1 = shortening of the elements being tightened

Δl_2 = lengthening of the bolt under the action of the tightening force

σ_p, τ_b = as defined above

Note 1 : For assembled steel parts, the equivalent section S_{eq} to be considered for Δl_1 should be :

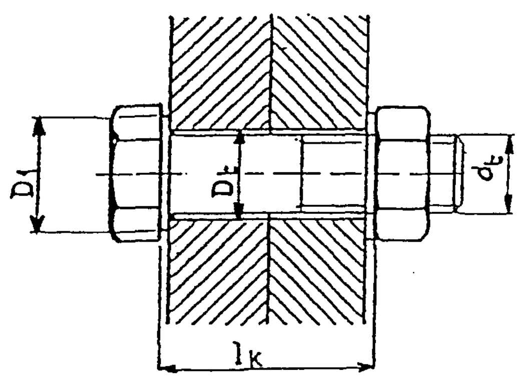
$$S_{eq} = \frac{\pi}{4}\left[\left(D_1 + \frac{l_k}{10}\right)^2 - D_t^2\right]$$

where

D_1 = bearing diameter under bolt head

l_k = length of tightening

D_t = diameter of bolt holes



Note 2 : For bolts whose shank diameter differs significantly from the root diameter of the thread, or where there is an appreciable threaded length contained within the bolt stretch length, a complete calculation of Δl_2 should be made.

b) Joint separation sould not occur, check that

$$\frac{F_1}{S_b} \leq \frac{\sigma_p}{K'.K''\ .\ (1-\delta_b)\ .\ \Omega}$$

where $\Omega = 1.1$ to allow for 10 % tolerance in tensioning equipment

Safety coefficients K, K' and K" :

K depends on the surface finish of the mating parts (K = 1 for machined surface)

K' factor of safety on elastic limit (see table T.3-2.3.3.1)

K" factor of safety on joint separation (see table T.3-2.3.3.1)

Table T.3-2.3.3.1

	case I	case II	case III
K'	1.5	1.33	1.2
K"	1.3	1	1

Note : The coefficients K' and K" should be applied to the most unfavourable condition arising from the scatter in applying the initial tightening effort.

3-2.3.3.2 PRECISION BOLTS IN SHEAR (BOLTS IN FITTED HOLES ONLY)

Preferably used (with or without preload) for joints subject to non-fluctuating loads.

Where fatigue loading occurs, friction grip joints should be used. See section 3-2.3.4.

.../

A) Bolts in fine, medium or coarse clearance holes

The calculated shear stress must not exceed $0.5\ \sigma_a$ as for ordinary bolting.

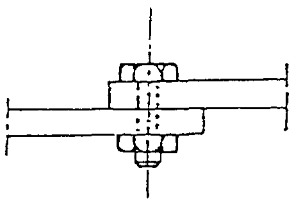
Note : The permissible stress in bearing should be based on the yield stress of the bolt or plate material, whichever has the lowest value.

B) Bolts in fitted holes

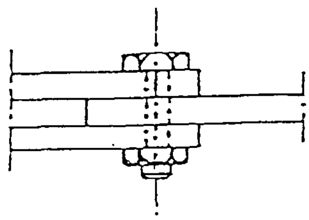
The calculated shear stress τ must be :

for single shear $\tau \leq 0.6\ \sigma_a$

for double shear $\tau \leq 0.8\ \sigma_a$



single shear



double shear

3-2.3.3.3 PRECISION BOLTS IN COMBINED TENSION AND SHEAR (BOLTS IN FITTED HOLES ONLY)

A check shall be made that :

in tension $\sigma \leq 0.625\ \sigma_a$

in single shear $\tau \leq 0.6\ \sigma_a$

in double shear $\tau \leq 0.8\ \sigma_a$

and that $\sqrt{\sigma^2 + 3\tau^2} \leq 0.625\ \sigma_a$

3-2.3.3.4 PRECISION BOLTS IN BEARING (BOLTS IN FITTED HOLES ONLY)

The bearing capacity of a bolt in any joint shall be taken as the lesser of the bearing capacity of the bolt and the connected plates.

A bolt can only be considered to be effective in bearing when none of the threaded portion is in contact with the joint plates.

However, care must be taken to ensure that there is still sufficient threaded length to enable the joint to be correctly tightened.

Bearing capacity

The bearing pressure shall not exceed :

1.3 σ_a for fitted bolts in single shear

1.75 σ_a for fitted bolts in double shear.

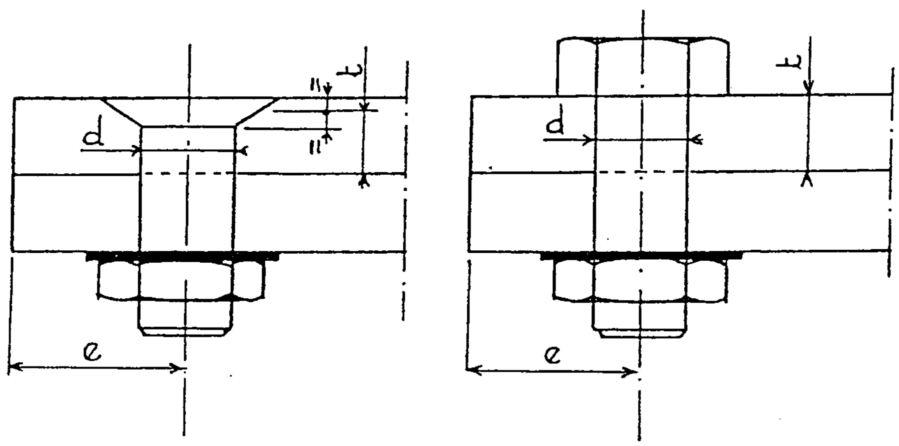
The bearing area for the bolt shall be considered as :

$A_b = d_t \cdot t$

where

d_t = nominal bolt diameter

t = the thickness of the connected plate less half the depth of the chamfer, if appropriate



The bearing area for the plate shall be considered as :

$$A_p = d_t \cdot t \leq \frac{1}{2}\, e \cdot t$$

where d_t and t are as above and e is the edge distance from the centre of the bolt hole to the nearest plate edge in the direction in which the bolt will shear.

Table T.3-2.3.3.5

PERMISSIBLE WORKING STRESSES FOR ORDINARY AND HIGHER GRADE BOLTS - SUMMARY

ISO bolt grade	load case	permissible tensile stress	bolts in fitted holes				bolts in non fitted holes	
			single shear		double shear			
			permissible shear stress	permissible bearing stress	permissible shear stress	permissible bearing stress	permissible shear stress	permissible bearing stress
		(0.625 σ_a)	(0.6 σ_a)	(1.3 σ_a)	(0.8 σ_a)	(1.75 σ_a)	(0.5 σ_a)	(1.0 σ_a)
		N/mm²	N/mm²	N/mm²	N/mm²	N/mm²	N/mm²	N/mm²
4.6	I	100	96	208	128	280 *	80	160
	II	113	108	235	144	316 *	90	180
	III	125	120	260 *	160	350 *	100	200
5.6	I	125	120	260 *	160	350 *	100	200
	II	141	135	293 *	180	395 x	113	226
	III	156	150	325 *	200	438 x	125	250 *
8.8	I	267	256	555 x	341	747 x	213	427 x
	II	301	289	626 x	385	842 x	241	481 x
	III	333	320	693 x	427	933 x	267	533 x
10.9	I	375	360	780 x	480	1050 x	300	600 x
	II	423	406	880 x	541	1184 x	338	677 x
	III	469	450	975 x	600	1313 x	375	750 x

* : bearing stress exceeds elastic limit of plate material Fe 360 steel (i.e. 240 N/mm²)

x : bearing stress exceeds elastic limit of plate material Fe 510 steel (i.e. 360 N/mm²)

3-2.3.4 HIGH TENSILE STEEL BOLTS WITH CONTROLLED TIGHTENING USED FOR FRICTION GRIP JOINTS

3-2.3.4.1 GENERAL

This type of joint is recommended for joints subject to fatigue where the primary loads are parallel to the joint faces.

The joint is made using high tensile steel bolts, in conjunction with high strength nuts and hardened steel washers. These elements are tightened to a specified minimum shank tensionso that loads can be transferred between connected parts by friction and not by shear or bearing on the bolt.

At the time of assembly, the mating surfaces must be free from paint, oil, dirt, loose rust or scale and any burrs and other defects which would prevent solid seating of the joint faces and interfere with the development of friction between them.

Where specific National codes or standards exist these may be used in place of this section.

3-2.3.4.1.1 BOLT QUALITY

Bolts used for this type of joint must have a high elastic limit, and the ultimate tensile strength σ_R must be greater than the values given hereunder :

σ_E 0.2 % N/mm²	σ_R N/mm²	ISO bolt grade
640	800	8.8
900	1,000	10.9
1,080	1,200	12.9

The diameter of the bolt hole shall be no more than 2 mm larger than that of the bolt.

3-2.3.4.1.2 TENSILE STRESS AREA

When determining the stress in the bolt, the tensile stress area shall be calculated by taking the arithmetic mean of the core (minor) diameter and the effective thread diameter. These values are given in the following table :

nominal diameter (mm)	8	10	12	14	16	18	20	22	24	27	30
tensile stress area (mm^2)	36.6	58	84.3	115	157	192	245	303	353	459	561

3-2.3.4.1.3 WHASHERS

A friction grip joint with high tensile steel bolts must always include two hard steel washers, one under the bolt head, one under the nut. Where these whashers have a 45° bevel on the internal rim, this must be turned towards the bolt head or nut. The washers must have been heat treated so that their hardness is at least equal to those of the bolt material.

3-2.3.4.1.4 BOLT TIGHTENING

The value of the tension induced in the bolt must reach the value determined by calculation and must be applied by an accurately controlled method.

Where a controlled torque method is used, the torquing device must be regularly calibrated against a known standard for each size of bolt used.

The torque required, M_a, can be derived from the equation :

$$M_a = 1.10\, C \,.\, d_t \,.\, F$$

d_t is the nominal bolt diameter

F is the clamping force to be induced in the bolt

C is a coefficient which includes the effects of thread form and thread and nut/head to washer friction. It is necessary to get (from the supplier for instance), the actual value of C to take it into account in the calculation of the tightening torque.

Note 1 : The use of grease in the thread and between nut/head and washer decreases the value of coefficient C and allows for a better idea of the exact value of this coefficient. But it is absolutely necessary to ensure that no grease is present between the plates to be joined. The use of MoS_2 grease (molykote for instance) is strongly recommended because MoS_2 sticks and stays where it has been applied to, while other lubricants tend to creep and enter between the plates to be joined. Furthermore the used of MoS_2 leads to lowest friction values for bolt tightening.

Note 2 : Where corrosion problems are anticipated, hot galvanized or sherardised bolts can be used. In that case, brittleness due to galvanizing has to be considered.

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3-2.3.4.1.5 DETERMINATION OF THE STRESSES IN THE MEMBERS JOINED

For members subject to compression, the stress is calculated on the gross section (cross-sectional area of the holes not deducted).

For members subjected to tension there are two cases :

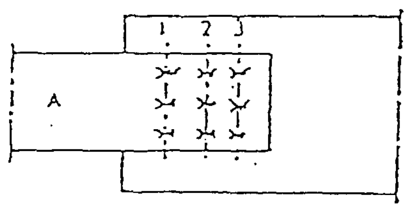
1st case : bolts set in a single row, perpendicular to the direction of the load ; the following conditions must be checked :

a) the total load on the gross section

b) 80 % of the total load on the net section (cross-sectional area of holes deducted)

2nd case : several rows of bolts perpendicular to the direction of the load.

The most heavily loaded section (corresponding to row 1 for the member A - see figure) must be analysed and the following two conditions checked :



a) the total load on the gross section

b) on the net section, the total load from the other rows (i.e. in the case of the figure, 2/3 of the total load of the joint) to which 80 % of the load taken by row 1 is added.

This assumes that the load within a row is equally divided amongst all the bolts and that the number of rows of bolts is small because if there are too many, the last bolts carry little load. It is therefore recommended that not more than two rows of bolts should be used or exceptionally three.

3-2.3.4.2 EXTERNAL LOADS ACTING IN THE PLANE OF THE JOINT (LOAD TYPE T)

The permissible load per bolt, T_a, which can be transmitted through the joint is determined by the equation (T_a values are given in tables T.3-2.3.4.5.4) :

$$T_a = \frac{\mu \cdot F}{\nu_T} \cdot m$$

where

μ is the friction coefficient from table T.3-2.3.4.2

F is the clamping force in the joint per bolt given in tables T.3-2.3.4.5.4

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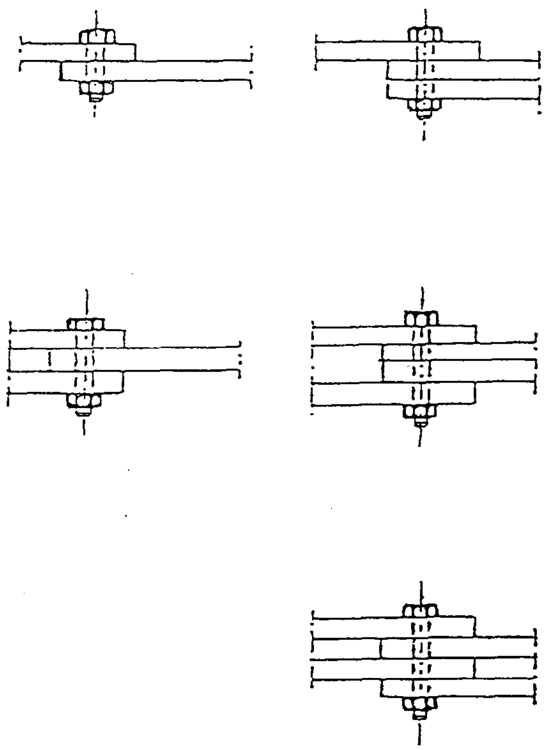
ν_T is the safety coefficient against slipping :
1.4 for the case I loading
1.25 for the case II loading
1.1 for the case III loading

m is the number of friction surfaces (see fig. 3-2.3.4.2)

Fig. 3-2.3.4.2
<u>NUMBER OF EFFECTIVE FRICTION SURFACES</u>



1 friction surface m = 1

2 friction surfaces m = 2

3 friction surfaces m = 3

Table T.3-2.3.4.2 : <u>Joint friction coefficient</u>

The coefficient of friction used for each joint depends on the method of preparation of the joint surfaces.

Before assembly, the surfaces must be thoroughly cleaned of all paint and oil. Loose rust, dirt and mill scale must be completely removed by thorough brushing with a clean wire brush.

Where surfaces have been machined, metal sprayed or otherwise treated, the coefficient of friction must be determined by test.

Table T.3-2.3.4.2
VALUES OF μ

joined material	normally prepared surfaces (degreasing and brushing)	(1) specially prepared surfaces (for example flame-cleaned, shot or sand-blasted or coated by non slip paint)
Fe 360	0.30	0.50
Fe 430	0.30	0.50
Fe 510	0.30	0.55 (2)

(1) such a special preparation is recommended for friction grip joints.
(2) if coated by non slip paint, the value of μ is 0.5

3-2.3.4.3 EXTERNAL LOADS ACTING PERPENDICULAR TO THE PLANE OF THE JOINT (LOAD TYPE N)

The load distribution within the joint must be evaluated and the effect of any external axial loads on the bolts must be checked in accordance with section 3-2.3.3.1.

3-2.3.4.4 JOINTS WITH EXTERNAL FORCE COUPLE (LOAD TYPE M)

If the bolted joint is subjected to an external couple of forces, the increased tensile loading within the joint must be determined and the stress in the highest loaded bolts added to the existing tensile (type N) load.

3-2.3.4.5 COMBINED EXTERNAL LOADINGS ON FRICTION GRIP JOINTS

For combined loading (load type T, N & M) two checks must be made :

a) That, for the most highly stressed bolt, the sum of the tensile stresses due to N and M loadings remains less than the permissible tensile stress as defined in 3-2.3.3. This requirement is met where the external tensile force doesn't exceed the values given in table T.3-2.3.4.5.1.

Table T.3-2.3.4.5.1
PERMISIBLE ADDITIONAL $N_{add.perm}$ TENSILE FORCE PER BOLT

load case		
I	II	III
0.6 F	0.7 F	0.8 F

where F is the clampint force per bolt.

b) That the mean load T per bolt which is transmitted by friction is less than the following value :

b1) $T \leq \left(0.2 + \dfrac{N_{add.perm} - N_{real}}{N_{add.perm}} \cdot 0.8\right) T_a$

where T_a is the admissible load per bolt if there is no additional external tensile force (3-2.3.4.2).

T_a values are given in table T.3-2.3.4.5.4.

b2) $T \leq \dfrac{\mu \cdot (F - N)}{\nu_T} \cdot m$

The additional tensile force N increases the bolt stress after tightening by a certain sum which depends on the elasticity of the bolt and of the compressed members. This relation can be taken into account by the "coefficient of elongation" which depends, for solid steel plates and for the types of bolts normally used in construction engineering, on the length of tightening l_k and the diameter of the bolt d_t.

For the normal case where the bolt is pretightened with :

$$\sigma_a = 0.7\ \sigma_E$$

the permissible additional tensile force N_a can be calculated with the following formula :

$$N_a = \frac{0.12\ \sigma_E \cdot Fs}{\nu_a \cdot \Phi}$$

in which

σ_E = elastic limit of the bolt metal

ν_a = safety coefficient for the load cases ($\nu_{aI} = 1.5$; $\nu_{aII} = 1.33$; $\nu_{aIII} = 1.2$)

Φ = coefficient of elongation on the basis of the ratio l_k/d_t according to table T.3-2.3.4.5.2

Fs = stress section of the bolt (see section 3-2.3.4.1.2)

Table T.3-2.3.4.5.2
COEFFICIENT OF ELONGATION

l_k = length of tightening d_t = diameter of the bolt

l_k/d_t	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0	7.5
Φ	0.43	0.42	0.40	0.38	0.36	0.33	0.32	0.30	0.29	0.27	0.26	0.25	0.24	0.22	0.21

Table T.3-2.3.4.5.3 gives permissible additional tensile forces N_a for different bolts materials, diameters and load cases.

The following table, T.3-2.3.4.5.4, gives per bolt and per friction surface, the values of the transmissible forces in the plane parallel to that of the joint for various friction coefficients for the steels Fe 360, Fe 430 and Fe 510.

To apply these values, the number of effective friction surfaces as indicated in 3-2.3.4.2 must be determined.

Table T.3-2.3.4.5.3

PERMISSIBLE ADDITIONAL TENSILE FORCES N_a IN kN
FOR BOLTS AFTER TIGHTENING WITH CLAMPING FORCE F IN kN

Bolt material : ISO grade 8.8
σ_R = 800 N/mm²

σ_E = 640 N/mm²

tightening $\sigma_a = 0.7\ \sigma_{E(0.2)}$

tightening length	d_t = 16 mm F = 70 kN			d_t = 20 mm F = 110 kN			d_t = 24 mm F = 158 kN		
	load case			load case			load case		
l_k (mm)	I	II	III	I	II	III	I	II	III
10	18.8	21.2	23.5	29.2	32.9	36.5			
20	19.6	22.1	24.5	29.9	33.7	37.3	42.7	48.2	53.4
30	20.9	23.5	26.1	31.4	35.4	39.2	44.1	49.7	55.1
40	22.3	25.2	27.9	33.0	37.2	41.3	45.9	51.8	57.4
50	24.5	27.7	30.7	34.8	39.3	43.6	48.0	54.1	60.0
60	25.9	29.2	32.4	38.0	42.9	47.5	50.2	56.6	62.8
70	27.5	31.0	34.4	39.2	44.2	49.0	54.0	60.8	67.4
80	29.8	33.6	37.2	41.8	47.2	52.3	55.9	63.0	69.9
90	31.2	35.2	39.0	43.3	48.8	54.1	58.3	65.8	72.9
100	32.9	37.0	41.0	46.5	52.4	58.1	60.9	68.7	76.2

Bolt material : ISO grade 10.9
σ_R = 1,000 N/mm²

σ_E = 900 N/mm²

tightening $\sigma_a = 0.7\ \sigma_{E(0.2)}$

tightening length	d_t = 16 mm F = 99 kN			d_t = 20 mm F = 154 kN			d_t = 24 mm F = 222 kN		
	load case			load case			load case		
l_k (mm)	I	II	III	I	II	III	I	II	III
10	26.4	29.8	33.1	41.0	46.3	51.3			
20	27.6	31.1	34.5	42.0	47.4	52.5	60.0	67.7	75.0
30	29.4	33.1	36.7	44.1	49.7	55.1	62.0	69.9	77.5
40	31.4	35.4	39.3	46.4	52.4	58.0	64.6	72.9	80.8
50	34.5	38.9	43.1	49.0	55.3	61.3	67.5	76.1	84.3
60	36.5	41.1	45.6	53.5	60.3	66.8	70.6	79.6	88.3
70	38.6	43.6	48.3	55.1	62.2	68.9	75.9	85.6	94.8
80	41.9	47.2	52.3	58.8	66.3	73.5	78.6	88.7	98.3
90	43.9	49.5	54.9	60.8	68.6	76.0	82.0	92.5	102.5
100	46.2	52.0	57.7	65.3	73.7	81.7	85.7	96.6	107.1

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Table T.3-2.3.4.5.4

TRANSMISSIBLE FORCES IN THE PLANE OF THE JOINT

ALL FORCES GIVEN PER BOLT AND PER FRICTION SURFACE T_a IN kN

bolts to ISO class 8.8 - σ_R = 800 N/mm² - σ_E = 640 N/mm²

bolt dia-meter	tensile stress area	clam-ping force	applied force c*=0,14	normally prepared surfaces using steels Fe 360, Fe 430, Fe 510 μ = 0.30			specially prepared surfaces using steels Fe 360, Fe 430, Fe 510 μ = 0.50			specially prepared surfaces using steels Fe 510 μ = 0.55		
d_t mm	Fs mm²	F kN	Ma Nm	I	II	III	I	II	III	I	II	III
10	58	26	40	5.6	6.2	7.1	9.3	10.4	11.8	10.2	11.4	13.0
12	84.3	37	68	7.9	8.9	10.1	13.2	14.8	16.8	14.5	16.3	18.5
14	115	52	112	11.1	12.5	14.2	18.6	20.8	23.6	20.4	22.9	26.0
16	157	70	172	15.0	16.8	19.1	25.0	28.0	31.8	27.5	30.8	35.0
18	192	86	238	18.4	20.6	23.5	30.7	34.4	39.1	33.8	37.8	43.0
20	245	110	336	23.6	26.4	30.0	39.3	44.0	50.0	43.2	48.4	55.0
22	303	136	460	29.1	32.6	37.1	48.6	54.4	61.8	53.4	59.8	68.0
24	353	158	584	33.9	37.9	43.1	56.4	63.2	71.8	62.1	69.5	79.0
27	459	205	852	43.9	49.2	55.9	73.2	82.0	93.2	80.5	90.2	102.5
30	561	249	1,150	53.3	59.7	67.9	88.9	95.6	113.1	97.8	109.5	124.5

bolts to ISO class 10.9 - σ_R = 1000 N/mm² - σ_E = 900 N/mm²

bolt dia-meter	tensile stress area	clam-ping force	applied force c*=0,14	normally prepared surfaces using steels Fe 360, Fe 430, Fe 510 μ = 0.30			specially prepared surfaces using steels Fe 360, Fe 430, Fe 510 μ = 0.50			specially prepared surfaces using steels Fe 510 μ = 0.55		
d_t mm	Fs mm²	F kN	Ma Nm	I	II	III	I	II	III	I	II	III
10	58	37	57	7.9	8.9	10.1	13.2	14.8	16.8	14.5	16.3	18.5
12	84.3	53	98	11.4	12.7	14.5	18.9	21.2	24.1	20.8	23.3	26.5
14	115	73	157	15.6	17.5	19.9	26.1	29.2	33.2	28.7	32.1	36.5
16	157	99	244	21.2	23.8	27.0	35.4	39.6	45.0	38.9	43.6	49.5
18	192	121	335	25.9	29.0	33.0	43.2	48.4	55.0	47.5	53.2	60.5
20	245	154	474	33.0	37.0	42.0	55.0	61.6	70.0	60.5	67.8	77.0
22	303	191	647	40.9	45.8	52.1	68.2	76.4	86.8	75.0	84.0	95.5
24	353	222	820	47.6	53.3	60.5	79.3	88.8	100.9	87.2	97.7	111.0
27	459	289	1,200	61.9	69.4	78.8	103.2	115.6	131.4	113.5	127.2	144.5
30	561	350	1,617	75.0	84.0	95.5	125.0	140.0	159.0	137.5	154.0	175.0

* c = 0.14 is used only as an example value. Where this coefficient is known or given by the supplier, the values of Applied Torque must be adjusted accordingly.

In the above tables, σ_a is limited to a maximum value of 0.7 σ_E.

In some circumstances, higher values based on a maximum of 0.8 σ_E can be used, but there is a higher risk of bolt failure, and bolts tightened to 0.8 σ_E must not be additionally loaded in tension by an external load.

For bolts with lower strength grades, the values given above may be multiplied by the ratio of their yield or 0.2 % proof stresses.

3-2.4 ROPES (GUY AND STAY ROPES)

Only static ropes are to be considered here, i.e. guy and stay ropes without any pulleys or sheaves (fixed or mobile on the rope).

The safety of these ropes must be ensured against the risk of breaking for the forces of case II of loading (main and additional loads) with a safety coefficient of minimum 3 against the breaking load of the rope.

Winch ropes with pulleys and sheaves and requiring replacement in the event of wearing (active ropes) are examined in chapter 4 Mechanisms.

It must be noted that static ropes with one or more equalizing sheaves for example are to be considered as active ropes (running ropes, see 4-2.2) and their safety coefficient must be 6 or above.

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3-3 CHECKING STABILITY OF PARTS SUBJECT TO CRIPPLING AND BUCKLING

3-3.1 CHECKING STABILITY OF STRUTS AND COLUMNS SUBJECT TO BUCKLING

The guiding principle shall be that parts subject to crippling must be designed with the same safety margin as that adopted for the strength calculation ; in other words, having determined the practical crippling stress, the maximum permissible stress shall be the crippling stress divided by the appropriate coefficient 1.5 or 1.33 or 1.2 specified in 3-2.1.1.

The choice or a practical method of calculation is left to the manufacturer who must state the origin of the method chosen.

Where the method chosen involves multiplying the computed stress by a crippling coefficient ω dependant upon the slenderness ratio of the member and then checking that this amplified stress remains less than a certain allowable stress, the value to be chosen for this allowable stress shall be as specified in 3-2.1.1.

The values of ω, as a function of the slenderness ratio λ, are given in the tables below for the following cases :

Table T.3-3.1.1 :	rolled sections in	Fe 360 steel
Table T.3-3.1.2 :	rolled sections in	Fe 510 steel
Table T.3-3.1.3 :	tubes in	Fe 360 steel
Table T.3-3.1.4 :	tubes in	Fe 510 steel

Determination of effective lengths for calculating the slenderness ratio λ

1 - In the ordinary case of bars pin joined at both ends and loaded axially, the effective length is taken as the length between between points of articulation.

2 - For an axially loaded bar encastered at one end and free at the other the effective length is taken as twice the length of the bar.

3 - For intermediate cases where uncertainty exists at present about the effect of fixity on members in compression, the effects of fixity are ignored and the member should be designed as if it were pin jointed at both ends, with the effective length being taken as the length between points of intersection of axes.

The case of bars subjected to compression and bending

In the case of bars loaded eccentrically or loaded axially with a moment causing bending in the bar :

- either check the following two formulae :

$$\frac{F}{S} + \frac{M_f\, v}{I} \leq \sigma_a$$

and

$$\frac{\omega F}{S} + 0.9 \frac{M_f \, v}{I} \leq \sigma_a$$

where :

F is the compressive load applied to the bar

S is the section area of the bar

M_f is the bending moment at the section considered

v is the distance of the extreme fibre from the neutral axis

I is the moment of inertia

- or perform the precise calculation in terms of the deformations sustained by the bar under the combined effect of bending and compression, the necessary calculation being effected either by integration or by successive approximations.

Table T.3-3.1.1

VALUE OF THE COEFFICIENT ω IN TERMS OF THE SLENDERNESS RATIO λ
FOR ROLLED SECTIONS IN Fe 360 STEEL

λ	0	1	2	3	4	5	6	7	8	9
20	1.04	1.04	1.04	1.05	1.05	1.06	1.06	1.07	1.07	1.08
30	1.08	1.09	1.09	1.10	1.10	1.11	1.11	1.12	1.13	1.13
40	1.14	1.14	1.15	1.16	1.16	1.17	1.18	1.19	1.19	1.20
50	1.21	1.22	1.23	1.23	1.24	1.25	1.26	1.27	1.28	1.29
60	1.30	1.31	1.32	1.33	1.34	1.35	1.36	1.37	1.39	1.40
70	1.41	1.42	1.44	1.45	1.46	1.48	1.49	1.50	1.52	1.53
80	1.55	1.56	1.58	1.59	1.61	1.62	1.64	1.66	1.68	1.69
90	1.71	1.73	1.74	1.76	1.78	1.80	1.82	1.84	1.86	1.88
100	1.90	1.92	1.94	1.96	1.98	2.00	2.02	2.05	2.07	2.09
110	2.11	2.14	2.16	2.18	2.21	2.23	2.27	2.31	2.35	2.39
120	2.43	2.47	2.51	2.55	2.60	2.64	2.68	2.72	2.77	2.81
130	2.85	2.90	2.94	2.99	3.03	3.08	3.12	3.17	3.22	3.26
140	3.31	3.36	3.41	3.45	3.50	3.55	3.60	3.65	3.70	3.75
150	3.80	3.85	3.90	3.95	4.00	4.06	4.11	4.16	4.22	4.27
160	4.32	4.38	4.43	4.49	4.54	4.60	4.65	4.71	4.77	4.82
170	4.88	4.94	5.00	5.05	5.11	5.17	5.23	5.29	5.35	5.41
180	5.47	5.53	5.59	5.66	5.72	5.78	5.84	5.91	5.97	6.03
190	6.10	6.16	6.23	6.29	6.36	6.42	6.49	6.55	6.62	6.69
200	6.75	6.82	6.89	6.96	7.03	7.10	7.17	7.24	7.31	7.38
210	7.45	7.52	7.59	7.66	7.73	7.81	7.88	7.95	8.03	8.10
220	8.17	8.25	8.32	8.40	8.47	8.55	8.63	8.70	8.78	8.86
230	8.93	9.01	9.09	9.17	9.25	9.33	9.41	9.49	9.57	9.65
240	9.73	9.81	9.89	9.97	10.05	10.14	10.22	10.30	10.39	10.47
250	10.55									

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Table T.3-3.1.2

VALUE OF THE COEFFICIENT ω IN TERMS OF THE SLENDERNESS RATIO λ FOR ROLLED SECTIONS IN Fe 510 STEEL

λ	0	1	2	3	4	5	6	7	8	9
20	1.06	1.06	1.07	1.07	1.08	1.08	1.09	1.09	1.10	1.10
30	1.11	1.12	1.12	1.13	1.14	1.15	1.15	1.16	1.17	1.18
40	1.19	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27
50	1.28	1.30	1.31	1.32	1.33	1.35	1.36	1.37	1.39	1.40
60	1.41	1.43	1.44	1.46	1.48	1.49	1.51	1.53	1.54	1.56
70	1.58	1.60	1.62	1.64	1.66	1.68	1.70	1.72	1.74	1.77
80	1.79	1.81	1.83	1.86	1.88	1.91	1.93	1.95	1.98	2.01
90	2.05	2.10	2.14	2.19	2.24	2.29	2.33	2.38	2.43	2.48
100	2.53	2.58	2.64	2.69	2.74	2.79	2.85	2.90	2.95	3.01
110	3.06	3.12	3.18	3.23	3.29	3.35	3.41	3.47	3.53	3.59
120	3.65	3.71	3.77	3.83	3.89	3.96	4.02	4.09	4.15	4.22
130	4.28	4.35	4.41	4.48	4.55	4.62	4.69	4.75	4.82	4.89
140	4.96	5.04	5.11	5.18	5.25	5.33	5.40	5.47	5.55	5.62
150	5.70	5.78	5.85	5.93	6.01	6.09	6.16	6.24	6.32	6.40
160	6.48	6.57	6.65	6.73	6.81	6.90	6.98	7.06	7.15	7.23
170	7.32	7.41	7.49	7.58	7.67	7.76	7.85	7.94	8.03	8.12
180	8.21	8.30	8.39	8.48	8.58	8.67	8.76	8.86	8.95	9.05
190	9.14	9.24	9.34	9.44	9.53	9.63	9.73	9.83	9.93	10.03
200	10.13	10.23	10.34	10.44	10.54	10.65	10.75	10.85	10.96	11.06
210	11.17	11.28	11.38	11.49	11.60	11.71	11.82	11.93	12.04	12.15
220	12.26	12.37	12.48	12.60	12.71	12.82	12.94	13.05	13.17	13.28
230	13.40	13.52	13.63	13.75	13.87	13.99	14.11	14.23	14.35	14.47
240	14.59	14.71	14.83	14.96	15.08	15.20	15.33	15.45	15.58	15.71
250	15.83									

Table T.3-3.1.3

VALUE OF THE COEFFICIENT ω IN TERMS OF THE SLENDERNESS RATIO λ
FOR TUBES IN Fe 360 STEEL

λ	0	1	2	3	4	5	6	7	8	9
20	1.00	1.00	1.00	1.00	1.01	1.01	1.01	1.02	1.02	1.02
30	1.03	1.03	1.04	1.04	1.04	1.05	1.05	1.05	1.06	1.06
40	1.07	1.07	1.08	1.08	1.09	1.09	1.10	1.10	1.11	1.11
50	1.12	1.13	1.13	1.14	1.15	1.15	1.16	1.17	1.17	1.18
60	1.19	1.20	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27
70	1.28	1.29	1.30	1.31	1.32	1.33	1.34	1.35	1.36	1.37
80	1.39	1.40	1.41	1.42	1.44	1.46	1.47	1.48	1.50	1.51
90	1.53	1.54	1.56	1.58	1.59	1.61	1.63	1.64	1.66	1.68
100	1.70	1.73	1.76	1.79	1.83	1.87	1.90	1.94	1.97	2.01
110	2.05	2.08	2.12	2.16	2.20	2.23				
for λ > 115, take the values of ω from table T.3-3.1.1										

Table T.3-3.1.4

VALUE OF THE COEFFICIENT ω IN TERMS OF SLENDERNESS RATIO λ
FOR TUBES IN Fe 510 STEEL

λ	0	1	2	3	4	5	6	7	8	9
20	1.02	1.02	1.02	1.03	1.03	1.03	1.04	1.04	1.05	1.05
30	1.05	1.06	1.06	1.07	1.07	1.08	1.08	1.09	1.10	1.10
40	1.11	1.11	1.12	1.13	1.13	1.14	1.15	1.16	1.16	1.17
50	1.18	1.19	1.20	1.21	1.22	1.23	1.24	1.25	1.26	1.27
60	1.28	1.30	1.31	1.32	1.33	1.35	1.36	1.38	1.39	1.41
70	1.42	1.44	1.46	1.47	1.49	1.51	1.53	1.55	1.57	1.59
80	1.62	1.66	1.71	1.75	1.79	1.83	1.88	1.92	1.97	2.01
90	2.05									
for λ > 90, take the values of ω from table T.3-3.1.2										

Note : The values of ω in tables T.3-3.1.3 and T.3-3.1.4 are valid for an axially loaded bar consisting of a single tube whose diameter is equal to at least six times its thickness.

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3-3.2 CHECKING ELEMENTS SUBJECT TO LATERAL BUCKLING

The manufacturer is free to choose the method for checking the strength of structural elements subject to lateral buckling. He must justify the origin of the method selected and apply it judiciously to load cases I, II and III.

The following list is not exhaustive, but, among the most commonly used methods, there are :

1 - in Germany, DIN 4114
2 - in Belgium the regulations NBN 1
3 - in France the rules CM 1966
4 - in United Kingdom, BS 2573

3-3.3 CHECKING PLATES SUBJECT TO BUCKLING

In determining the new 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 dependant on the ratio of stresses at the plate edges Ψ.

In addition it was found necessary to determine the critical buckling stress for circular cylinders and the spacing and moment of inertia of the transverse stiffeners in order to avoid too great divergences in the effective safety due to the used of highly divergent data in technical literature.

It shall be verified that the calculated stress is not higher than the critical buckling stress divided by the following coefficients ν_V :

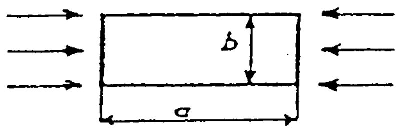
	load case	buckling safety ν_V
buckling of plane members	I II III	$1.70 + 0.175\ (\Psi - 1)$ $1.50 + 0.125\ (\Psi - 1)$ $1.35 + 0.075\ (\Psi - 1)$
buckling or curved members circular cylinders (e.g. : tubes)	I II III	1.70 1.50 1.35

The edge-stresses ratio Ψ varies between - 1 and + 1. For the definition of ν_V, Ψ corresponds to the plate to be stiffened.

From a theoretical standpoint, the critical buckling stress σ_{cr}^{v} is regarded as a multiple of the Euler Stress and is given by the formula :

$$\sigma_R^E = \frac{\pi^2 E}{12\,(1 - \eta^2)} \cdot \left(\frac{e}{b}\right)^2$$

representing the critical buckling stress for a strip of thickness e, having a width equal to b, this being the plate dimension measured in the direction perpendicular to the compression forces (see sketch below).



In this formula, E is the modulus of elasticity and η Poisson's Ratio.

For normal steels in which $E = 210{,}000$ N/mm² and $\eta = 0.3$, the Euler Stress becomes :

$$\sigma_R^E = 189{,}800 \cdot \left(\frac{e}{b}\right)^2$$

The critical buckling stress σ_{cr}^{v} must be a multiple of this value, whence, for compression :

$$\sigma_{cr}^{v} = K_\sigma\, \sigma_R^E$$

For shear the critical buckling stress is :

$$\tau_{cr}^{v} = K_\tau\, \sigma_R^E$$

The coefficients K_σ and K_τ known as the buckling coefficients, depend on :

- the ratio $\alpha = \frac{a}{b}$ of the two sides of the plate
- the manner in which the plate is supported along the edges
- the type of loading sustained by the plate in its own plane
- any reinforcement of the plate by stiffeners.

Values of coefficients K_σ and K_τ

Without wishing to enter into the details of this problem, which is the subject of specialized works and of particular standards, typical values of K_σ and K_τ are given hereafter (see table T.3-3.3.1).

For more complex cases, reference should be made to specialized literature.

Combined compression and shear

Taking σ and τ to be the calculated stresses in compression and in shear the critical comparison stress $\overset{v}{\sigma}_{cr.c}$ is determined from the expression :

$$\overset{v}{\sigma}_{cr.c} = \frac{\sqrt{\sigma^2 + 3\tau^2}}{\frac{1+\Psi}{4} \cdot \frac{\sigma}{\overset{v}{\sigma}_{cr}} + \sqrt{\left[\frac{3-\Psi}{4} \cdot \frac{\sigma}{\overset{v}{\sigma}_{cr}}\right]^2 + \left[\frac{\tau}{\overset{v}{\tau}_{cr}}\right]^2}}$$

Ψ being defined in the table T.3-3-3.1.

Important note : It is essential to note that the formulae above giving the critical stresses $\overset{v}{\sigma}_{cr}$ and $\overset{v}{\sigma}_{cr.c}$ apply only when the values determined thus are below the limit of proportionality (i.e. 190 N/mm² for Fe 360 steel, 290 N/mm² for Fe 510 steel).

Similary, the formula giving $\overset{v}{\tau}_{cr}$ applies only when the value $\sqrt{3}\ \overset{v}{\tau}_{cr}$ is below the limit of proportionality.

Whenever the formulae give values above these limits, it is necessary to adopt a limiting critical value, obtained by multiplying the calculated critical value by the coefficient ρ given in the table T.3-3.3.2, which also indicates the reduced values corresponding to various calculated values of $\overset{v}{\sigma}_{cr}$ and $\overset{v}{\tau}_{cr}$.

Table T.3-3.3.1

VALUES OF THE BUCKLING COEFFICIENTS K_σ AND K_τ
FOR PLATES SUPPORTED AT THEIR FOUR EDGES

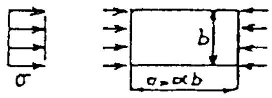
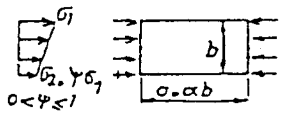
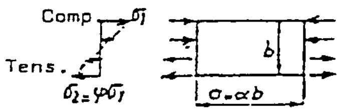
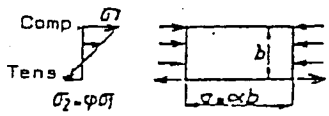
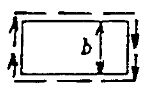
No.	case	$\alpha = \frac{a}{b}$	K_σ or K_τ
1	Simple uniform compression 	$\alpha \geq 1$ $\alpha \leq 1$	$K_\sigma = 4$ $K_\sigma = (\alpha + \frac{1}{\alpha})^2$
2	Non-uniform compression 	$\alpha \geq 1$ $\alpha \leq 1$	$K_\sigma = \frac{8.4}{\Psi + 1.1}$ $K_\sigma = (\alpha + \frac{1}{\alpha})^2 \cdot \frac{2.1}{\Psi + 1.1}$
3	Pure bending $\Psi = -1$ or bending with tension preponderant $\Psi < -1$ 	$\alpha \geq \frac{2}{3}$ $\alpha \leq \frac{2}{3}$	$K_\sigma = 23.9$ $K_\sigma = 15.87 + \frac{1.87}{\alpha^2} + 8.6\,\alpha^2$
4	Bending with compression preponderant $-1 < \Psi < 0$ 	$K\sigma = (1 + \Psi)\,K' - \Psi\,K'' + 10\,\Psi\,(1 + \Psi)$ where : K' = value of K_σ for $\Psi = 0$ in case no. 2 K" = value of K_σ for pure bending (case no. 3)	
5	Pure shear 	$\alpha \geq 1$ $\alpha \leq 1$	$K_\tau = 5.34 + \frac{4}{\alpha^2}$ $K_\tau = 4 + \frac{5.34}{\alpha^2}$

Table T.3-3.3.2

VALUES OF ρ AND THE REDUCED CRITICAL STRESSES

σ^{v}_{cr}, $\sigma^{v}_{cr.c}$ AND τ^{v}_{cr} (N/mm²)

σ^{v}_{cr} or $\sigma^{v}_{cr.c}$ calculated	τ^{v}_{cr} calculated	ρ	σ^{v}_{cr} or $\sigma^{v}_{cr.c}$ reduced	τ^{v}_{cr} reduced	σ^{v}_{cr} or $\sigma^{v}_{cr.c}$ calculated	τ^{v}_{cr} calculated	ρ	σ^{v}_{cr} or $\sigma^{v}_{cr.c}$ reduced	τ^{v}_{cr} reduced
steel Fe 360					steel Fe 510				
192.0	111.0	1.00	192.0	111.0					
200.0	116.0	0.99	198.3	114.5					
210.0	121.0	0.97	203.7	117.6					
220.0	127.0	0.94	207.7	119.9					
230.0	133.0	0.92	210.9	121.8					
240.0	139.0	0.89	213.6	123.3					
250.0	144.0	0.86	215.9	124.6					
260.0	150.0	0.83	217.8	125.7					
270.0	156.0	0.81	219.4	126.7					
280.0	162.0	0.79	220.9	127.5	280.0	162.0	1.00	280.0	161.7
290.0	167.0	0.77	222.2	128.3	290.0	167.0	1.00	289.9	167.4
300.0	173.0	0.74	223.3	128.9	300.0	173.0	0.99	297.4	171.7
320.0	185.0	0.70	225.2	130.0	320.0	185.0	0.96	307.7	177.7
340.0	196.0	0.67	226.7	130.9	340.0	196.0	0.93	314.9	181.8
360.0	208.0	0.63	228.0	131.6	360.0	208.0	0.89	320.4	185.0
380.0	219.0	0.60	229.1	132.3	380.0	219.0	0.85	324.8	187.5
400.0	231.0	0.58	230.0	132.8	400.0	231.0	0.82	328.4	189.6
420.0	243.0	0.55	230.8	133.3	420.0	243.0	0.79	331.3	191.3
440.0	245.0	0.53	231.5	133.7	440.0	254.0	0.76	333.8	192.7
460.0	266.0	0.50	232.1	134.0	460.0	266.0	0.73	335.9	193.9
480.0	277.0	0.48	232.6	134.3	480.0	277.0	0.70	337.8	195.0
500.0	289.0	0.47	233.1	134.6	500.0	289.0	0.68	339.4	196.0
550.0	318.0	0.43	234.0	135.1	550.0	318.0	0.62	342.6	197.8
600.0	346.0	0.39	234.8	135.6	600.0	346.0	0.58	345.0	199.2
650.0	375.0	0.36	235.3	135.9	650.0	375.0	0.53	346.9	200.3
700.0	404.0	0.34	235.8	136.1	700.0	404.0	0.50	348.4	201.1
800.0	462.0	0.30	236.5	136.5	800.0	462.0	0.44	350.6	202.4
1,000.0	577.0	0.24	237.4	137.1	1,000.0	577.0	0.35	353.3	204.0
2,000.0	1,155.0	0.12	238.9	137.9	2,000.0	1,155.0	0.18	357.4	206.3
10,000.0	5,774.0	0.02	239.8	138.5	10,000.0	5,774.0	0.04	359.6	207.6
∞	-	-	240.0	138.6	∞	-	-	360.0	207.8

.../

Determination of permissible buckling stresses

After the critical buckling stresses have been determined as indicated above, the permissible stress is obtained by dividing the critical stress by the coefficient ν_V (from clause 3-3.3).

The calculations are then performed as follows :

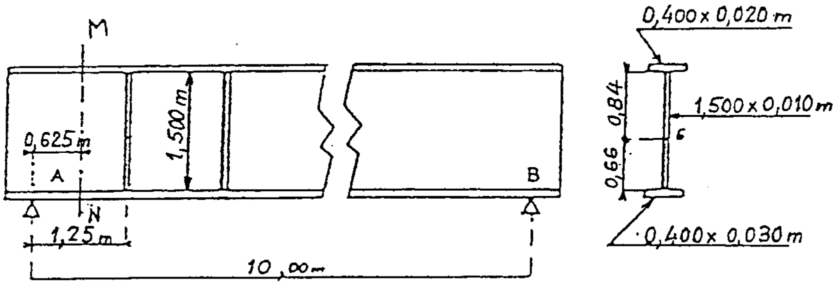
The maximum stresses are determined for each load case and a check is made to ensure that these calculated stresses do not exceed the permissible stresses determined as indicated above.

Note : In the case of combined compression and shear, the critical comparison stress $\sigma^V_{cr.c}$ must be compared with the equivalent stress calculated from the formula in clause 3-2.1.3 :

$$\sigma_{cp} = \sqrt{\sigma^2 + 3\tau^2} \leq \frac{\sigma^V_{cr.c}}{\nu_V}$$

Example of a check for buckling

Take the case of a plate girder in Fe 360 steel, having a span of 10 m, a depth of 1.50 m, a web thickness of 0.010 m, a uniformly distributed load of 162 kN/m and stiffeners 1.25 m apart.



Reactions on supports : A = B = 810 kN
Moment of inertia of the beam I = 1 419 000 cm⁴
Selected at section MN, located 0.625 m from A
Bending moment at MN :

$$M_f = 810 \ . \ 0.625 - \frac{162 \ . \ 0.625^2}{2} = 474.7 \text{ kN/m}$$

Upper stress (compression) :

$$\sigma_1 = \frac{474.7 \ . \ 10^6 \ . \ 0.84 \ . \ 10^3}{1\ 419\ 000 \ . \ 10^4} = 28 \text{ N/mm}^2$$

Lower stress (tension) :

$$\sigma_2 = \frac{474.7 \,.\, 10^6 \,.\, 0.66 \,.\, 10^3}{1\ 419\ 000 \,.\, 10^4} = 22 \text{ N/mm}^2$$

These stresses are calculated at the upper and lower edges of the web.

Shear stress :

$$\frac{810 \,.\, 10^3 - 162 \,.\, 0.625 \,.\, 10^3}{10 \,.\, 1{,}500} = 47 \text{ N/mm}^2$$

Bending (with compression preponderant = case no. 4 in table T.3-3.3.1.) :

$$\Psi = \frac{0.22}{-0.28} = -\ 0.79 \qquad \alpha = \frac{1.25}{1.50} = 0.83\ (< 1)$$

giving $K_\sigma = (1 + \Psi)K' - \Psi\, K'' + 10\, \Psi\, (1 + \Psi)$

in which $K' = \left(\alpha + \frac{1}{\alpha}\right)^2 . \frac{2.1}{0 + 1.1} = \left(0.83 + \frac{1}{0.83}\right)^2 . \frac{2.1}{1.1} = 7.90$

and $K'' = 23.9$

whence $K_\sigma = (1 - 0.79)\ 7.90 + 0.79 \,.\, 23.9 - 10 \,.\, 0.79\ (1 - 0.79) = 18.88$

The Euler Stress :

$$\sigma_R^E = 189\ 800 \left(\frac{e}{b}\right)^2 = 189\ 800 \left(\frac{10}{1{,}500}\right)^2 = 8.4 \text{ N/mm}^2$$

giving a critical buckling stress :

$$\sigma_{cr}^v = K_\sigma \,.\, \sigma_R^E = 18.89 \,.\, 8.4 = 158.6 \text{ N/mm}^2$$

Shear :

$$K_\tau = 4 + \frac{5.34}{\alpha^2} = 4 + \frac{5.34}{0.83^2} = 11.75$$

and

$$\tau^{v}_{cr} = K_\tau \, \sigma^{E}_{R} = 11.75 \, . \, 8.4 = 99 \text{ N/mm}^2$$

The critical comparison stress then becomes :

$$\sigma^{v}_{cr.c} = \frac{\sqrt{28^2 + 3 \, . \, 47^2}}{\frac{1 - 0.79}{4} \cdot \frac{28}{158.5} + \sqrt{\left[\frac{3 + 0.79}{4} \cdot \frac{28}{158.5}\right]^2 + \left(\frac{47}{99}\right)^2}} = 168 \text{ N/mm}^2$$

<u>Conclusion</u> :

The comparison stress in the case of tension (or compression) combined with shear is given in clause 3-2.1.3 :

$$\sqrt{\sigma^2 + 3\tau^2} = 86 \text{ N/mm}^2$$

This value is smaller than the admissible buckling stress (calculated with $\nu_V = 1.386$) i.e. $\frac{168}{1.386} = 121$ N/mm² for loading case I.

The permissible buckling stress is therefore not exceeded in load case I.

Naturally, a check must also be made to ensure that the permissible buckling stresses are not exceeded in load cases II and III.

<u>Checking for buckling in circular cylinders</u> :

Thin wall circular cylinders such as, for example, large tubes, which are subject to central of eccentric axial compression have to be checked for local buckling if :

$$\frac{t}{r} \leq \frac{25 \, . \, \sigma_E}{E}$$

where :

t = thickness of the wall

r = radius from the middle of the wall thickness

σ_E = elastic limit of the steel type, as in table T.3-2.1.1

E = modulus of elasticity see 3-2.1.1

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The "ideal" buckling stress σ_i^v can be determined from :

$$\sigma_i^v = 0.2 \cdot \frac{E \cdot t}{r}$$

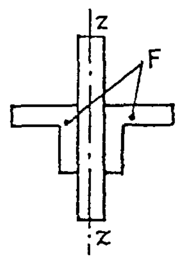
In all cases where σ_i^v is situated above the limit of proportionality of the structural steel, the "ideal" buckling stress σ_i^v has to be reduced to σ^v by means of the factor ρ.

At a maximum spacing of 10r, transverse stiffeners have to be provided whose moment of inertia has to be at least :

$$I = \frac{r \cdot t^3}{2} \sqrt{\frac{r}{t}}$$

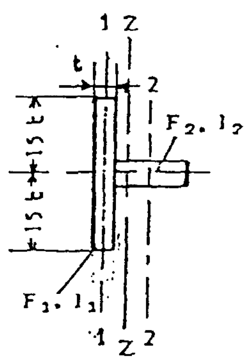
the moment of inertia is calculated from the following formulae :

1 - Central disposition of the stiffener(s) F (centre of gravity of the stiffener section in the median plane of the wall thickness)



$$I = I_Z$$

2 - Eccentric disposition of the stiffener F_2 (centre of gravity of the stiffener section F_2 outside the median plane of the wall F_1)



$$I = I_1 + I_2 + F_1 \cdot e_1^2 + F_2 \cdot e_2^2$$

It is accepted that in the calculation of σ_i^v and σ^v account is taken of geometrical divergence due to local construction defects between the real and the ideal cylinder surfaces up to a maximum of $\frac{t}{2}$.

3-4 CHECKING MEMBERS SUBJECTED TO FATIGUE

A risk of failure from fatigue occurs when a member is subjected to varying and/or repeated loads which may be considerably smaller than the static load capacity of the structure.

For structural members and joints the fatigue strength shall be checked for load case I forces (main loads) where these are expected to fluctuate significantly and where the total number of load cycles is expected to be more than $2.5 \,.\, 10^5$ in the anticipated lifetime of the machine.

All the static loads which may also occur in various magnitudes, for example incrustation, have to be calculated with that value which produces the highest tensile stress.

Fatigue strength is calculated by considering the following parameters :

1 - the scheduled number of cycles of load variations to which the member is subjected

2 - the material used and the notch category at the point being considered

3 - the extreme maximum stress σ_{max} which can occur in the member

4 - the ratio κ between the values of the extreme stresses.

With these parameters, it is possible for the given number of load cycles to determine a permissible stress below which no damage is expected of variations of loading.

3-4.1 PREDICTED NUMBER OF CYCLES AND STRESS SPECTRUM

The number of load cycles and the stress spectrum to be taken into consideration are discussed in clause 2-1.4.2 and in clause 2-1.4.3.

These two parameters are taken into account when considering the group in which the member is classified in accordance with clause 2-1.4.4.

This group classification is later used in the determination of the allowable stress for fatigue.

3-4.2 MATERIAL USED AND NOTCH EFFECT

The fatigue strength of a member depends upon the quality of the material used and upon the shape and the method of making the joints. The shapes of the parts joined and the means of doing it have the effect of producing stress concentrations (or notch effects) which considerably reduce the fatigue strength of the member.

In order to classify the importance of these notch effects, the various joint details are divided into categories as follows :

Unwelded parts :

These members present three classes of notch effect :

Case W_0 concerns the material itself without notch effect.

Cases W_1 and W_2 concern perforated members (see tables T.3-4.5.2.3).

Welded parts :

These joints are arranged in order of the severity of the notch effect increasing from K_0 to K_4, and relate to structural parts located close to the weld details.

The table T.3-2.2.1 gives some indications as to the quality of the welding required and the weld classification and tables T.3-4.5.2.3 give the various types of joints that are most often used in the construction of handling appliances.

3-4.3 DETERMINATION OF THE MAXIMUM STRESS σ_{max}

The maximum stress σ_{max} is the highest absolute stress (i.e. it may be tension or compression) which occurs in the member in load case I (see clause 2-3.1).

When checking members in compression for fatigue the crippling coefficient, ω, given in clause 3-3 should not be applied.

3-4.4 RATIO κ BETWEEN THE EXTREME STRESSES

This ratio is determined by calculating the extreme values of the stresses to which the component is subjected under case I loadings.

The ratio may vary depending upon the operating cycles but it is on the safe side to determined this ratio κ by taking the two extreme values which can occur during possible operations under case I loadings.

Where σ_{max} and σ_{min} are the values of these extreme stresses, σ_{max} being the extreme stress having the higher absolute value, the ratio κ may be written :

$$\kappa = \frac{\sigma_{min}}{\sigma_{max}} \quad \text{or, in the case of shear,} \quad \frac{\tau_{min}}{\tau_{max}}$$

This ratio, which varies from + 1 to - 1, is positive it the extreme stresses are both of the same sense (fluctuating stresses) and negative when the extreme stresses are of opposite sense (alternting stresses).

3-4.5 CALCULATING MEMBERS SUBJECT TO FATIGUE

Using the parameters defined in clauses 3-4.1 to 3-4.4 the adequacy of structural members and joints subject to fatigue is ensured by checking that the stress σ_{max}, as defined in clause 3-4.3, is not greater than the permissible stress for fatigue of the member under consideration.

This permissible stress for fatigue is derived from the critical stress, which, on the basis of tests made with test pieces, corresponds to a 90 % probability of survival. To this, a coefficient of safety of 1.33 is applied thus :

$$\sigma_a \text{ for fatigue} = \frac{\sigma(\text{at } 90\% \text{ survival})}{1.33} = 0.75.\sigma \text{ at } 90\% \text{ survival}$$

The determination of these permissible stresses is a complex problem and it is generally advisable to refer to specialized books on the subject.

Table T.3-4.5.1.1 gives basic values σ_w, based on the result of research in this field, for the determination of permissible stresses in Fe 360, Fe 430 and Fe 510 steels, according to the various groups in which the components are classified, and the notch effect classes for the main types of joints used in the manufacture of handling appliance.

3-4.5.1 FATIGUE CHECK FOR STRUCTURAL ELEMENTS (DETERMINATION OF THE PERMISSIBLE STRESSES FOR FATIGUE)

3-4.5.1.1 TENSILE AND COMPRESSIVE LOADS

The number of cycles and stress spectrum has been taken into account in determining the group classification for each member. A value of σ_w can therefore be selected from a knowledge of the component classification group and the material concerned.

For unwelded parts, the values σ_w are identical for steel Fe 360 and Fe 430. They are higher for Fe 510.

For welded parts, the values σ_w are identical for the three steel grades.

It is to be noted that, these basic values for permissible stresses are also applicable to weld seams (see 3-4.5.2.1).

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Table T.3-4.5.1.1
VALUES OF σ_W (N/mm²) DEPENDING ON THE COMPONENT GROUP AND CASE OF NOTCH EFFECT

component group	unwelded components - notch effect						welded components - notch effect (steels Fe 360 to Fe 510)				
	W_0		W_1		W_2						
	Fe 360 Fe 430	Fe 510	Fe 360 Fe 430	Fe 510	Fe 360 Fe 430	Fe 510	K_0	K_1	K_2	K_3	K_4
E1	249.1	298.0	211.7	253.3	174.4	208.6	361.9	323.1	271.4	193.9	116.3
E2	224.4	261.7	190.7	222.4	157.1	183.2	293.8	262.3	220.3	157.4	94.4
E3	202.2	229.8	171.8	195.3	141.5	160.8	238.4	212.9	178.8	127.7	76.6
E4	182.1	201.8	154.8	171.5	127.5	141.2	193.5	172.8	145.1	103.7	62.2
E5	***164.1***	***177.2***	***139.5***	***150.6***	***114.9***	***124.9***	***157.1***	***140.3***	***117.8***	***84.2***	***50.5***
E6	***147.8***	***155.6***	***125.7***	***132.3***	***103.5***	***108.9***	***127.5***	***113.8***	***95.6***	***68.3***	***41.0***
E7	***133.2***	***136.6***	***113.2***	***116.2***	***93.2***	***95.7***	***103.5***	***92.4***	***77.6***	***55.4***	***33.3***
E8	***120.0***	***120.0***	***102.0***	***102.0***	***84.0***	***84.0***	***84.0***	***75.0***	***63.0***	***45.0***	***27.0***

Note : All values which are greater than the admissible values with respect to the elastic limit are theoretical values only (see note at the end of clause 3-4.5.1.3).

For all values of κ, the following formulae give the permissible stresses for fatigue :

a) $\kappa \leq 0$

- for tension : $$\sigma_t = \sigma_w \frac{5}{3 - 2\kappa} \qquad (1)$$

- for compression : $$\sigma_c = \sigma_w \frac{2}{1 - \kappa} \qquad (2)$$

σ_w is given in table above.

b) $\kappa > 0$

- for tension : $$\sigma_t = \frac{\sigma_0}{1 - \left[1 - \frac{\sigma_0}{\sigma_{+1}}\right] \kappa} \qquad (3)$$

- for compression : $$\sigma_c = 1.2\ \sigma_t \qquad (4)$$

where σ_0 = tensile stress for $\kappa = 0$ is given by the formula (1) that is :

$$\sigma_0 = 1.66\ \sigma_w$$

σ_{+1} = tensile stress for $\kappa = +1$ that is the ultimate strength σ_R divided by the coefficient of safety 1.33 :

$$\sigma_{+1} = 0.75\ \sigma_R$$

σ_t is limited in every case to 0.66 σ_E.

3-4.5.1.2 SHEAR STRESSES IN THE MATERIAL OF STRUCTURAL PARTS

For each of the group from E1 to E8 the permissible fatigue stress in tension of the case W_0 divided by $\sqrt{3}$ is taken :

$$\tau_a = \frac{\sigma_t \text{ of case } W_0}{\sqrt{3}}$$

3-4.5.1.3 COMBINED LOADS IN TENSION (OR COMPRESSION) AND SHEAR

In this case the permissible stresses for fatigue for each normal load in tension (or compression) σ_{xa} and σ_{xa} and shear τ_{xya} are determined by assuming that each acts separately taking respectively fhe following values of κ in accordance with clause 3-4.4 :

$$\kappa_x = \frac{\sigma_{x\,min}}{\sigma_{x\,max}} \quad ; \quad \kappa_y = \frac{\sigma_{y\,min}}{\sigma_{y\,max}} \quad \text{and} \quad \kappa_{xy} = \frac{\tau_{xy\,min}}{\tau_{xy\,max}}$$

Then the following three conditions are checked :

$$\sigma_{x\,max} \le \sigma_{xa} \quad ; \quad \sigma_{y\,max} \le \sigma_{ya} \quad \text{and} \quad \tau_{xy\,max} \le \tau_{xya}$$

None of the calculated stresses should exceed the permissible value of σ_a (or τ_a) in case I loading (see table T.3-2.1.1).

Then for the verification under the effect of a combination of these three types of loads, the following two cases are considered :

a) If any one stress is markedly greater than the other two for a given load case, it is only necessary to check the member for fatigue under that load, neglecting the effect of the other two.

b) In the other cases, in addition to checking for each loading assumed to act alone, it is recommended that the following relationship be checked :

.../

$$\left(\frac{\sigma_{x\,max}}{\sigma_{xa}}\right)^2 + \left(\frac{\sigma_{y\,max}}{\sigma_{ya}}\right)^2 - \frac{\sigma_{x\,max} \cdot \sigma_{y\,max}}{|\sigma_{xa}| \cdot |\sigma_{ya}|} + \left(\frac{\tau_{xy\,max}}{\tau_{xya}}\right)^2 \leq 1 \qquad *(1) \qquad (5)$$

where the stress values σ_{xa}, σ_{ya} and τ_{xya} are those resulting from the application of formulae (1), (2), (3) and (4) of clause 3-4.5.1.1 and limited to 0.66 σ_E.

In applying this formula, reference should be made to the indications given in clause 3-2.1.3 (including the note at the end).

In order to facilitate the calculations, table T.3-4.5.1.3 gives the permissible values of :

$\frac{\tau_{xy\,max}}{\tau_{xya}}$ as a function of $\frac{\sigma_{x\,max}}{\sigma_{xa}}$ and of $\frac{\sigma_{y\,max}}{\sigma_{ya}}$

In this table, the values of $\frac{\sigma_{x\,max}}{\sigma_{xa}}$ are given in the left hand column with the following convention : the ratio is considered to be positive if $\sigma_{x\,max}$ and $\sigma_{y\,max}$ have the same sign, and negative otherwise.

---

*(1) As this inequality constitutes a severe requirement, values slightly higher that 1 are acceptable, but in this case it is necessary to check the relation :

$$\sqrt{\left(\frac{\sigma_{x\,max}}{\sigma_{xa}}\right)^2 + \left(\frac{\sigma_{y\,max}}{\sigma_{ya}}\right)^2 - \frac{\sigma_{x\,max} \cdot \sigma_{y\,max}}{|\sigma_{xa}| \cdot |\sigma_{ya}|} + \left(\frac{\tau_{xy\,max}}{\tau_{xya}}\right)^2} \leq 1.05$$

It should also be noted that the values $|\sigma_{xa}|$ and $|\sigma_{ya}|$ in the denominator for the third term should be taken as absolute values, $\sigma_{x\,max}$ and $\sigma_{y\,max}$ being asigned their algebraic values.

Table T.3-4.5.1.3

VALUES OF $\frac{\tau_{xy\ max}}{\tau_{xya}}$ IN TERMS OF $\frac{\sigma_{x\ max}}{\sigma_{xa}}$ AND $\frac{\sigma_{y\ max}}{\sigma_{ya}}$

$\frac{\sigma_{x\ max}}{\sigma_{xa}}$	$\frac{\sigma_{y\ max}}{\sigma_{ya}}$										
	1.0	0.9	0.8	0.7	0.6	0.5	0.4	0.3	0.2	0.1	0
+ 1.0	0	0.300	0.400	0.458	0.490	0.500	0.490	0.458	0.400	0.300	0
+ 0.9	0.300	0.436	0.520	0.575	0.608	0.625	0.625	0.608	0.575	0.520	0.436
+ 0.8	0.400	0.520	0.600	0.656	0.693	0.714	0.721	0.714	0.693	0.656	0.600
+ 0.7	0.458	0.575	0.656	0.714	0.755	0.781	0.794	0.781	0.781	0.755	0.714
+ 0.6	0.490	0.608	0.693	0.755	0.800	0.831	0.849	0.854	0.849	0.831	0.800
+ 0.5	0.500	0.625	0.714	0.781	0.831	0.866	0.889	0.900	0.900	0.889	0.866
+ 0.4	0.490	0.625	0.721	0.794	0.849	0.889	0.917	0.933	0.938	0.933	0.917
+ 0.3	0.458	0.608	0.714	0.794	0.854	0.900	0.933	0.954	0.964	0.964	0.954
+ 0.2	0.400	0.575	0.693	0.781	0.849	0.900	0.938	0.964	0.980	0.985	0.980
+ 0.1	0.300	0.520	0.656	0.755	0.831	0.889	0.933	0.964	0.985	0.995	0.995
0	0	0.436	0.600	0.714	0.800	0.866	0.916	0.954	0.980	0.995	1.000
- 0.1		0.300	0.520	0.656	0.755	0.831	0.889	0.933	0.964	0.985	0.995
- 0.2			0.400	0.575	0.693	0.781	0.849	0.900	0.938	0.964	0.980
- 0.3			0.173	0.458	0.608	0.714	0.794	0.854	0.900	0.933	0.954
- 0.4				0.265	0.490	0.625	0.721	0.781	0.849	0.889	0.917
- 0.5					0.300	0.500	0.625	0.714	0.781	0.831	0.866
- 0.6						0.300	0.490	0.608	0.693	0.755	0.800
- 0.7							0.265	0.458	0.575	0.656	0.714
- 0.8								0.173	0.400	0.520	0.600
- 0.9										0.300	0.436
- 1.0											0

If $\sigma_{x\ max}$ and $\sigma_{y\ max}$ are of opposite sign (tension or compression) read the values of $\frac{\tau_{xy\ max}}{\tau_{xya}}$ starting from the negative values of $\frac{\sigma_{x\ max}}{\sigma_{xa}}$.

Note : From table T.3-4.5.1.1 it can be seen that in groups E1 and E2 much higher stresses than those usually permitted in structures are quoted. These values are theoretical values obtained by extrapolation of the test results on higher groups (E3 to E8) with medium or severe notch cases (K_2, K_3 and K_4). Therefore there is no need to attach any material significance to these values in brackets, consideration of which could in some cases lead to the conclusion that an assembly of type K_0 or K_1 could resist fatigue better than the unwelded metal (case W_0). This apparent anomaly illustrates the well known fact that it is not always necessary to carry out fatigue checks for the lower groups with slight or moderate notch effect classes.

For the purpose of calculation it must be remembered that these theoretical σ_w values are used only to determine the permissible fatigue stresses σ_{xa}, σ_{ya} and τ_{xya} for use in the formula which covers the case of combined loads.

.../

3-4.5.2 JOINTS (WELDS AND BOLTS)

3-4.5.2.1 WELDS

a) Tensile and compressive loads in welds

Welds subject to fatigue under tensile and compressive loads are checked using the same permissible stresses as those of the metal joined. Tables T.3-4.5.3.1 to T.3-4.5.3.4 give the permissible stress according to κ and the group of the component.

Note : The limits indicated under 3-2.2.2 for certain particular cases of transverse tension and compression in weld seams must be observed.

Chapter 3-2.2.3 gives, in addition, some indications for the determination of the stresses in the weld seams.

b) Shear loads in the welds

The permissible shear fatigue stresses in the welds are determined by dividing the permissible stresses in tension for case K_0 by $\sqrt{2}$. Tables T.3-4.5.3.5 give the permissible stress according to κ and the group of the component.

c) Combined loads

The method set out above for structural members is used when considering the effect of fatigue in weld seams subjected to variable combined loads.

3-4.5.2.2 BOLTED JOINTS

A) PRECISION BOLTS

a) Shear loads and bearing pressure

Single and double shear loads as defined under 3-2.3.3.2 must be treated separately :

The permissible shear stresses for fatigue in bolts are fixed by multiplying the permissible stresses intension for case W_2 by :

0.6 for single shear
0.8 for double shear

The permissible bearing pressure values are obtained by multiplying the permissible shear values in the bolts by 2.5.

The permissible fatigue stresses are given directly, according to factor κ (ratio between extreme stresses) on the following tables :

.../

- single and double shear in fitted bolts, grade 4.6 and 5.6 for groups E5 to E8, table T.3-4.5.3.6
- bearing stresses for fitted bolts, grade 4.6 and 5.6 for single and double shear and for groups E5 to E8, table T.3-4.5.3.7.

b) Bolts with controlled tightening

Under the effect of the service load F_1, the true tensile stress varies between the values :

$$\sigma_p \text{ and } \sigma_p + \frac{F_1 . \delta_b}{S_b}$$

where :

σ_p = theoretical tensile stress under the tightening effect

S_b = section of the root (< section of the shank)

$$\delta_b = \frac{\Delta l_1}{\Delta l_1 + \Delta l_2} \text{ (see 3-2.3.3.1)}$$

The following equation must be verified :

$$\sigma_1 = \frac{F_1}{S_b} \leq \frac{2\,\sigma_A}{\delta_b}$$

σ_a is the amplitude of the maximum permissible stress for fatigue given in the following graph 3-4.5.2.2.

For any other type of bolt or design method the σ_a value should ensure at least an equivalent level of safety against fatigue.

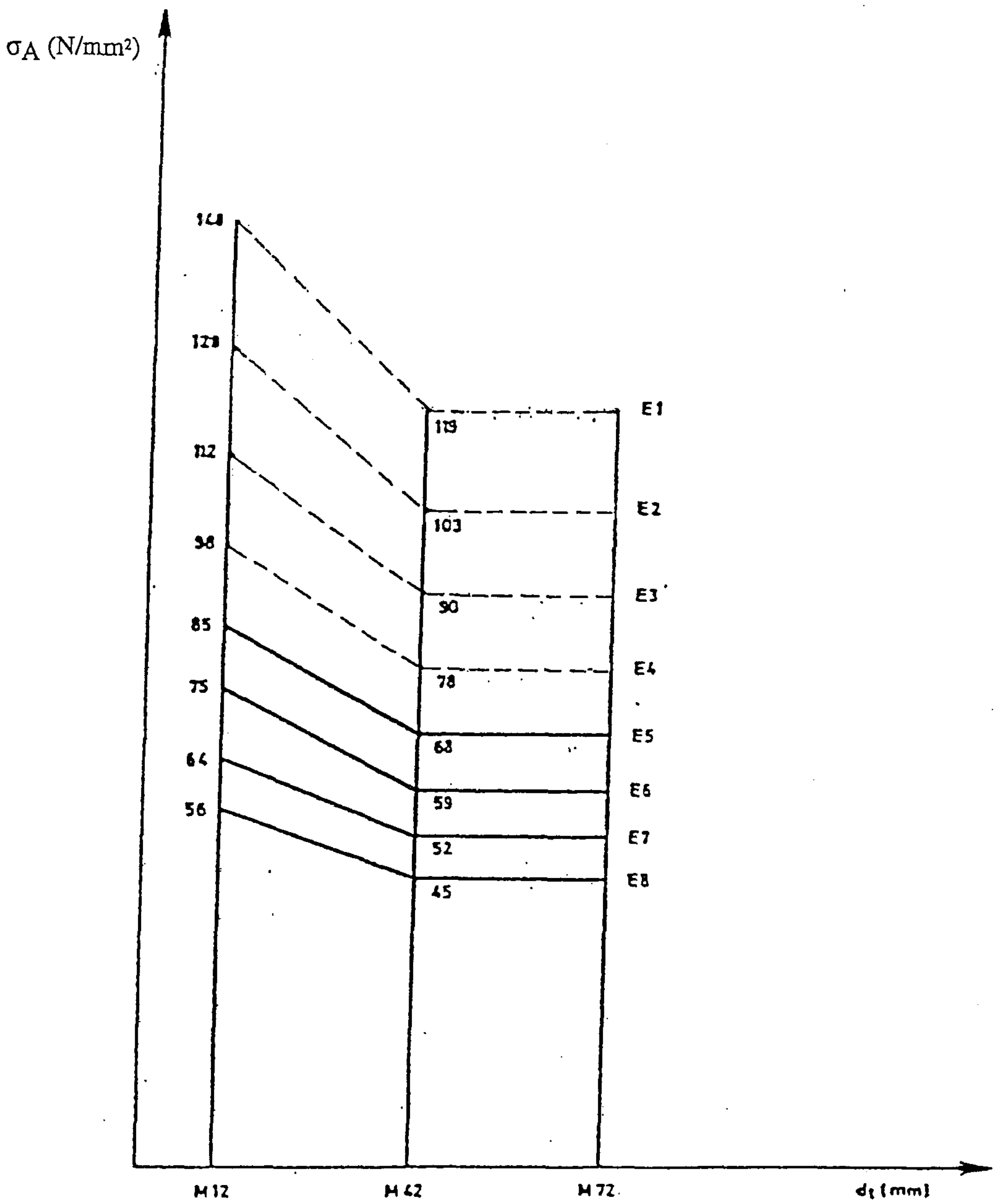
Any conformity tests should be carried out according to ISO specification ISO 3800/1 (Threaded fasteners - Axial load fatigue testing - Part 1 : Test methods) with $\sigma_m = 0.8\ R_E$ ($R_E = \sigma_E$).

B) Bolts in friction grip joints with controlled tightening

High tensile steel bolts with controlled tightening as defined in 3-2.3.4 have not to be verified for resistance to fatigue.

.../

Graph 3-4.5.2.2



<u>Amplitude of maximum permissible fatigue stress</u> σ_A

Graph for ISO bolts

- standard thread
- classes 8.8, 10.9, 12.9
- cold rolled thread with heat treatment after rolling.

Tables T.3-4.5.2.3
CASES OF NOTCH EFFECT

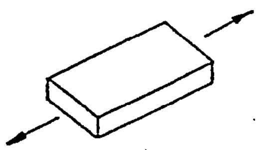
In the tables below the various joint details are classified in terms of the magnitude of the notch effect they produce.

It should be noted that, for any given weld, the notch effect differs according to the type of loading to which the joint is subjected.

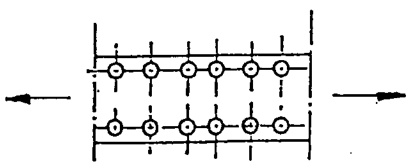
For example, a fillet welded joint is classified under case K_0 for longitudinal tension or compression loads (ref. 0.31) or longitudinal shear (ref. 0.51), and under cases K_3 or K_4 for transverse tension or compression loads (ref. 3.2 or 4.4).

1 - NON WELDED PARTS

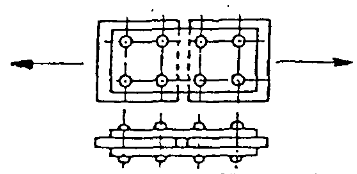
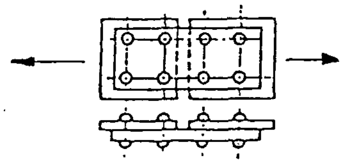
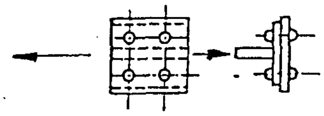
Case W_0

Reference	Description	Figure	Symbol
W_0	Parent metal, homogeneous surface. Part without joints or breaks in continuity (solid bars) and without notch effects unless the latter can be defined by calculation		

Case W_1

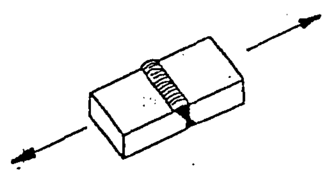
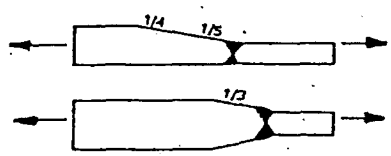
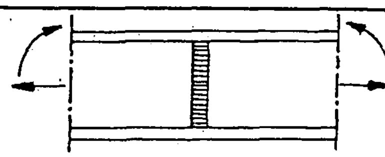
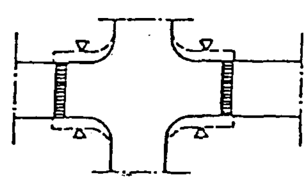
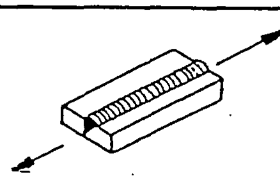
Reference	Description	Figure	Symbol
W_1	Part drilled for riveting or bolting with rivets and bolts loaded up to 20 % of permissible values. Parts drilled for joints using high strength bolts (Cl. 3-2.3.4) loaded up to 100 % or permissible values (Cl. 3-2.3.3)		

Case W_2

Reference	Description	Figure	Symbol
$W_{2.1}$	Parts drilled for riveting or bolting in which the rivets or bolts are loaded in double shear		
$W_{2.2}$	Parts drilled for riveting or bolting, in which the rivets or bolts are loaded in single shear (allowing for eccentric loads), the parts being unsupported		
$W_{2.3}$	Parts drilled for assembly by means of rivets and bolts loaded in single shear, the parts being supported or guided		

2 - WELDED PARTS

Case K_0 - Slight stress concentration

Reference	Description	Figure	Symbol see page 3-14
0.1	Part of equal thickness, butt-welded (special quality) at right angles to direction of forces		P 100
0.11	Parts of different thickness butt-welded (special quality) at right angles to direction of forces Asymmetrical slope : 1/4 to 1/5 Symmetrical slope : 1/3		P 100
0.12	Butt-weld (special quality) forming transverse joint in web plate		P 100
0.13	Gusset secured by butt-welding (special quality) at right angles to the direction of the forces		P 100
0.3	Parts joined by butt-welding parallel to the direction of the forces		P 100/P

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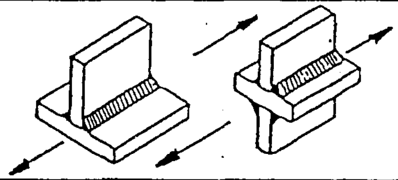
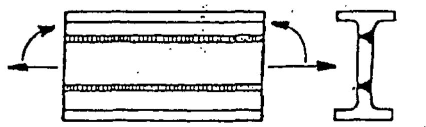
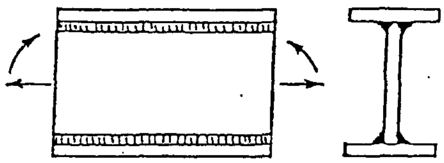
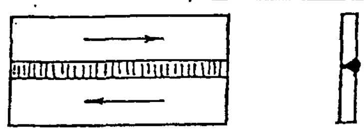
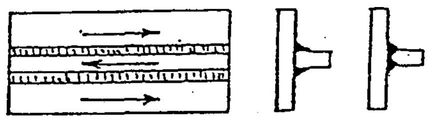
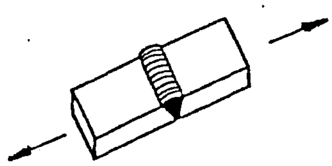
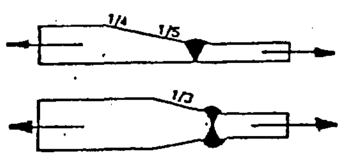
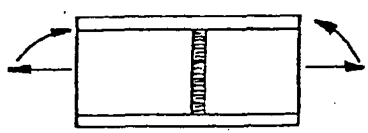
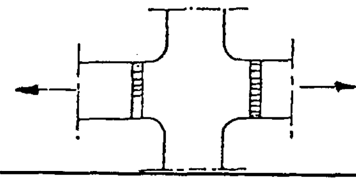
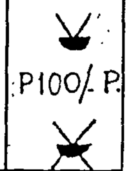
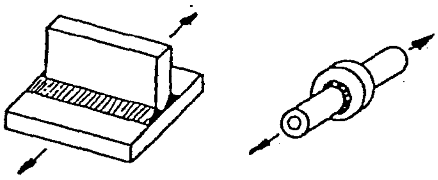
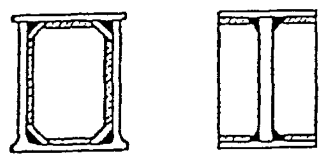
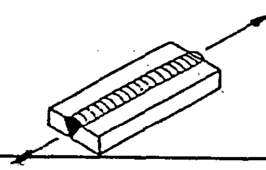
Case K_0 - Slight stress concentration (continued)

Reference	Description	Figure	Symbol see page 3-14
0.31	Parts joined by filled welds parallel to the direction of the forces		
0.32	Butt weld between sections forming flange and web of a beam (combined bending and tension)		P100/-P
0.33	K- or fillet weld between flange and web of a beam (combined bending and tension)		K
0.5	Butt-weld in longitudinal shear		P100/-P
0.51	K-welded or fillet weld in longitudinal shear		K

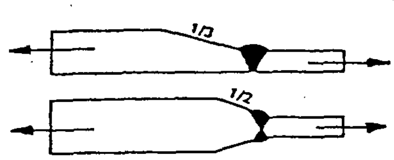
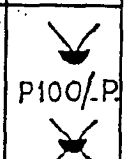
Case K_1 - Moderate stress concentration

Reference	Description	Figure	Symbol see page 3-14
1.1	Parts joined by butt-welding at right angles to the direction of the forces		P100/-P
1.11	Parts of different thickness butt-welded at right angles to the direction of the forces : Asymmetrical slope : 1/4 to 1/5 (or symmetrical slopes : 1/3)		P100/-P
1.12	Butt-weld executed for transverse joint of web plate		P100/-P

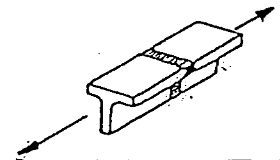
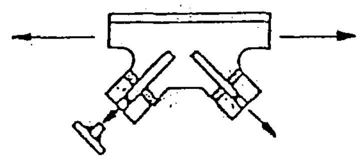
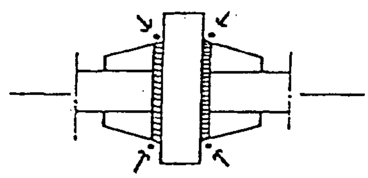
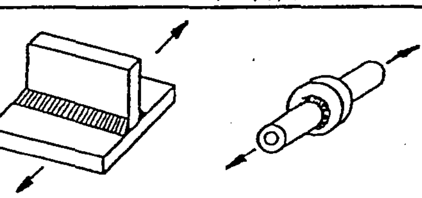
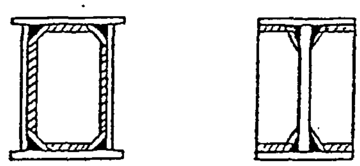
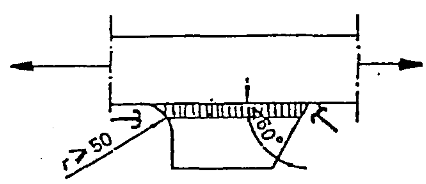
Case K_1 - Moderate stress concentration (continued)

Reference	Description	Figure	Symbol see page 3-14
1.13	Gusset joined by butt-welding at right angles to the direction of the forces		
1.2	Continuous main member to which parts are joined at right angles to the direction of forces by continuous K-welds (special quality)		
1.21	Compressed flanges and webs fixed by fillet weld (special quality) to transverse web or stiffeners with corners cut-off (classification only applies to fillet weld area), the welds extending round the corners of the web stiffeners		
1.3	Parts joined by butt-welding parallel to the direction of the forces (without checking the weld)		
1.31	K-weld (special quality) between curbed flange and web		

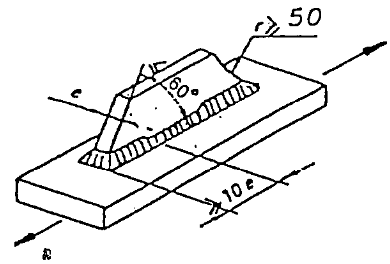
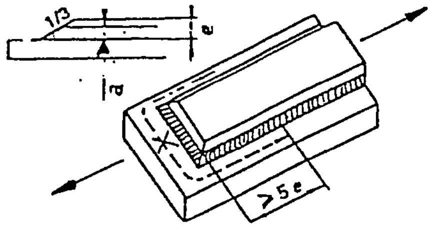
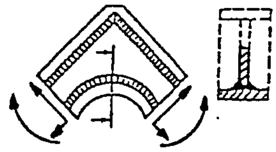
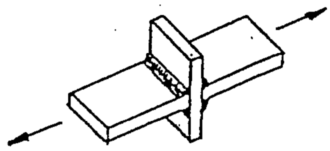
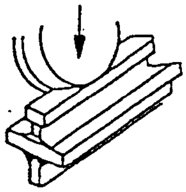
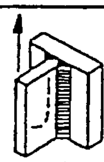
Case K_2 - Medium stress concentration

Reference	Description	Figure	Symbol see page 3-14
2.1	Parts of different thickness butt-welded at right angles to the direction of the forces. Asymmetrical slope : 1/3 Symmetrical slopes : 1/2		

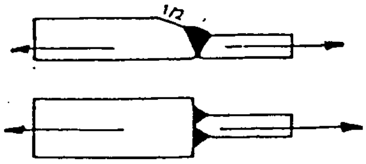
Case K_2 - Medium stress concentration (continued)

Reference	Description	Figure	Symbol see page 3-14
2.11	Sections joined by butt-welds (special quality) at right angles to the direction of the forces		P100/-P
2.12	Section joined to a gusset by a butt-weld (special quality) at right angles to the direction of the forces		P 100
2.13	Butt-weld (special quality) at right angles to the direction of the forces, made at intersection of flats, with welded auxiliary gussets. The ends of the welds are ground, avoiding notches		P 100
2.2	Continuous main member to which transverse diaphragms, web stiffeners, rings or hubs are fillet welded (special quality) at right angles to the direction of the forces		
2.21	Web in which fillet welds (special quality) are used to secure transverse web stiffeners with cut corners, the welds not extending round the corners		
2.3	Continuous main member to the edges of which are butt-welded (special quality) parts parallel to the direction of the forces. These parts terminate in vebels or radii. The ends of the welds are ground avoiding notches		

Case K_2 - Medium stress concentration (continued)

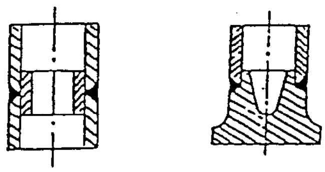
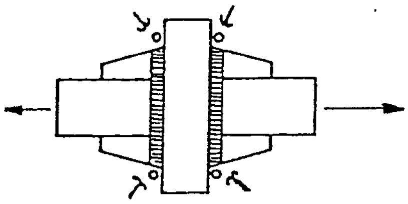
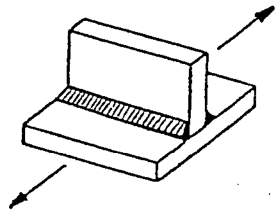
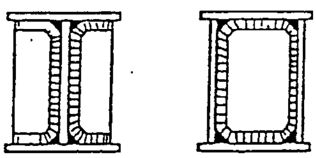
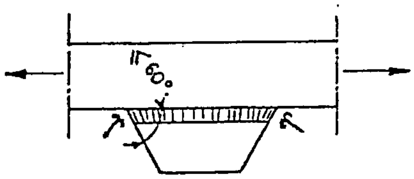
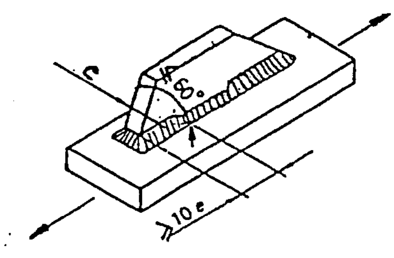
Reference	Description	Figure	Symbol see page 3-14
2.31	Continuous main member to which are welded parts parallel to the direction of the forces. These parts terminate in bevels or radii. Valid where the ends of the welds are K-welds (special quality) over a length equal to ten times the thickness provided that the ends of the welds are ground avoiding notches		
2.33	Continuous member to which a flat (1/3 bevel) is joined by a fillet weld the fillet weld (special quality) being executed in the X area, with : a = 0.5 e		
2.34	K-weld made between curved flange and web		K
2.4	Cruciform joint made with K-welds (special quality) perpendicular to the direction of the forces		D
2.41	K-weld (special quality) between flange and web in the case of load concentrated in the plane of the web at right angles to the weld		
2.5	K-weld (special quality) joining parts stressed in bending or shear		D

Case K_3 - Severe stress concentration

Reference	Description	Figure	Symbol see page 3-14
3.1	Parts of different thickness connected by butt-welds at right angles to the direction of the forces. 1/2 asymmetrical slope, or symmetrical position without blend slope		P100/-P

.../

Case K3 - Severe stress concentration (continued)

Reference	Description	Figure	Symbol see page 3-14
3.11	Butt-weld with backing strip and no backing run. (Backing strip secured by intermittent tack welds inside the bevel)		
3.12	Tubes joined by butt-welds whose root is supported by a backing piece and not covered by a backing run		
3.13	Butt-weld at right angles to the direction of the forces at the intersection of flats with welded auxiliary gussets. The ends of the welds are ground, avoiding notches		 P100/P 
3.2	Continuous main member to which parts are fillet welded at right angles to the direction of the forces. These parts take only a small portion of the loads transmitted by the main member		
3.21	Web and stiffener or transverse diaphragm secured by uninterrupted double fillet weld		
3.3	Continuous member to the edges of which are butt-welded parts parallel to the direction of the forces. These parts terminate in bevels and ends of the welds are ground avoiding notches		
3.31	Continuous member to which are welded parts parallel to the direction of the forces. These parts terminate in bevels or radii. Valid where the ends of the welds are fillet welds (special quality) over a length equal to 10 times the thickness, provided that the ends of the welds are ground, avoiding notches		

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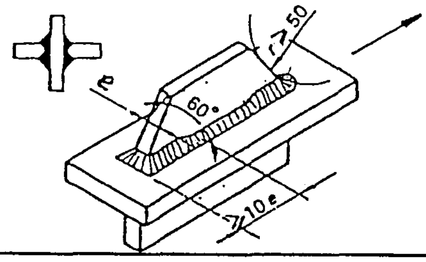
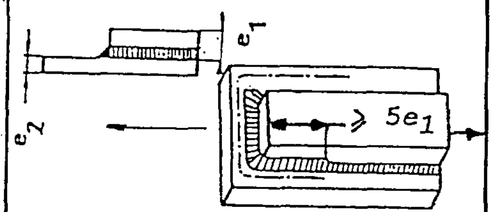
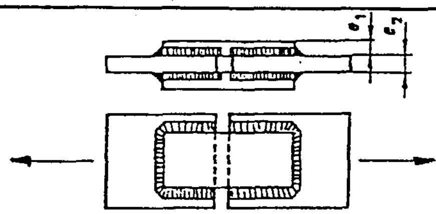
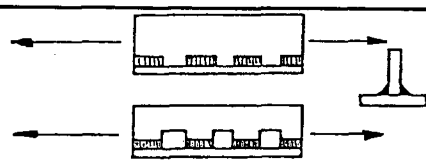
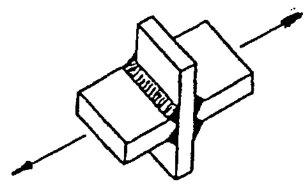
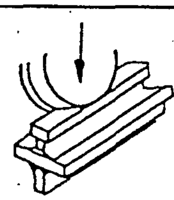
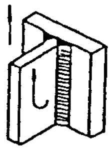
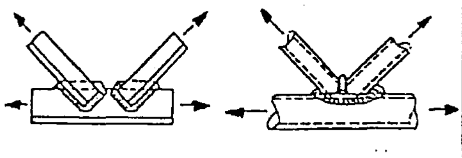
Case K_3 - Severe stress concentration (continued)

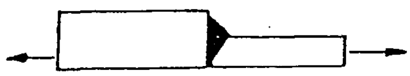
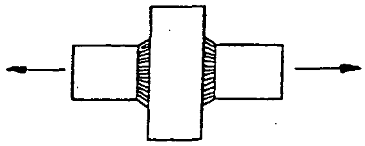
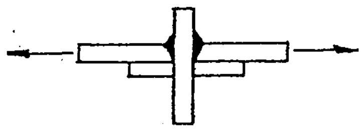
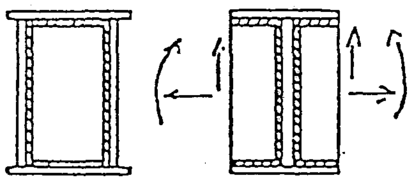
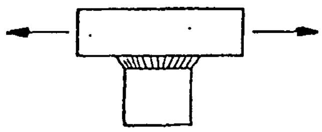
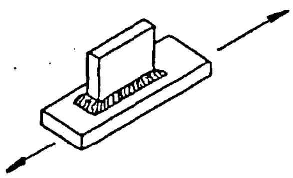
Reference	Description	Figure	Symbol see page 3-14
3.32	Continous member through which a plate extends, terminating in bevels or radii parallel to the direction of the forces, secured by K-weld over a length equal to 10 times the thickness		
3.33	Continuous member to which a flat is welded parallel to the direction of the forces, by means of fillet weld (special quality) in the indicated area when : $e_1 \leq 1.5\ e_2$		
3.34	Members at the extremity of which connecting gussets are secured by a fillet weld (special quality) $e_1 \leq e_2$. In the case of a single gusset, allow for eccentric load		
3.36	Continuous member to which stiffeners are secured parallel to the direction of the forces by fillet welds which are intermittent		
3.4	Cruciform joint made with K-weld at right angles to the direction of the forces		D K
3.41	K-weld between flange and web with a concentrated load in the plane of the web at right angles to the weld		K

.../

Case K3 - Severe stress concentration (continued)

Reference	Description	Figure	Symbol see page 3-14
3.5	K-weld joining parts stressed in bending and shear		D K
3.7	Continuous member to which sections or tubes are fillet welded (special quality)		

Case K3 - Very severe stress concentration

Reference	Description	Figure	Symbol see page 3-14
4.1	Parts of different thickness butt-welded at right angles to the direction of the forces. Asymmetrical position without blend slope		P-
4.11	Butt-welds at right angles to the direction of the forces, at the intersection of flats (no auxiliary gussets)		P- X
4.12	Single bevel weld at right angles to the direction of the forces, between intersecting parts (cruciform joint)		D M
4.21	Flanges and webs fixed by one-side continuous fillet weld to the traverse web, perpendicular to the stress direction		
4.3	Continuous member to the sides of which are welded parts ending at right angles, parallel to the direction of the forces		
4.31	Continuous member to which parts are filled welded ending at right angles, parallel to the direction of the forces, and receiving a large proportion of the loads transmitted by the main member		

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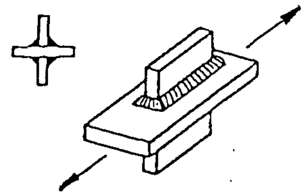
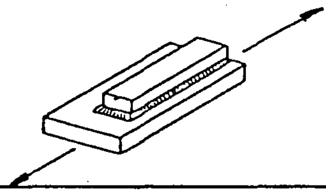
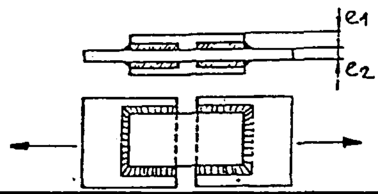
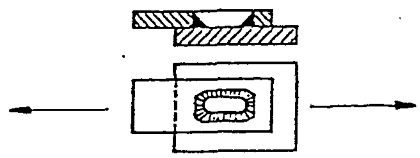
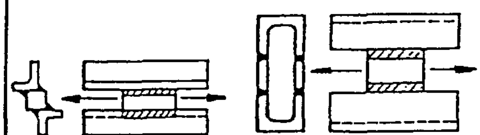
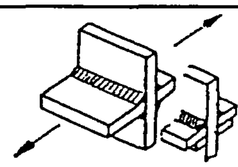
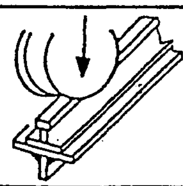
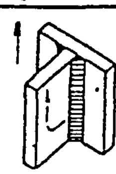
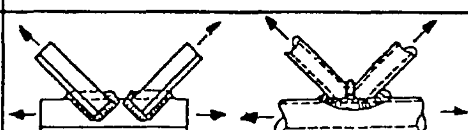
Case K_3 - Very severe stress concentration (continued)

Reference	Description	Figure	Symbol ISO 2553
4.32	Continuous member through which extends a plate ending at right angles and secured by fillet welding		
4.33	Continuous member on which a flat is secured by means of a fillet weld parallel to the direction of the forces		
4.34	Joint plate secured by fillet welds ($e_1 \leq e_2$). In the case of a single joint plate, allow for eccentric loads		
4.35	Parts welded one on the other secured by fillet welds in a slot or in holes		
4.36	Continuous members between which connecting gussets are secured by fillet welds or butt-welds		
4.4	Cruciform joint made with fillet weld at right angles to the direction of the forces		D
4.41	Fillet weld between flange and web with a case of concentrated load in the plane of the web at right angles to the weld		
4.5	Fillet welds joining parts stressed in bending and shear		D
4.7	Continuous member to which sections or tubes are connected by fillet welds		

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3-4.5.3 PERMISSIBLE FATIGUE STRESSES

The permissible fatigue stresses are given directly according to the ratio between extreme stresses κ on the following tables :

- Tension and compression in the material and in the weld seams :

Components of group E5	Table T.3-4.5.3.1
Components of group E6	Table T.3-4.5.3.2
Components of group E7	Table T.3-4.5.3.3
Components of group E8	Table T.3-4.5.3.4

- Shear in parent metal and weld seams :

Components of groups E5 to E8	Table T.3-4.5.3.5

- Single and double shear in fitted bolts :

Components of groups E5 to E8	Table T.3-4.5.3.6

- Bearing stresses for fitted bolts for single and double shear :

Components of groups E5 to E8	Table T.3-4.5.3.7

.../

Table T.3-4.5.3.1

PERMISSIBLE FATIGUE STRESSES

Tension and compression in the material and in the weld seams
for construction cases W_0 to W_2 and K_0 to K_4
and for steels Fe 360, Fe 430 and Fe 510

COMPONENTS OF GROUP E5

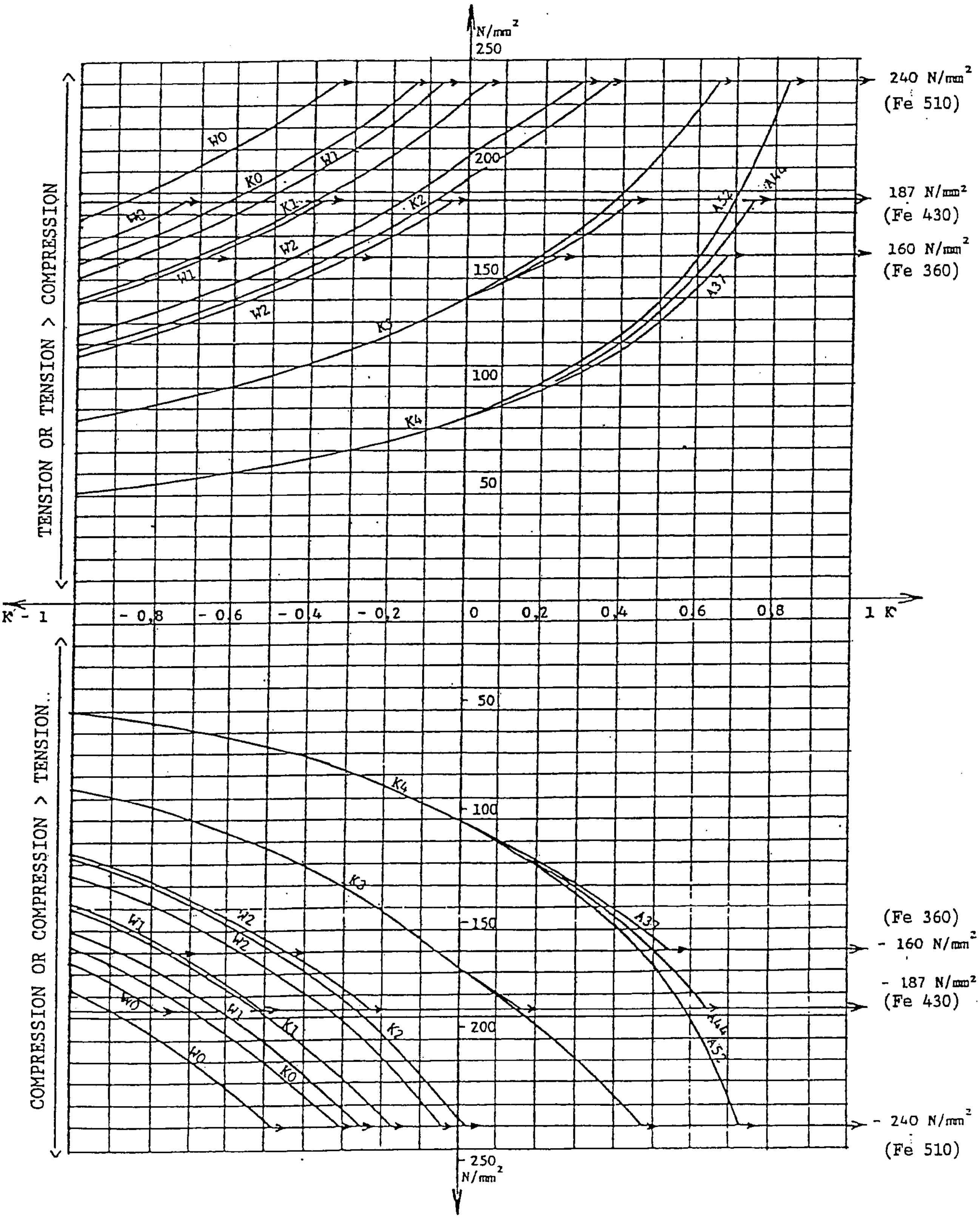


Table T.3-4.5.3.2

PERMISSIBLE FATIGUE STRESSES

Tension and compression in the material and in the weld seams
for construction cases W_0 to W_2 and K_0 to K_4
and for steels Fe 360, Fe 430 and Fe 510

COMPONENTS OF GROUP E6

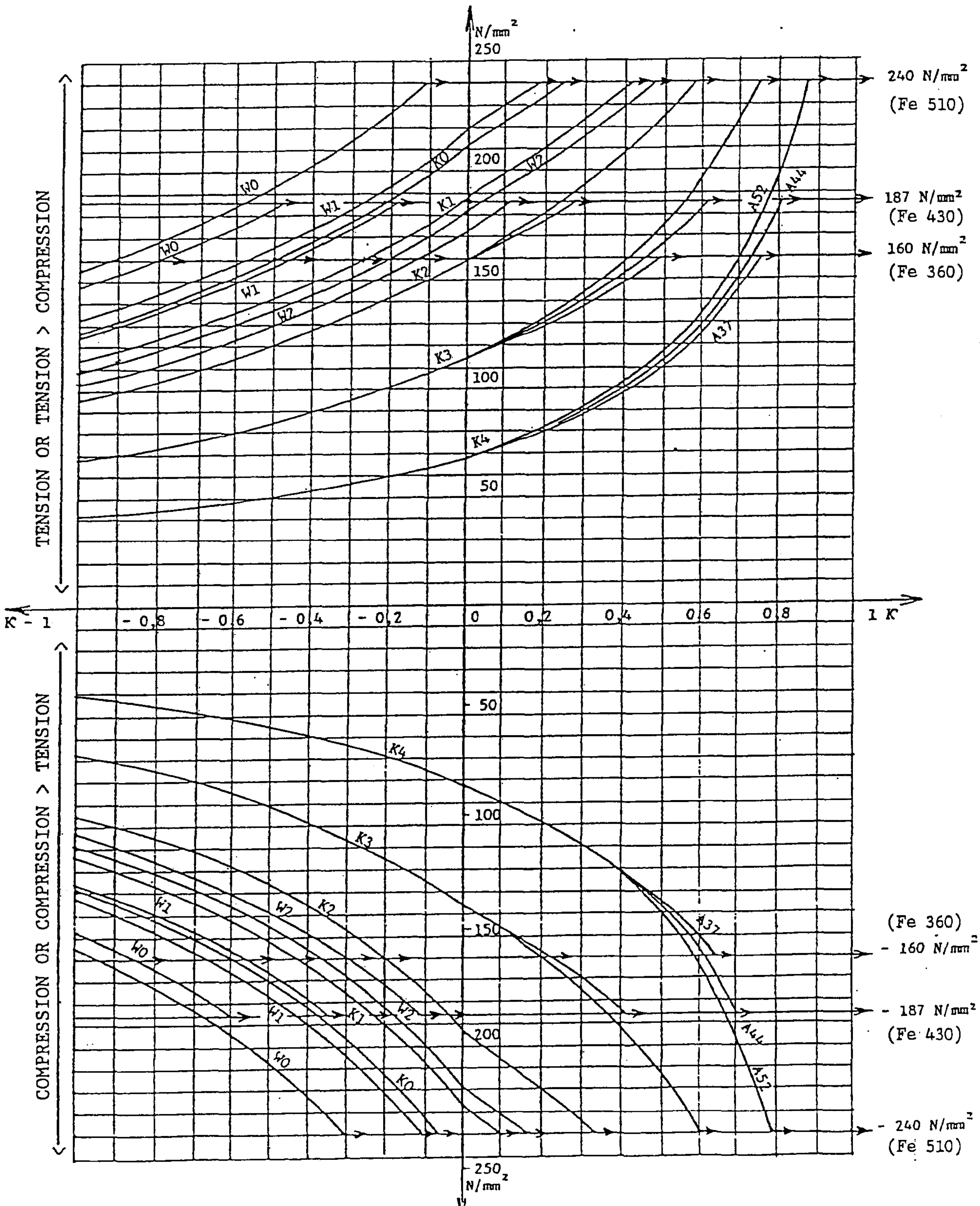


Table T.3-4.5.3.3

PERMISSIBLE FATIGUE STRESSES

Tension and compression in the material and in the weld seams
for construction cases W_0 to W_2 and K_0 to K_4
and for steels Fe 360, Fe 430 and Fe 510

COMPONENTS OF GROUP E7

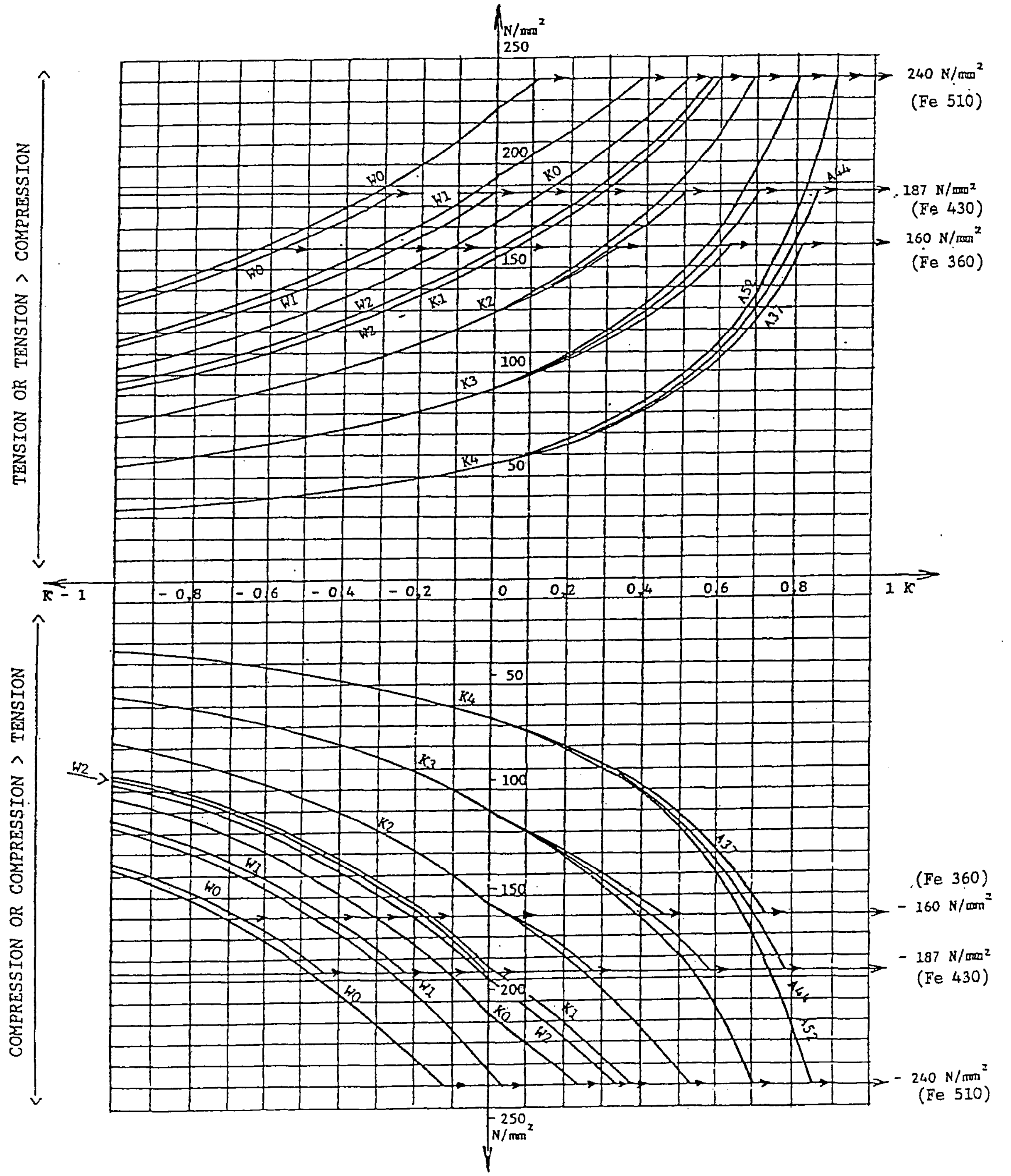
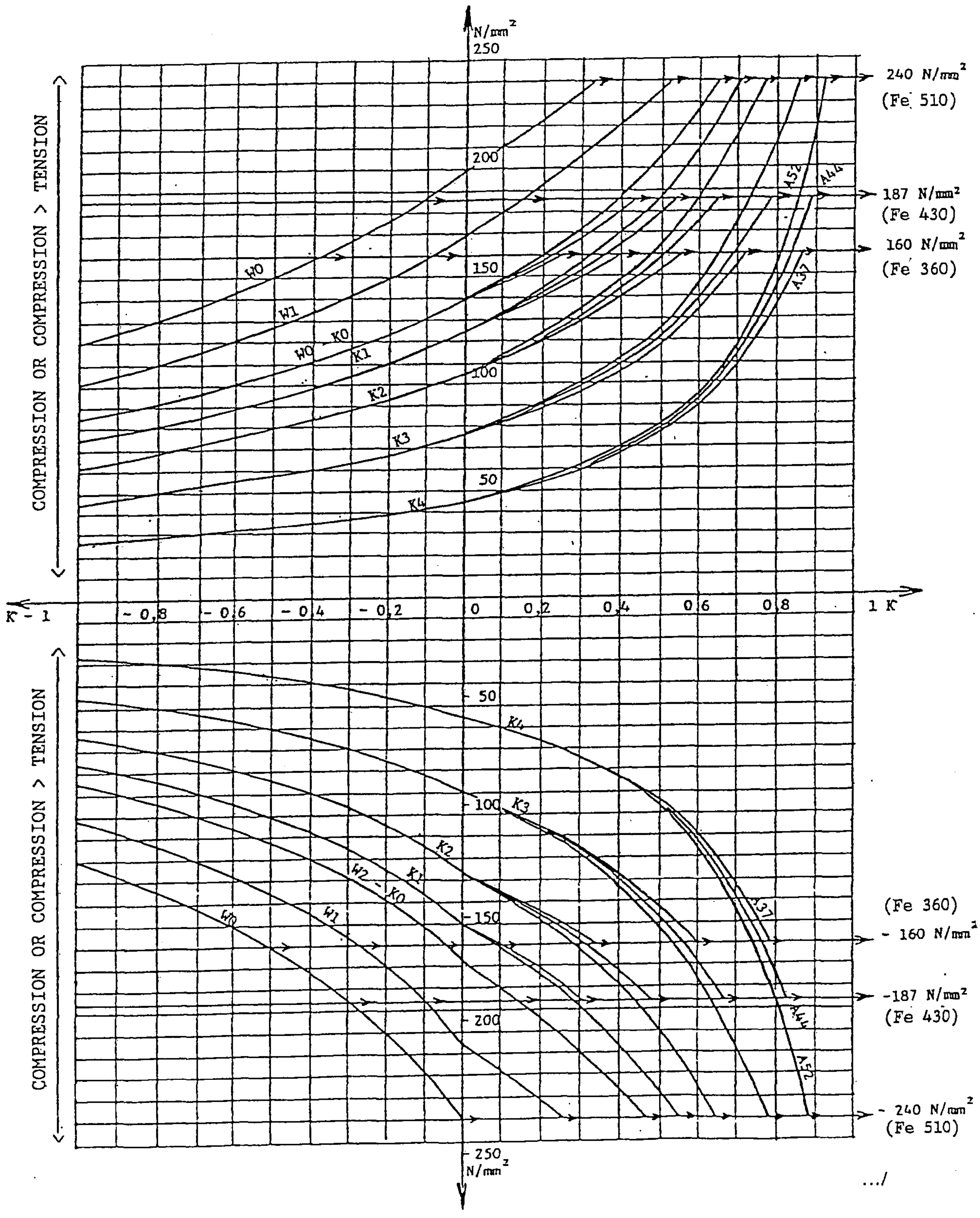


Table T.3-4.5.3.4

PERMISSIBLE FATIGUE STRESSES

Tension and compression in the material and in the weld seams
for construction cases W_0 to W_2 and K_0 to K_4
and for steels Fe 360, Fe 430 and Fe 510

COMPONENTS OF GROUP E8



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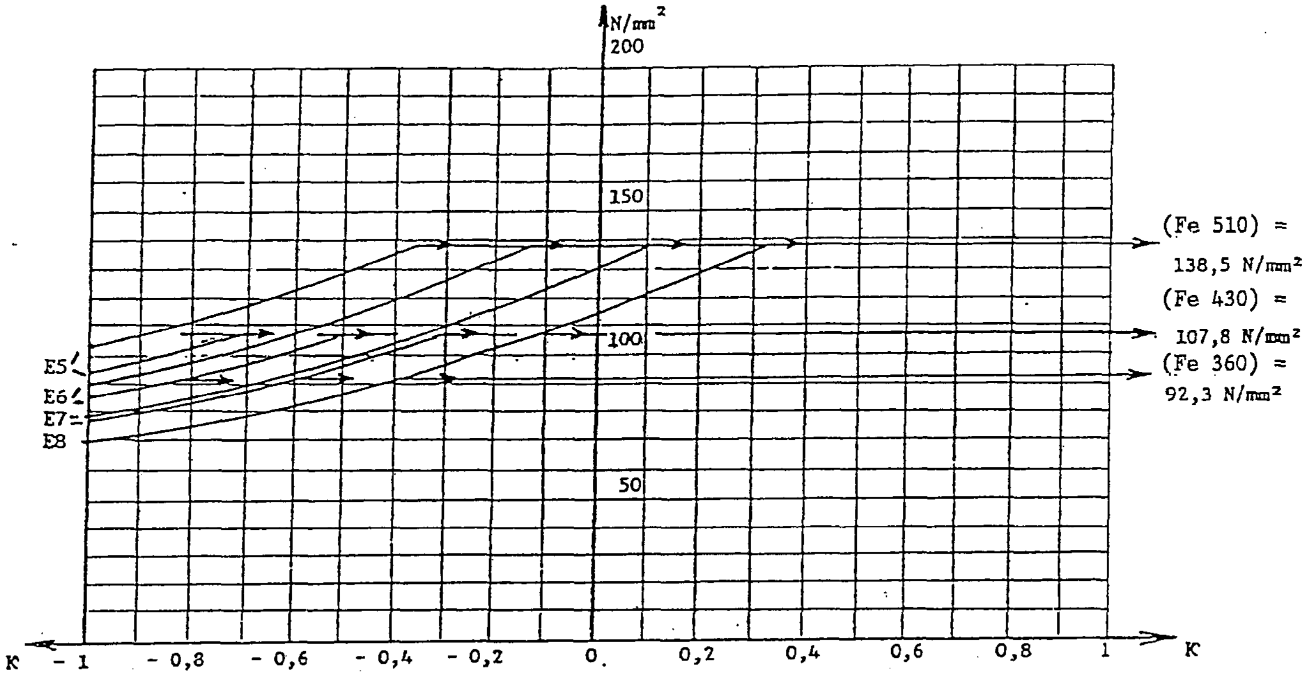
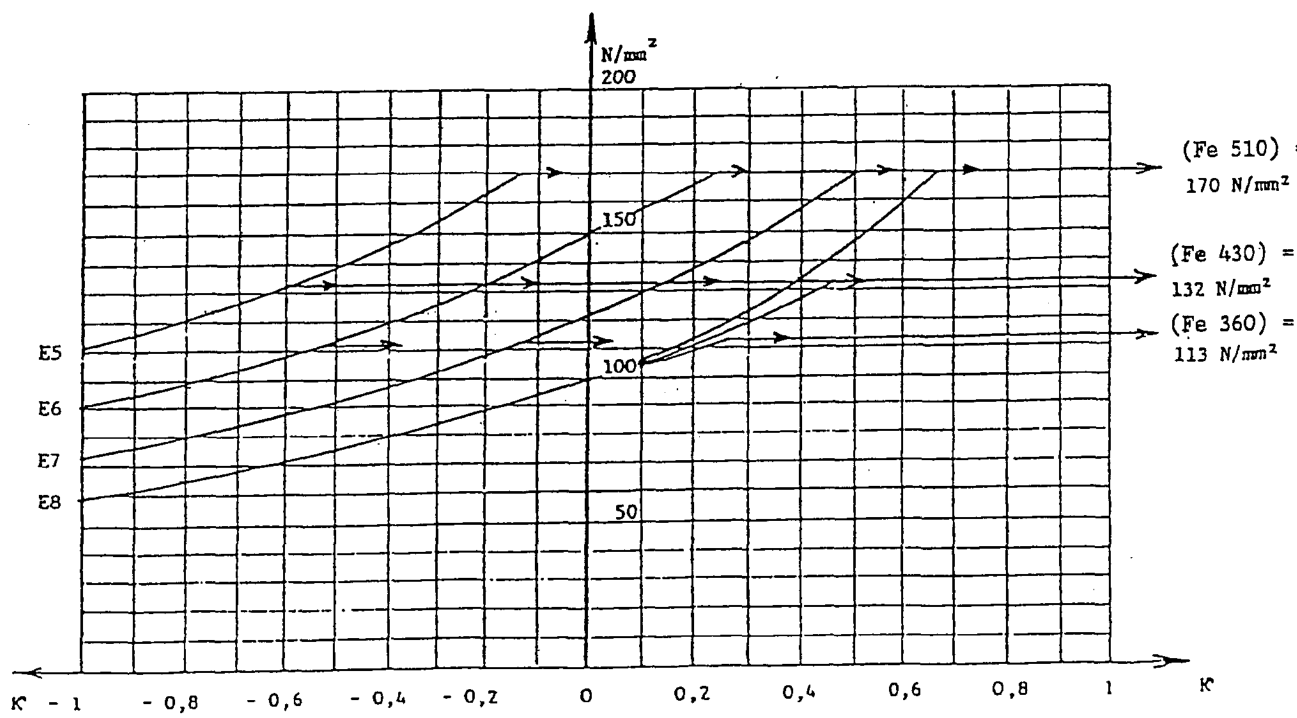
TableS T.3-4.5.3.5
PERMISSIBLE FATIGUE STRESSES
for shear in parent metal and in weld seams steels Fe 360, Fe 430 and Fe 510
and for components of groups E5 to E8

Shear in parent metal



Shear in weld seams



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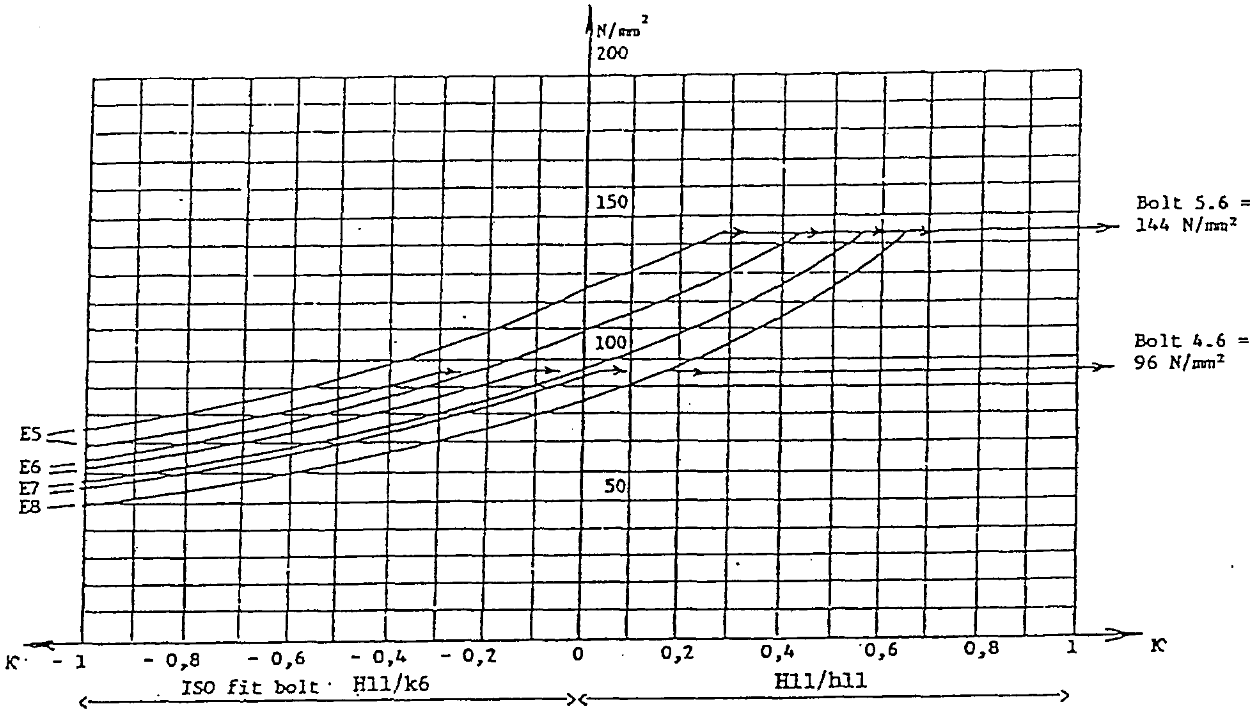
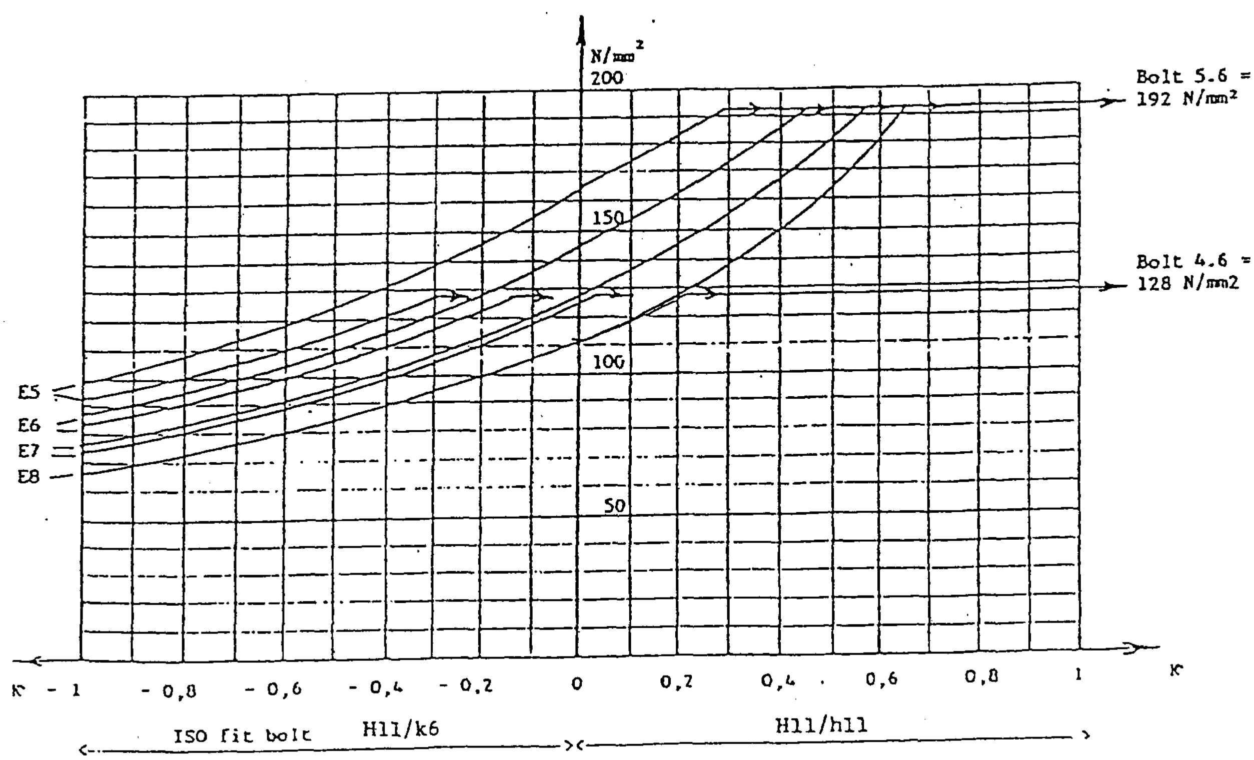
Tables T.3-4.5.3.6

PERMISSIBLE FATIGUE STRESSES

Shear in fitted bolts - grade 4.6 and 5.6
for groups E5 to E8



Double shear joint



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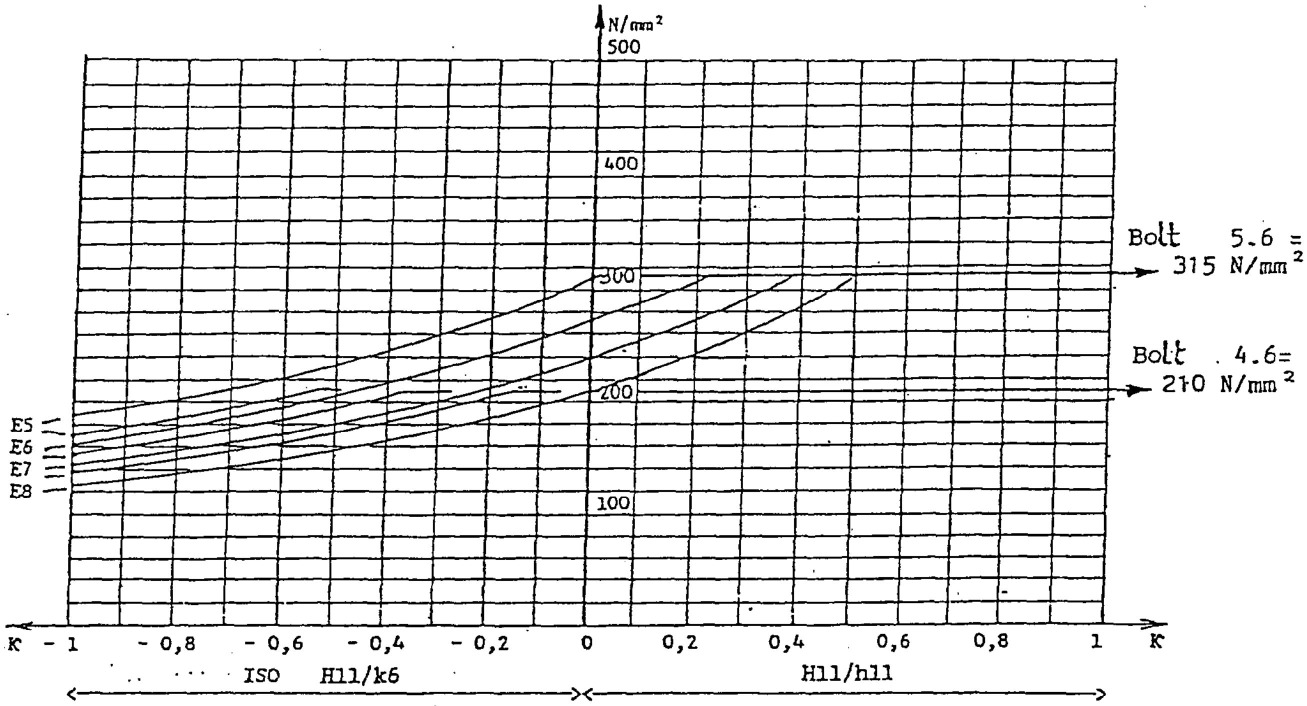
Tables T.3-4.5.3.7

PERMISSIBLE BEARING STRESSES

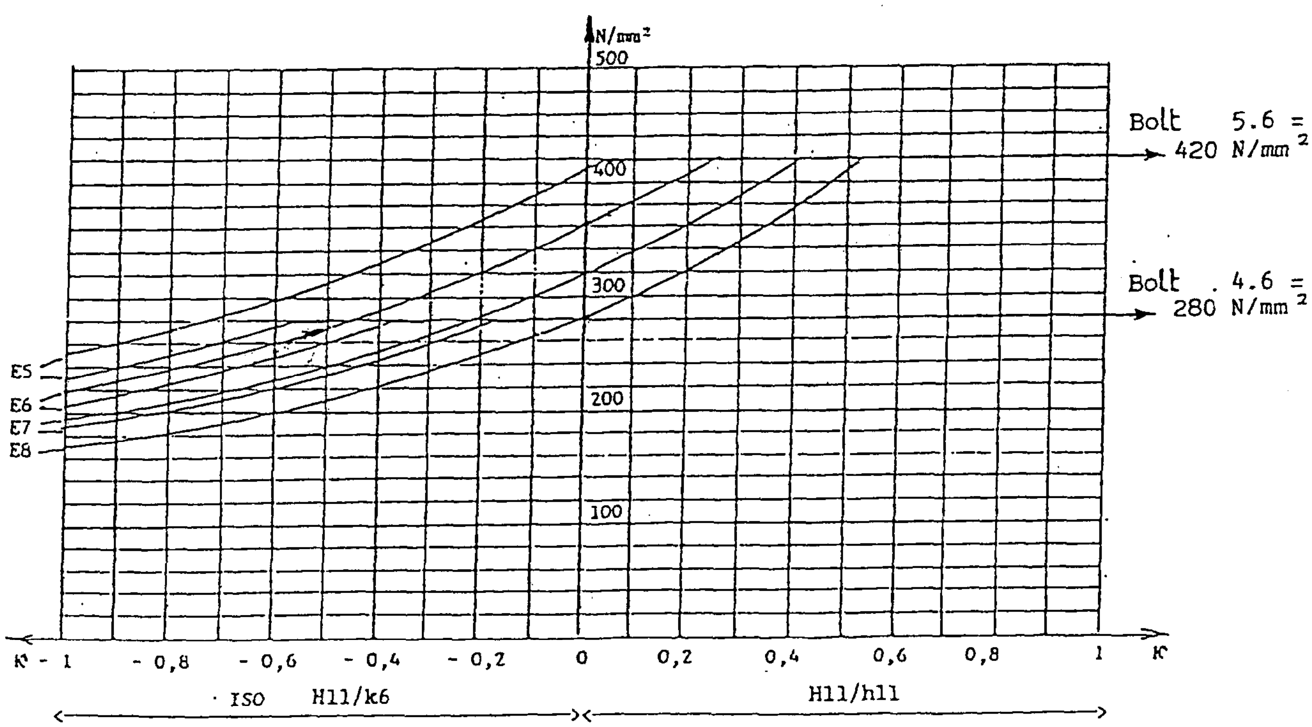
for fitted bolts - grade 4.6 and 5.6

for groups E5 to E8

Single shear joint



Double shear joint



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3-5 CHECK ON "AS BUILT" STRUCTURE

The final construction weights must be compared with the weights used in the static calculation. Where the final dead loads do not exceed the weights used in the static calculation by more than 5 %, there is no need to carry out a new check.

3-6 SAFETY AGAINST OVERTURNING

3-6.1 CHECKING FOR STABILITY

For safety against overturning, the following ratio shall be applied :

$$\nu_K = \frac{M_S}{M_k}$$

where

M_S is the stabilizing moment of the total permanent load G referred to a possible tipping axis

M_k is the overturning moment resulting from all the variable horizontal and vertical forces ($\Sigma P_H + \Sigma P_V$) of load cases I, II and III, to the extent these forces increase the overturning moment.

The check must be carried out for the tipping axis with the smallest overturning safety, by assuming that the movable parts of the dead load are in the most unfavourable position.

The same safety regarding overturning can be written in the following form :

$$\nu_K = \frac{M_S}{M_k} = \frac{f}{e} \cdot \frac{1}{1 - \frac{\Sigma P_V}{G}\left(\frac{f}{e} - 1\right)}$$

where

f is the horizontal distance of the center of gravity of the dead load G with respect to a possible overturning axis

and

e is calculated from the following formula :

$$e = \frac{\Sigma P_H \cdot h + \Sigma P_V \cdot (a + f)}{G + \Sigma P_V}$$

where

h is the vertical distance of the sum of all (ΣP_H) from the tipping axis

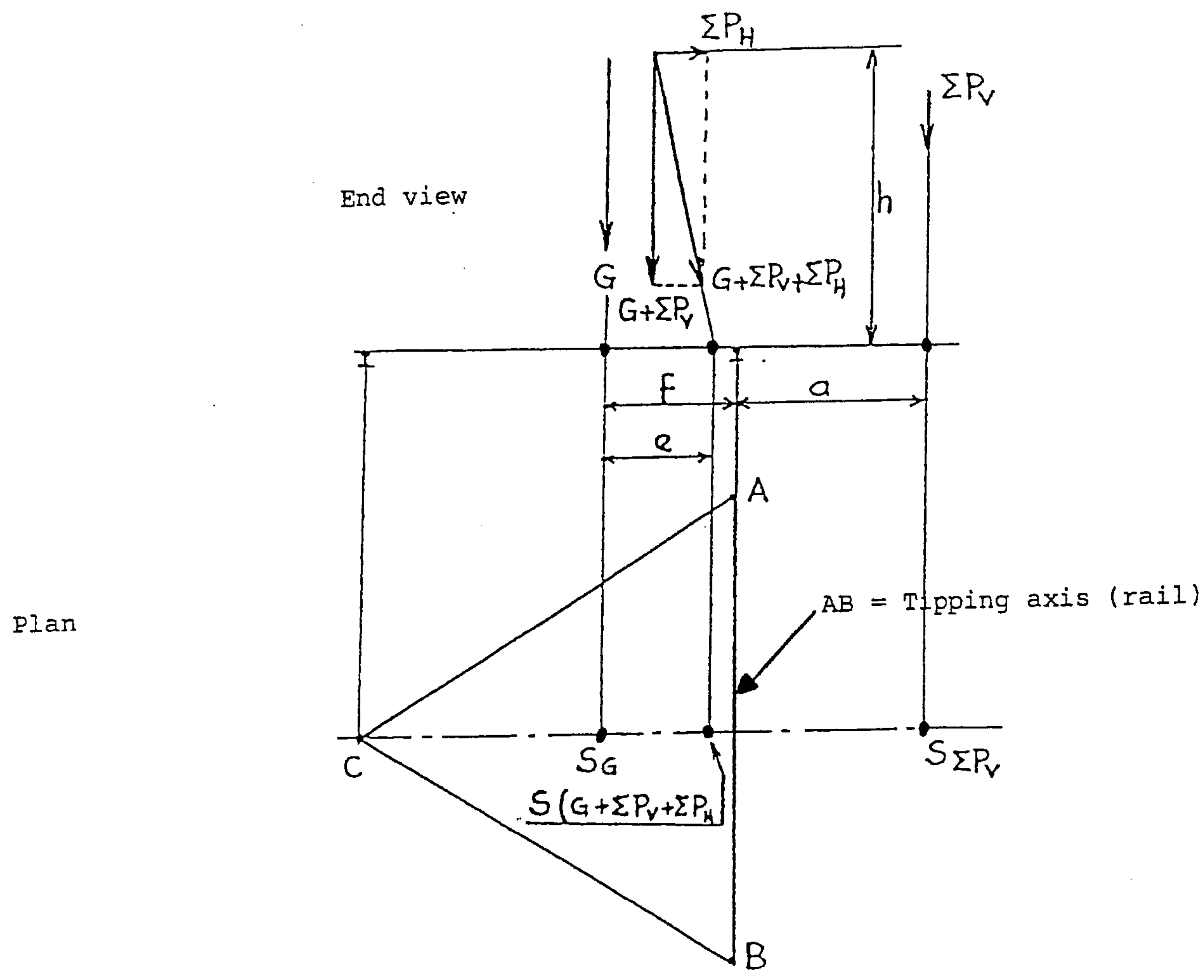
a is the horizontal distance of the sum of all the vertical forces (ΣP_V) from the tipping axis.

The following safeties against overturning are at least request for the load cases I to III :

Table T.3-6.1

SAFETY AGAINST OVERTURNING $v_K = \frac{M_s}{M_k}$

Load case	v_K
I	≥ 1.50
II	≥ 1.30
III	≥ 1.20



3-6.2 ADDITIONAL PRECAUTIONS

In agreement with the user, it can be specified that for special situations, some structural members must occupy definite positions, taking into account the stability of the appliances when idle or out of service (for example : reclaimer boom). Such measures must appear in the operating instructions.

3-7 SAFETY AGAINST DRIFTING

As safety regarding drifting, the ratio is taken between the sum of the drag forces by the sum of the drift forces due to the wind or the inclination. The calculation shall be based on the greatest inclination at which the machine has to work. The resting of the reclaiming device on the face or ground (see 2-2.3.2) must not, in this case, be taken into consideration.

The friction values to be used are as follows :

- for driven wheels on rails : $\mu = 0.14$
- for non driven ball mounted wheels : $\mu = 0.01$
- for non driven wheels with bushes : $\mu = 0.015$
- for rail clamps, if no higher values are found by testing : $\mu = 0.25$

Note : the values of friction coefficients given above are greater thant the minimum values of chapter 2-4.2.1 since they correspond to a static state.

The safety regarding drifting must be :

a) during operation, when only the automatic brakes of the drive motors act, in case of wind induced load, according to 2-2.2.1.2 : $\nu \geq 1.30$,

b) outside of operation, with wind induced stresses according to 2-2.3.6 : $\nu \geq 1.20$.

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Chapter 4

CALCULATION AND CHOICE OF MECHANISM COMPONENTS

CHAPTER 4

CALCULATION AND CHOICE OF MECHANISM COMPONENTS

CONTENTS

APPENDIXES

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4-1 CALCULATION PROCEDURE

Mechanism components are designed by checking that they offer adequate safety against failure due to fracture, crippling, fatigue or excessive wear.

Other factors must also be taken into consideration and it is particularly important to avoid overtheating or deflection which could interfere with correct functionning of the mechanism.

4-1.1 CHECKING FOR ULTIMATE STRENGTH

Mechanism components are checked for ultimate strength by verifying that the calculated stress does not exceed a permissible stress which is dependent on the breaking strength of the material used.

4-1.1.1 VALUE OF THE PERMISSIBLE STRESS

The value of the permissible stress σ_a is given by the following formula :

$$\sigma_a = \frac{\sigma_R}{\nu_R}$$

where :

σ_R is the ultimate stress for the material
ν_R is a safety coefficient corresponding to each load case mentioned at section 2-5.

4-1.1.2 VALUES OF THE COEFFICIENT ν_R

The values to be adopted for ν_R are given intable T.4-1.1.2.

Table T.4-1.1.2
Values of ν_R

case of loading	I	II
value of ν_R	2.2	1.8

In the case of grey cast iron, the values of ν_R are to be multiplied by 1.25.

.../

4-1.1.3 RELATIONS BETWEEN THE CALCULATED STRESSES AND THE PERMISSIBLE STRESSES

According to the type of loading considered, the following relations must be verified in which :

σ_t is the calculated tensile stress,
σ_c is the calculated compressive stress,
σ_f is the calculated bending stress,
τ is the calculated shear stress.

1) Pure tension : $1.25\ \sigma_t \leq \sigma_a$

2) Pure compression : $\sigma_c \leq \sigma_a$

3) Pure bending : $\sigma_f \leq \sigma_a$

4) Combined bending and tension : $1.25\ \sigma_t + \sigma_f \leq \sigma_a$

5) Combined bending and compression : $\sigma_c + \sigma_f \leq \sigma_a$

6) Pure shear : $\sqrt{3}\ \tau \leq \sigma_a$

7) Combined tension, bending and shear : $\sqrt{(1.25\ \sigma_t + \sigma_f)^2 + 3\ \tau^2} \leq \sigma_a$

8) Combined compression, bending and shear : $\sqrt{(\sigma_c + \sigma_f)^2 + 3\ \tau^2} \leq \sigma_a$

4-1.2 CHECKING FOR CRIPPLING

Parts subject to crippling should be designed in accordance with section 3-3, checking that the calculated stress does not exceed a limiting crippling stress.

4-1.3 CHECKING FOR FATIGUE

General remark :

The manufacturer shall be free to choose the relevant method to check the fatigue strength of an element. He should of course refer to a recognized and proven method, thus being able to justify his choice.

Many books have been written on this subject and a list of a few books dealing with fatigue problems is given at the end of this chapter

Many studies are also under way and these will no doubt help to improve current knowledge in this area.

If a manufacturer does not use an alternative method, he can use the method described hereafter which is based on specimen tests.

4-1.3.1 GENERAL METHOD

The fatigue strength of a given component is mainly determined by :

- the material from which the component is constructed,
- the shape, surface condition, state of corrosion, size (scale effect) and other factors producing stress concentrations,
- the ratio κ between the minimum and maximum stresses which occur during the various stress cycles,
- the stress spectrum,
- the number of stress cycles.

The fatigue strength of a complete mechanical component is known only in exceptional cases. Generally speaking, it has to be derived from the characteristics of the material and the component and from accepted laws concerning their behaviour.

The starting point is provided by the endurance limit under alternating tensile fatigue loading ($\kappa = -1$) of a polished specimen, made from the material under consideration. The reduction of this fatigue strength as a result of the geometric shape of the piece, its surface condition, its state of corrosion and its size is allowed for by introducing appropriate factors.

From the endurance limit under alternating loading the corresponding limit with respect to other values of κ can be obtained with the aid of a Smith diagram, in which certain hypotheses are made as to the shape of the strength curve.

The endurance limtit thus determined for the actual component, with respect to a given ratio κ between extreme stresses, is taken as the basis for plotting the Wöhler curve. From this Wöhler curve (fatigue strength under the effect solely of stress cycles, all having the same ratio κ between extreme stresses), and by using the Palmgren-Miner hypothesis on fatigue damage accumulation, the fatigue strength of a component can be determined according to the component group in which it is classified.

This method for determining the fatigue strength is applicable only to components in which the structure of the material is homogenous over the entire section being considered. It cannot, therefore, be used in the case of components which have undergone a surface treatment (e.g. hardening, nitriding, casehardening). In such cases the fatigue strength can be derived from the Wöhler curve only if the latter has itself been determined for components which have been made from the same material, have a comparable shape and size and have undergone exactly the same surface treatment.

Checking for fatigue strength only needs to be performed for load case I.

Where the number of stress cycles is less than $8 . 10^3$, such checks are not necessary.

.../

4-1.3.2 ENDURANCE LIMIT FOR A POLISHED SPECIMEN UNDER ALTERNATING LOADING

The specialized works on the subject provide the endurance limit value σ_{bw} under alternating rotational bending of a polished specimen for materials used regularly in the construction of mechanisms.

The tables T.4-1.3.2a and b give this fatigue strength σ_{bw} for some commonly used steels as an example.

The same values of σ_{bw} may be accepted as an approximation for the endurance limit under alternating bending in a plane.

To obtain the endurance limit under alternating axial tension and compression, the values of σ_{bw} have to be decreased by 20 % *(1).

The endurance strength τ_w under alternating shear (pure shear or torsion) is derived from σ_{bw} by the relation :

$$\tau_w = \frac{\sigma_{bw}}{\sqrt{3}}$$

The values given for σ_{bw} are generally those corresponding statistically to a 90 % survival probability. In the case of carbon steels commonly used in mechanisms, it is permissible to adopt :

$$\sigma_{bw} = 0.5\ \sigma_R$$

σ_R being the minimum ultimate strength.

The components subject to combined loads must be checked using methods from specialized works.

---

*(1) An element of material, when subjected to the same stress as an adjacent element, supports the latter less effectively than if it were subjected to a lower stress, as is the case with bending. A stress gradient, i.e. :

$$\frac{\text{difference in stress between two adjacent elementary parts}}{\text{distance between these two elementary parts}}$$

which is higher, produces a strengthening effect.

Table T.4-1.3.2.a
Characteristics of some commonly used steels

Ultimate strength σ_R, yield strength σ_E and fatigue strength σ_{bw}, at room temperature. For other steels or others temperatures, refer to the relevant standard.

Note : The new european standards under preparation give comparison tables for steel grades between european standards, ISO 683 - 1 (1987) and other anterior national standards.

We give, in appendix, at the end of chapter 4, a comparison table taken from EN 10 083 - 1.

order	symbols	according to	σ_E (*1) [N/mm²]	σ_R [N/mm²]	σ_{bw} (*2) [N/mm²]
1	Fe 360 B	ISO 630	240	370	185
2	A St 41	DIN 17 135	d ≤ 16 : 260	410	205
			16 < d ≤ 40 : 250		200
			40 < d ≤ 60 : 240		195
3	St 50	DIN 17 100	300	500	250
4	Fe 510 C	ISO 630	d ≤ 16 : 360	520	260
			16 < d ≤ 30 : 350		255
			30 < d ≤ 50 : 340		250
5	St 60	DIN 17 100	340	600	300
6	St 70	DIN 17 100	370	700	350
7	C 35 N	DIN 17 200	280	500	250
8	C 35 V Ck 35 V	DIN 17 200	d ≤ 16 : 430	630	315
			16 < d ≤ 40 : 370	590	295
			40 < d ≤ 100 : 330	550	275
9	C 45 N	DIN 17 200	340	600	300
10	C 45 V Ck 45 V	DIN 17 200	d ≤ 16 : 490	710	355
			16 < d ≤ 40 : 420	670	335
			40 < d ≤ 100 : 380	630	315
11	C 60 N	DIN 17 200	390	700	350
12	C 60 V Ck 60 V	DIN 17 200	d ≤ 16 : 580	850	425
			16 < d ≤ 40 : 500	800	400
			40 < d ≤ 100 : 400	750	375
13	34 CrMo 4	ISO 683-1	380 (*3)	650	325
14	34 CrMoS 4	ISO 683-1	d ≤ 16 : 800	1,000	500
			16 < d ≤ 40 : 680	900	450
			40 < d ≤ 100 : 570	800	400
			100 < d ≤ 160 : 520	750	375
			160 < d ≤ 250 : 470	700	350
15	42 CrMo 4	ISO 683-1	400 (*3)	700	350
16	42 CrMoS 4	ISO 683-1	d ≤ 16 : 900	1,100	550
			16 < d ≤ 40 : 780	1,000	500
			40 < d ≤ 100 : 650	900	450
			100 < d ≤ 160 : 570	800	400
			160 < d ≤ 250 : 520	750	375
17	50 CrMo 4 N	ISO 683-1	430 (*3)	750	375
18	50 CrMo 4 V	ISO 683-1	d ≤ 16 : 900	1,100	550
			16 < d ≤ 40 : 800	1,000	500
			40 < d ≤ 100 : 700	900	450
			100 < d ≤ 160 : 650	850	425
			160 < d ≤ 250 : 600	800	400

Table T.4-1.3.2.b
Characteristics of some commonly used cast materials

Ultimate strength σ_R, yield strength σ_E and fatigue strength σ_{bw} at the normal ambiant temperature. For other cast materials or others temperatures refer to the relevant standards.

order	symbols	according to	σ_E (*1) [N/mm²]	σ_R [N/mm²]	σ_{bw} (*2) [N/mm²]
19	GS-45	DIN 1 681	230	450	
20	GS-60		300	600	
21	GS-70		420	700	
22	GS-34 CrMo 4 V		$s \leq 30$: 520	750	
			$30 < s \leq 100$: 450	700	
23	GS-42 CrMo 4 V		$s \leq 30$: 550	750	
			$30 < s \leq 100$: 500	700	
24	GGG-42	DIN 1 693	280	420	210
25	GGG-60		420	600	300
26	GGG-70		500	700	350

Notes on tables T.4-1.3.2.a and b :

(*1) In this column d is the diameter of the part in mm according to DIN 17135 (March 1964 edition) table 2, or DIN 17200 (December 1969 edition) table 6 ; other section shapes according to DIN 17200 (December 1969 edition), figure 3. For the cast materials, S is the maximum wall thickness.

(*2) The values given will most probably be reached in 90 % of the test with the polished test sample for the rotary bending test according to DIN 50113 (December 1972 edition) (profiled test sample, $r_a : d_o \geq 3$ and constant moment over the whole length of the test sample).

(*3) Approximate values.

4-1.3.3 INFLUENCE OF THE SHAPE, SIZE, SURFACE CONDITION AND CORROSION

The shape, size, surface finish (machining) and state of corrosion of the component under consideration bring about a decrease in the endurance limit under alternating loading for the ideal case of a polished specimen.

These influences are allowed for by introducing factors k_s, k_d, k_u (or k_{uc}) respectively. Examples illustrating the determination of these factors are given in appendix A.4-1.3.

The endurance limit under alternating loading σ_{wk} or τ_{wk} for any component is given for tension, compression, bending and torsional shear by the equation :

$$\sigma_{wk} = \frac{\sigma_{bw}}{k_s.k_d.k_{u(c)}}$$

or

$$\tau_{wk} = \frac{\tau_w}{k_s.k_d.k_{u(c)}}$$

In the case of pure shear we take :

$$\tau_{wk} = \tau_w$$

4-1.3.4 ENDURANCE LIMIT AS A FUNCTION OF κ

Fig. 4-1.3.4 expresses, in the form of a Smith diagram, the hypotheses made concerning the relationship between the endurance limit σ_d (or τ_d) and the ratio κ between the extreme stresses.

This gives the following relations :

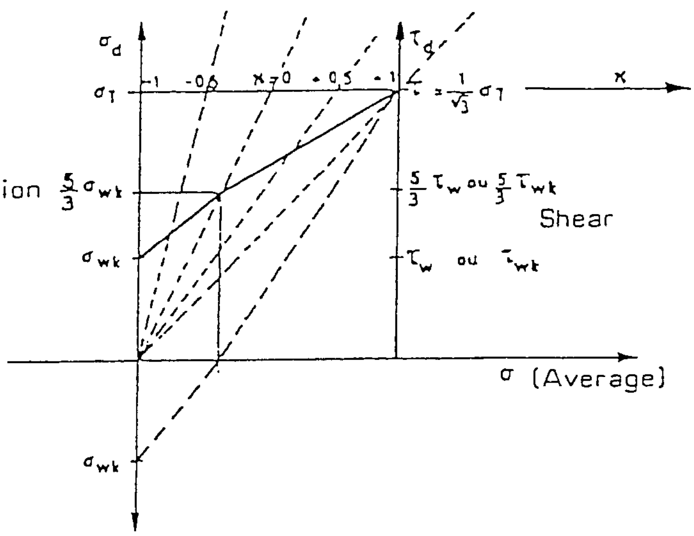
normal stresses	$-1 \leq \kappa < 0$	$\sigma_d = \frac{5}{3 - 2\kappa}\ \sigma_{wk}$	alternating stresses
	$0 \leq \kappa \leq 1$	$\sigma_d = \frac{\frac{5}{3}\sigma_{wk}}{1 - [1 - \frac{\frac{5}{3}\sigma_{wk}}{\sigma_R}] . \kappa}$	pulsating stresses
shear stresses	$-1 \leq \kappa < 0$	$\tau_d = \frac{5}{3 - 2\kappa}\ \tau_{wk}$	alternating stresses
	$0 \leq \kappa \leq 1$	$\tau_d = \frac{\frac{5}{3}\tau_{wk}}{1 - [1 - \frac{\frac{5}{3}\sqrt{3}\,\tau_{wk}}{\sigma_R}] . \kappa}$	pulsating stresses

in which

σ_R = tensile strength of the material
σ_{wk} and τ_{wk} = endurance limit of the component under alternating loading, $\kappa = -1$

.../

Figure 4-1.3.4



4-1.3.5 WÖHLER CURVE

In this context, the "Wöhler curve" is showing 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 κ between extreme values.

The following hypotheses are made :

- for $n \leq 8 \cdot 10^3$:

$$\sigma = \sigma_R$$

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 system comprising two logatithmic scale axes (fig. 4-1.3.5).

  In this region, the slope of the Wöhler curve is given by the factor :

$$c = \text{tg}\ \varphi = \frac{\log(2.10^6) - \log(8.10^3)}{\log \sigma_R - \log \sigma_d}$$

or

$$c = \text{tg}\ \varphi = \frac{\log(2.10^6) - \log(8.10^3)}{\log\left(\frac{\sigma_R}{\sqrt{3}}\right) - \log \tau_d}$$

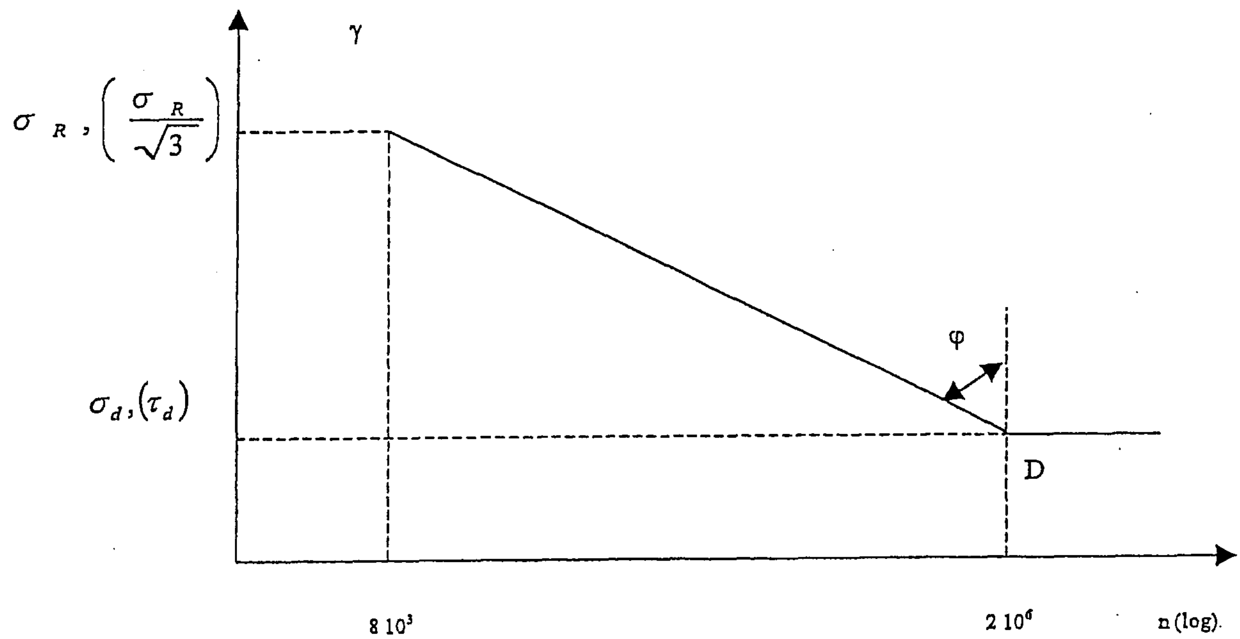
- for $n \geq 2 \cdot 10^6$:

$$\sigma = \sigma_d$$

or

$$\tau = \tau_d$$

Figure 4-1.3.5



The spectrum factor k_{sp} of the component (see section 2-1.4.3) 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 :

$$\sigma k = \left(2^{\frac{8-j}{c}} \right) . \sigma d$$

or

$$\tau k = \left(2^{\frac{8-j}{c}} \right) . \tau d$$

where j is the component's group number (see section 2-1.4.4)

The classification of components, grouped on the basis of their total duration of use N and their spectrum factor k_{sp}, as well as the critical fatigue stresses associated with each group, is illustrated in figure 4-1.3.6 where σ_{jk} represents the critical stress applying to group Ej. For the critical shear stresses, the letter σ must be replaced by τ.

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 ν_k.
One takes :

$$\nu k = 3.2^{\frac{1}{c}}$$

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

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

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

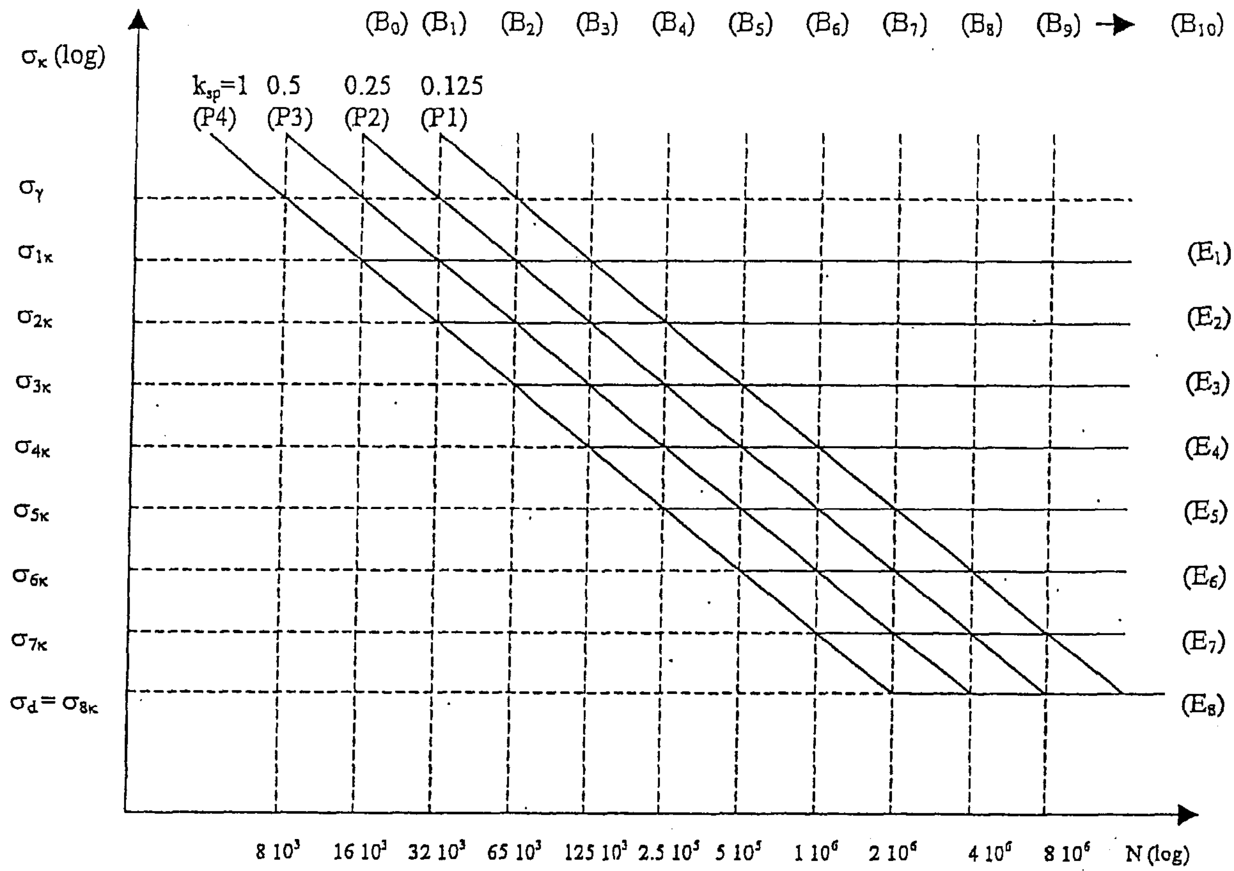
and one verifies that :

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

with

σ maximum calculated normal stress
τ maximum calculated shear stress

Figure 4-1.3.6



For components simultaneously experiencing both normal and shear stresses with different ratios κ between extreme stresses, the following condition must be satisfied :

$$\left(\frac{\sigma_x}{\sigma_{kx}}\right)^2 + \left(\frac{\sigma_y}{\sigma_{ky}}\right)^2 - \left(\frac{\sigma_x \cdot \sigma_y}{\sigma_{kx} \cdot \sigma_{ky}}\right) + \left(\frac{\tau}{\tau_k}\right)^2 \leq \frac{1.1}{\nu_k^2}$$

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 from this to determine the most unfavourable case calculations must be performed separately for the loads $\sigma_{x\ max}$, $\sigma_{y\ max}$ and τ_{max} and the most unfavourable corresponding stresses used.

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.

4-1.4 CHECKING FOR WEAR

In the case of parts subjected to wear, the specific physical quantities which affect this, such as the surface pressure or the speed, must be determined. The figures must be such that they will not lead to excessive wear on the basis of present experience.

4-2 DESIGN CALCULATIONS FOR PARTICULAR COMPONENTS

4-2.1 CHOICE OF ANTI-FRICTION BEARINGS

To select anti-friction bearings, it is first necessary to check that the bearing is capable of withstanding :

- the static load to which it can be subjected under either load case I or II, whichever is the most unfavourable, and
- the maximum dynamic load in the more unfavourable of load case I or II.

4-2.1.1 THEORETICAL LIFE

In addition, anti-friction bearings must be selected to give an acceptable theoretical life in hours (see table T.2-1.3.2) suitable for the mechanism's class of operation under an "equivalent" constant mean load as defined below.

4-2.1.2 MEAN LOADING OF BEARINGS

. In order to allow for variations in the loads during the cycles of operation, an equivalent mean loading S_{mean} is determined which is supposed to be applied constantly during the theoretical expected life.

S_{mean} is obtained by multiplying S_{maxI} defined by clause 2-5.1, by the cube root of the spectrum factor k_m defined in 2-1.3.3.

$$S_{mean} = \sqrt[3]{k_m} \cdot S_{max\ I}$$

. Specific case :
The bearings of travelling wheels are designed as follows :

The extreme loads S_{max} and S_{min} developed in loading case I are considered and the bearing is designed for an equivalent mean load given by the expression :

$$S_{mean} = \frac{2\ S_{max\ I} + S_{min\ I}}{3}$$

and applied for the theoretical expected life.

4-2.1.3 CHOICE OF BEARINGS

After defining the maximum and the equivalent mean values of loads, both radial and axial loads, it is necessary to combine these values in accordance with the recommendations given by the bearing manufacturer. It is then possible to select the appropriate type of bearing from the catalogue.

For special bearings such as slewing rings for instance, the selection should be made by the bearing manufacturer once he has been informed of critical values such as : axial and radial loads, overturning moments in "normal" operation, in "maximum" operation, maximum (static) out-of-operation conditions, etc, as well as tooth forces in the case of a toothed ring.

It is absolutely necessary for the bearing manufacturer to have, together with all these values, a clear understanding as to the operation of the equipemnt and therefore of what the given values correspond to.

4-2.2 CHOICE OF ROPES

The following rules are aimed at defining reasonable minimum requirements for the choice of ropes.

They do not purport to resolve every problem not to serve as a substitute for the dialogue which is essential between the rope manufacturer and the manufacturer of handling appliances.

They apply to preferred ropes conforming to ISO recommendation 2408 "Steel ropes for general use - Characteristics".

They do not exclude, however, ropes which are not specified in ISO recommendation 2408, for which it is incumbent upon the rope manufacturer to validate for the user the minimum values of parameters detailed in the ISO recommendation.

The terminology of the rope parameters complies with that used in ISO recommendation 2408.

The methods stated hereafter assume that the ropes are greased correctly, that the winding diameters of the pulleys and the drums are suitably selected in compliance with 4-2.3 and that, when in service, the ropes are properly maintained, inspected and periodically replaced in accordance with ISO recommendation 4309 "Rope inspection".

The selection of rope diameter (and winding diameters in 4-2.3) is based on the group of the mechanism.

4-2.2.1 CHOICE OF ROPE DIAMETER

The following method is applicable to running ropes (active ropes).

4-2.2.1.1 MAXIMUM TENSILE FORCE S

The maximum tensile force S in the rope is obtained by taking account of the following factors :

- maximum safe working load of the appliance,
- mechanical advantage due to the rope reeving,
- efficiency of the rope reeving,
- loads due to acceleration if they exceed 10 % of the main loads,
- rope inclination to the load axis in the "worst case" if this angle exceeds 22°30'.

4-2.2.1.2 PRACTICAL FACTOR OF SAFETY Z_p

The practical factor of safety Z_p is the ratio between :

- the minimum breaking load F_0 of the rope (minimum load which must be attained when carrying out the rope breaking test),
- and the maximum tensile force S in the rope :

$$Z_p = \frac{F_0}{S} \geq Z_{p\ min}$$

The chosen rope must have a practical factor of safety at least equal to the minimum value Z_p for the mechanism group to which the rope in question belongs (see table T.4-2.2.1.2).

Table T.4-2.2.1.2

group of mechanism	minimum value Z_p (running ropes)
M 1	3.15
M 2	3.35
M 3	3.55
M 4	4
M 5	4.5
M 6	5.6
M 7	7.1
M 8	9

Nevertheless in the case where the failure of a running rope (luffing mechanism for instance) would affect the stability of the machine, these ropes must be chosen with the following minimum safety coefficients :

case	winch rope (running)	safety coefficient
I and II	one rope system two ropes system two ropes system after failure of one rope	6 6 3

Note : Two ropes systems should be preferred.

4-2.2.1.3 MINIMUM BREAKING LOAD F_0

The minimum breaking load is :

$$F_0 = d^2 \cdot \frac{\pi}{4} \cdot f \cdot k \cdot R_0 = d^2 \cdot K' \cdot R_0$$

where :

d = nominal diameter of the rope (dimension by which the rope is designated)
f = fill factor of the rope
k = spinning loss factor due to the rope construction
R_0 = minimum ultimate tensile stress of the wire composing the rope

and

$$K' = \frac{\pi}{4} \cdot f \cdot k$$

4-2.2.1.4 ROPE DIAMETER SELECTION

For a rope of a given construction, having a given minimum steel strength, and for a given mechanism group there is a factor C which is expressed by the formula :

$$C = \sqrt{\frac{Z_p \min}{k \cdot f. \frac{\pi}{4} . R_0}} = \sqrt{\frac{Z_p \min}{K' . R_0}}$$

where Z_{pmin} is the minimum value for running ropes in table T.4-2.2.1.2.

The nominal diameter d must be such that :

$$d \geq C \sqrt{S}$$

The factor K' (or factors k and f) can either :

. be taken from ISO recommendation 2408 for the ropes covered therein,
. or be guaranteed by the rope manufacturer if the rope is of special construction. In this case, the certificate supplied by the rope manufacturer must clearly state the guaranteed values.

4-2.3 CHOICE OF PULLEYS, DRUMS AND ROPE ATTACHMENT MEANS

4-2.3.1 MINIMUM WINDING DIAMETER

The minimum winding diameter for the rope is determined by checking the relationship :

$$D \geq H . d$$

where :

D is the winding diameter on pulleys, drums or compensating pulleys measured to the axis of the rope
H is a coefficient depending upon the mechanism group
d is the nominal diameter of the rope.

4-2.3.1.1 VALUES OF COEFFICIENT H

The minimum values of the coefficient H depend upon the group in which the mechanism is classified, and are given in table T.4-2.3.1.1 for drums, pulleys and compensating pulleys.

They correspond to ropes in common use and are based on experience of their working conditions.

These guidelines do not however serve as substitute for the dialogue which is indispensable between the rope manufacturer and the manufacturer of handling appliances, especially when the use of new ropes with non standard flexibility characteristics is being considered.

Table T.4-2.3.1.1
VALUES OF H

mechanism group	drums	pulleys	equalizing sheaves and compensating pulleys
M 1	11.2	12.5	11.2
M 2	12.5	14	12.5
M 3	14	16	14
M 4	16	18	14
M 5	18	20	14
M 6	20	22.4	16
M 7	22.4	25	16
M 8	25	28	18

NOTE : When the formula given in clause 4-2.2.1 has been used to determine a minimum rope diameter from which in turn the minimum diameters for drums and pulleys have been determined, a rope of diameter greater than the minimum calculated diameter can be used with these latter diameters, provided that the diameter of the rope used does not exceed the minimum diameter by more than 25 % and that the pull in the rope does not exceed the value S used for calculating this minimum diameter.

4-2.3.2 RADIUS OF THE BOTTOM OF THE GROOVE

The useful life of the rope depends not only on the diameter of the pulleys and drums, but also on the pressure exerted between the rope and the groove supporting the rope.

The winding ratios above are given on the assumption of a radius of supporting groove r where :

$$r = 0.53\ d$$

d being the nominal diameter of the rope.

4-2.3.3 ROPE ATTACHMENT MEANS

Rope attachment must be so designed as to withstand a tensile force at least 2.5 times the maximum force S without showing permanent deformation.

The means attaching the rope to the drum must be of such a design that, taking account of the friction in the turns remaining around the drum, the sum of the frictional and fixing forces withstands a tensile force at least 2.5 times the maximum tensile force S.

The coefficient of friction between the rope and the drum used in the calculations shall be :

$$\mu = 0.1$$

when the rope is unwound from the drum for the length corresponding to the maximum service position, at least two complete turns of rope must remain on the drum before the rope end attachment.

4-2.4 CHOICE OF RAIL WHEELS

In order to choose a rail wheel, its diameter should be determined by considering :

- the load on the wheel,
- the quality of the material from which it is made,
- the type of rail on which it runs,
- the speed of rotation of the wheel,
- the class of utilization of the mechanism.

Studies concerning the choice of rail wheels exist or are in progress, but the method given hereafter can be used as proved method which lead to satisfactory results in the case of handling machines.

4-2.4.1 RAIL WHEEL SIZE

To determine to size of a rail wheel, the following checks must be made :

- that it is capable of withstanding the maximum load to which it will be subjected, and
- that it will allow the appliance to perform its normal duty without abnormal wear.

The two requirements are checked by means of the following two formulae :

$$\frac{P_{mean\ II}}{b \cdot D} \leq p_L \cdot C_{1max} \cdot C_{2max} \leq 1.38\ p_L \cong 1,.4\ p_L$$

taking $C_{1max} = 1.2$ and $C_{2max} = 1.15$

and

$$\frac{P_{mean\ I}}{b \cdot D} \leq p_L \cdot C_1 \cdot C_2$$

where :

D	the wheel diameter in mm
b	the useful width of the rail in mm
p_L	a (admissible) pressure dependent upon the material used for the wheel, in N/mm²
C_1	a coefficient depending on the speed of rotation of the wheel
C_2	a coefficient depending on the class of utilization of the mechanism
P_{meanII}	the mean wheel load, in N, in load case II calculated according to the formulae hereafter (see 4-2.4.1.1)
P_{meanI}	the mean wheel load in case I.

4-2.4.1.1 DETERMINING THE MEAN WHEEL LOAD

In order to determine the mean wheel loads, the maximum and minimum loads on the wheel should be considered for the appropriate load case I and II.

$$P_{mean} = \frac{P_{min} + 2P_{max}}{3}$$

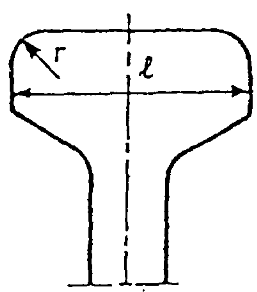
4-2.4.1.2 DETERMINING THE USEFUL RAIL WIDTH b

For rails having a flat bearing surface and a total width l with rounded corners of radius r at each side, we have :

$$b = l - 2\,r$$

For rails with a convex bearing surface, we have :

$$b = l - \frac{4}{3}\,r \qquad *(1)$$



*(1) For the same width of rail head, these formulae give a greater useful bearing width for convex rails than for flat rails. This allows for the superior adaptation of a slightly convex rail to the rolling motion of the wheel.

4-2.4.1.3 DETERMINING THE LIMITING PRESSURE p_L

The value of p_L is given in table T.4-2.4.1.3 as a function of the ultimate strength of the material of which the rail wheel is made.

Table T.4-2.4.1.3
VALUES OF p_L

Ultimate strength for material used for rail wheel	p_L in N/mm²
σ_R > 500 N/mm²	5.0
σ_R > 600 N/mm²	5.6
σ_R > 700 N/mm²	6.5
σ_R > 800 N/mm²	7.2
σ_R > 1,000 N/mm²	8

The qualities of material refer to cast, forged or rolled steels, and spheriodal graphite cast iron.

In the case of rail wheel with steel tire, consideration must obviously be given to the quality of the steel tire, which should be sufficiently thick not to roll itself out.

In the case of wheels made of high tensile steel and treated to ensure a very high surface hardness, the value of p_L is limited to that for the quality of the steel composing the wheel prior to surface treatment, according to table T.4-2.4.1.3, since a higher value would risk causing premature wear of the rail.

For a given load, however, wheels of this type have a much longer useful life than wheels of lesser surface hardness, which makes their use worthwhile for appliances performing intensive service.

Alternatively, it is possible to use wheels of ordinary cast iron, especially chilled cast iron, which has good surface hardness.

It must be remembered that such wheels are brittle and that their use should be avoided for high speed motions or when shock loadings are anticipated.

When these are used, their diameter is determined by taking p_L equal to 5 N/mm².

4-2.4.1.4 DETERMINING THE COEFFICIENT C_1

The values of C_1 depend on the speed of rotation of the wheel and are given in table T.4-2.4.1.4.a.

These same values are also given in table T.4-2.4.1.4.b as a function of the wheel diameter and the speed in m/mn.

Table T.4-2.4.1.4.a
VALUES OF C_1

wheel rotation speed in R.P.M.	C_1	wheel rotation speed in R.P.M.	C_1	wheel rotation speed in R.P.M.	C_1
200	0.66	50	0.94	16	1.09
160	0.72	45	0.96	14	1.1
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	5.6	1.16
63	0.91	20	1.06	5	1.17
56	0.92	18	1.07		

Table T.4-2.4.1.4.b
VALUES OF C_1 AS A FUNCTION OF THE WHEEL DIAMETER
AND THE SPEED OF TRAVEL

wheel diameter in mm	values of C_1 for travel speeds in m/mn														
	10	12.5	16	20	25	31.5	40	50	63	80	100	125	160	200	250
200	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77	0.72	0.66	-	-	-
250	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77	0.72	0.66	-	-
315	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77	0.72	0.66	-
400	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77	0.72	0.66
500	1.15	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77	0.72
630	1.17	1.15	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82	0.77
710	-	1.16	1.14	1.13	1.12	1.1	1.07	1.04	1.02	0.99	1.96	0.92	0.89	0.84	0.79
800	-	1.17	1.15	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87	0.82
900	-	-	1.16	1.14	1.13	1.12	1.1	1.07	1.04	1.02	0.99	0.96	0.92	0.89	0.84
1,000	-	-	1.17	1.15	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91	0.87
1,120	-	-	-	1.16	1.14	1.13	1.12	1.1	1.07	1.04	1.02	0.99	0.96	0.92	0.89
1,250	-	-	-	1.17	1.15	1.14	1.13	1.11	1.09	1.06	1.03	1	0.97	0.94	0.91

4-2.4.1.5 DETERMINING THE COEFFICIENT C_2

The coefficient C_2 depends on the class of utilization of the mechanism and is given in table T.4-2.4.1.5.

Table T.4-2.4.1.5
VALUES OF C_2

classes of utilization	C_2
T0 to T2	1.25
T3 to T5	1.12
T6	1
T7	0.9
T8 - T9	0.8

4-2.4.2 NOTES

Note 1

The formulae apply only to wheels whose diameters do not exceed 1.250 m. For larger diameters experience shows that the permissible pressures between the rail and the wheel must be lowered. The use of wheels of greater diameter is not recommended.

Note 2

It should be noted that the limiting pressure p_L is a notional pressure determined by supposing that contact between the wheel and the rail takes place over a surface whose width is the useful width defined earlier (clause 4-2.4.1.2) and whose length is the diameter of the wheel. The calculation method set out above is derived from the application of the Hertz formula, which may be written :

$$\frac{\sigma_{cg}^2}{0.35\,E} = \frac{P}{b \cdot D}$$

where :

σ_{cg}	is the compressive stress in the wheel and the rail in N/mm²
E	the modulus of elasticity of the material in N/mm²
P	the wheel load in N
b and D	in mm, being as defined above (clause 4-2.4.1)

.../

Taking K_L to represent the value $\frac{\sigma_{cg}^2}{0.35\ E}$ which has the units of a pressure (in N/mm²), the relation may be written :

$$K_L = \frac{P}{b\ .\ D}$$

K_L characterizes the wheel pressure on the rail. The formula of clause 4-2.4.1 is obtained by putting :

$$K_L \leq p_L\ .\ C_1\ .\ C_2$$

4-2.5 DESIGN OF GEARS

The choice of the method for design calculations for gears is left to the manufacturer, who must indicate the origin of the method adopted, the loads to be taken into account being determined in accordance with the directions given in 2-5.

For calculations which take account of the operating time the conventional hours determined in 2-1.3.2 should be used.

APPENDIX A - 4-1.3

DETERMINATION OF PERMISSIBLE STRESSES IN MECHANISM COMPONENTS SUBJECTED TO FATIGUE

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 stress concentrations or "notch effects", which increase the actual stresses in the part. For a given section the load must therefore be reduced to maintain the actual stress (including stress concentration effect) below any allowable value. This is allowed by introducing factors k_s, k_d, k_u, k_{uc} (refer to 4-1.3.3). These factors are respectively all greater than or equal to unity, by the product of which the endurance limit for a polished specimen is devided.

Guidelines concerning the determination of these coefficients are set out below :

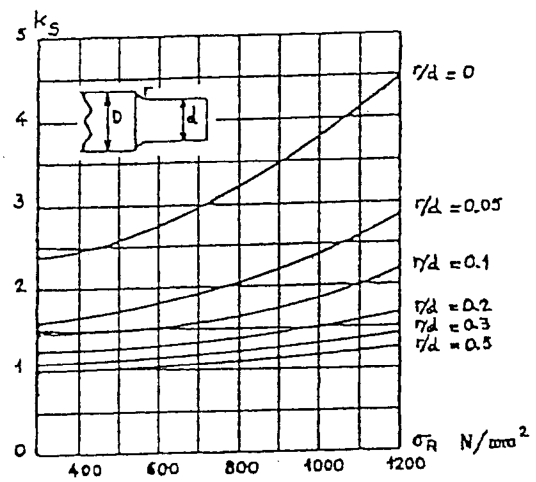
A. DETERMINATION OF k_s

This coefficient specifies the stress concentrations caused by changes of section at radii, annular grooves, transverse holes keyways and other methods of securing hubs.

Figures A.4-1.3.1.a and b give the values of the shape coefficient k_s, as a function of the ultimate strength of the material, valid for a diameter D of 10 mm.

- The curves A.4-1.3.1.a give the coefficient k_s for a change of section of ratio D/d = 2, with a correction table T.A.4-1.3.1 for other values of D/d.

Figure A.4-1.3.1.a
SHAPE COEFFICIENT k_s (DIAMETER D = 10 mm)
CHANGE OF SECTION D/d = 2



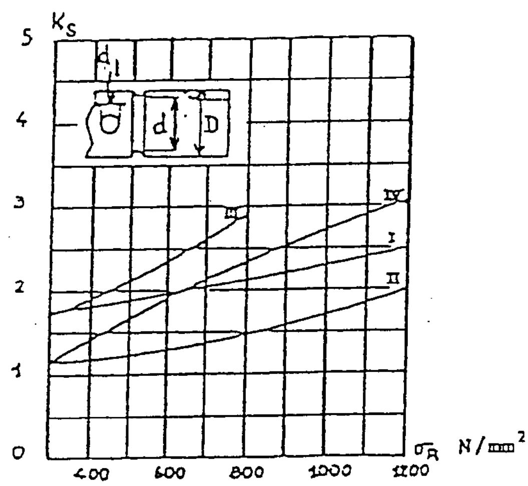
For other values of D/d read k_s from the curve (r/d) + q with the following values for q :

Table T.A.4-1.3.1
CORRECTION FACTORS q FOR D/d ≤ 2

D/d	1.05	1.1	1.2	1.3	1.4	1.6	2
q	0.13	0.1	0.07	0.052	0.04	0.022	0

- The A.4-1.3.1.b curves give, for guidance some values of k_s for holes, annular grooves keyways and press-fitted hubs.

Figure A.4-1.3.1.b
SHAPE COEFFICIENT k_s (DIAMETER D = 10 mm)
HOLE, ANNULAR GROOVE, KEYWAY, PRESS-FITTED HUB



Curve I : transverse hole $d_1 = 0.175$ d
II : annular groove : depth 1 mm
III : keyed hub
IV : press-fitted hub

B. DETERMINATION OF SIZE COEFFICIENT k_d

For diameters greater than 10 mm the stress concentration effect increases and this increase is allowed for by introducing the size coefficient k_d.

The values of the coefficient k_d are given in table T.A.4-1.3.2 for values of d from 10 mm to 400 mm.

Table T.A.4-1.3.2
VALUES OF k_d

d mm	10	20	30	50	100	200	400
k_d	1	1.1	1.25	1.45	1.65	1.75	1.8

C. DETERMINATION OF SURFACE CONDITION COEFFICIENTS

1. INITIAL SURFACE FINISH DUE TO MANUFACTURING PROCESSES - COEFFICIENT k_u

Experience shows that parts produced with a rough finish have a lower endurance limit than carefully polished parts.

This is allowed for by applying a machining coefficient k_u given in figure A.4-1.3.2 for ground or finely polished surfaces, for rough machined parts and for forged and rolled sections.

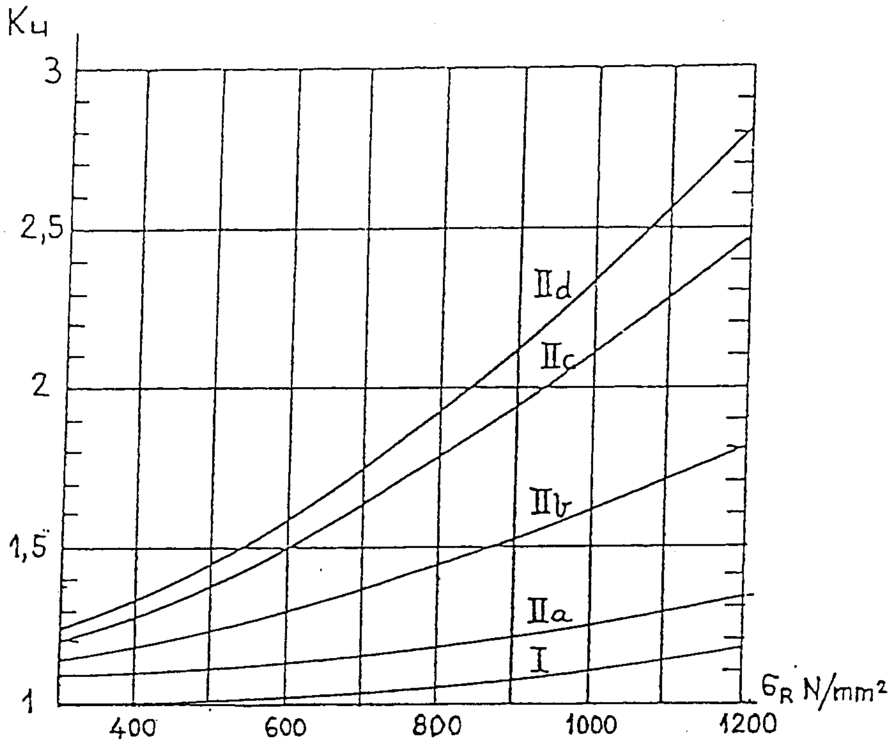
2. INFLUENCE OF CORROSION

Corrosion can have a very appreciable effect on the endurance limit of steels ; this is allowed for by applying a correction to factor k_u which becomes k_{uc}.

In the case of particular hazard or corrosion, the values of k_{uc} to be used are those given for k_u for forged or rolled pieces and this without taking into account the method of machining.

Obviously, the used of the factor k_{uc} does not expect to take into account the effects of the possible corrosion on the time, as for instance the reduction of the size of the structural components, effects which must be taken into account separately. The factor k_{uc} only take into consideration the fact that the surface condition of the component subject to corrosion will be rapidly modified.

Figure A.4-1.3.2
VALUES OF THE MACHINING COEFFICIENT k_u (CORROSION COEFFICIENT k_{uc})

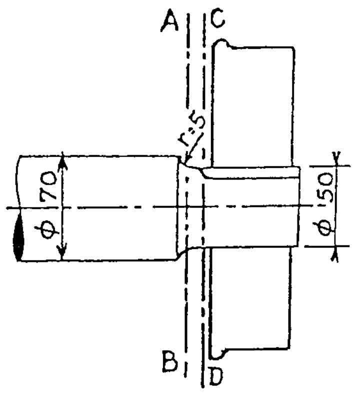


Values of k_u

Curve I : surface ground or finely machined : $6.3 < Rt \leq 16$ µm
IIa : surface machined : $16 < Rt \leq 63$ µm
IIb : surface rough machined : $63 < Rt \leq 160$ µm
IIc : properly forged or rolled part
IId : rough forged or rolled part or properly cast (sand mould)

Note : In the case of hazard of corrosion, k_{uc} must be chosen in curves IIc or IId.

Example of application



Shaft in A-550 steel with change of section. Diameter D = 70 mm and d = 50 mm with transition radius r = 5 mm. Turned on lathe, with keyed wheel.

The component will be deemed to be classified in group E4.

We shall assume alternating loading ($\kappa = -1$) and the shaft to be of A-550 steel (minimum $\sigma_R = 550$ N/mm²). We can therefore adopt :

$$\sigma_{bw} = 0.5 \,.\, 550 = 275 \text{ N/mm}^2$$

Section A-B

$$D/d = 70/50 = 1.4$$
$$r/d = 5/50 = 0.1$$

Determination of k_s (shape)

For D/d = 1.4, we have :

$q = 0.04$ (table T.A.4-1.3.1)

From the curve (r/d) + q = 0.1 + 0.04 = 0.14, we find by interpolation :

$k_s = 1.4$ (figure A.4-1.3.1.a)

Determination of k_d (size)

For d = 50, we have :

$k_d = 1.45$ (table T.A.4-1.3.2)

Determination of k_u (machining)

For a part turned on a lathe, we have :

$k_u = 1.15$ (figure A.4-1.3.2 curve II)

From the foregoing values, we derive :

$$\sigma_{wk} = \frac{275}{1.4 \,.\, 1.45 \,.\, 1.15} = 117.8 \text{ N/mm}^2$$

For $\kappa = -1$, we have :

$\sigma_d = \sigma_{wk}$ $= 117.8$ N/mm²

$$c = \frac{\log (2\,000\,000 \,/\, 8\,000)}{\log (550 \,/\, 117.8)} = 3.58$$

For group E4, the fatigue strength is therefore :

$$\sigma_k = \sigma_d \cdot 2^{\frac{8-4}{c}}$$

$$= 117.8 \cdot 2^{(4\,/\,3.58)} = 255.4 \text{ N/mm}^2$$

The safety coefficient νk is given by :

$$\nu_k = 3.2^{\,1/c} = 3.2^{1/3.58} = 1.38$$

The permissible stress σ_{af} is therefore :

$$\sigma_{af} = \frac{255.4}{1.38} = 184.6 \text{ N/mm}^2$$

<u>Section C-D</u>

We have :

$k_s = 2.2$ (figure A.4-1.3.1.b)

$k_d = 1.45$ (same value as above)

$k_u = 1.15$ (same value as above)

Hence :

$$\sigma_{wk} = \frac{275}{2.2 \cdot 1.45 \cdot 1.15} = 75.0 \text{ N/mm}^2$$

$\sigma_d = \sigma_{wk}$ $= 75.0$ N/mm²

$$c = \frac{\log (2\,000\,000 \,/\, 8\,000)}{\log (550 \,/\, 75)} = 2.77$$

$$\sigma_k = 75 \cdot 2^{(4 / 2.77)} \qquad = 204.0 \text{ N/mm}^2$$

$$\nu_k = 3.2^{1 / 2.77} \qquad = 1.52$$

$$\sigma_{af} = \frac{204}{1.52} \qquad = 134 \text{ N/mm}^2$$

ANNEXE

Comparison of steel grades as per the European standard EN 10083-1, the standard ISO 683.1 and other national standards previously issued

EN 10 083-1	ISO 683.1 1987 (1)	Germany		Finland	Unitec Kingdom	France	Sweden SS-Stahl	Spain	
		Abbreviated designation	No. of material					Abbreviated designation	No. of material
2 C 22	-	(Ck 22)	(1.11.51)	-	(070M20)	[XC 18]	-	-	-
2 C 22	-	(Cm 22)	(1.1149)	-	-	[XC18 u]	-	-	-
2 C 25	(C 25 E4)	Ck 25	1.1158	-	(070M26)	[XC 25]	-	C25k	F1120
3 C 25	(C 25 M2)	Cm 25	1.1163	-	-	[XC 25 u]	-	C25k-1	F1125 (1)
2 C 30	(C 30 E4)	Ck 30	1.1178	-	(080M30)	[XC 32]	-	-	-
3 C 30	(C 30 M2)	Cm 30	1.1179	-	-	[XC 32 u]	-	-	-
2 C 35	(C 35 E4)	Ck 35	1.1181	C 35	(080M36)	[XC 38 H 1]	1572	C35k	F1130
3 C 35	(C 35 M2)	Cm 35	1.1180	-	-	[XC 38 H 1 u]	-	C 35k-1	FF1135 (1)
2 C 40	(C 40 E4)	Ck 40	1.1186	-	(080M40)	[XC 42 H 1]	-	-	-
3 C 40	(C 40 M2)	Cm 40	1.1189	-	-	[XC 42 H 1 u)	-	-	-
2 C 45	(C 45 E4)	Ck 45	1.1191	C 45	(080M46)	[XC 48 H 1]	1672	C45k	F1140
3 C 45	(C 45 M2)	Cm 45	1.1201	-	-	[XC 48 H 1 u)	-	C45k-1	F1145 (1)
2 C 50	(C 50 E4)	Ck 50	1.1206	-	(080M50)	-	1674	-	-
3 C 50	(C 50 M2)	Cm 50	1.1241	-	-	-	-	-	-
2 C 55	(C 55 E4)	Ck 55	1.1203	-	(070M55)	[XC 55 H 1]	-	C55k	F1150
3 C 55	(C 55 M2)	Cm 55	1.1209	-	-	[XC 55 H 1 u]	-	C55k-1	F1155(1)
2 C 60	(C 60 E4)	Cj 60	1.1221	-	(070M60)	-	-	-	-
3 C 60	(C 60 M2)	Cm 60	1.1223	-	(070M60)	-	-	-	-
28 Mn 6	(28 Mn 6)	28 Mn 6	1.1170	-	(150M19)	-	-	-	-
38 Cr 2	-	38 Cr 2	1.17003	-	-	(38 C 2)	-	-	-
38 CrS 2	-	38 CrS 2	1.7023	-	-	(38 C 2 u)	-	-	-
46 Cr 2	-	46 Cr 2	1.7006	-	-	-	-	-	-
46 CrS 2	-	46 CrS 2	1.7025	-	-	-	-	-	-
34 Cr 4	34 Cr 4	34 Cr 4	1.7033	-	(530M32)	(32 C 4)	-	-	-
34 CrS 4	34 CrS 4	34 CrS 4	1.7037	-	-	(32 C 4 u)	-	-	-
37 Cr 4	37 Cr 4	37 Cr 4	1.7034	-	(530M36)	(38 C 4)	-	38Cr4	F1201
37 CrS 4	37 CrS 4	36 CrS 4	1.7038	-	-	(38 C 4 u)	-	38Cr4-1	F1206 (1)
41 Cr 4	41 Cr 4	41 Cr 4	1.7035	-	(530M40)	42 C 4	-	42Cr4	F1202
41 CrS 4	41 CrS 4	41 CrS 4	1.7039	-	-	42 C 4 u	2245	42 Cr4-1	F1207 (1)
25 CrMo 4	25 CrMo 4	25 CrMo 4	1.7218	25 CrMo 4	(708M25)	25 CD 4	2225	-	-
25 CrMoS 4	25 CrMoS 4	25 CrMoS 4	1.7213	-	-	25 CD 4 u	-	-	-
34 CrMo 4	34 CrMo 4	34 CrMo 4	1.7220	34 CrMo 4	(708M32)	(34 CD 4)	2234	-	-
34 CrMoS 4	34 CrMoS 4	34 CrMoS 4	1.7226	-	-	(34 CD 3 u)	-	-	-
42 CrMo 4	42 CrMo 4	42 CrMo 4	1.7225	42 CrMo 4	(708M40)	42 CD 4	2244	40 CrMo 4	F1252
42 CrMoS 4	42 CrMoS 4	42 CrMoS 4	1.7227	-	-	42 CD 4 u	-	40 CrMo4-1	F1257 (1)
50 CrMo 4	50 CrMo 4	50 CrMo 4	1.7228	-	(708M50)	-	-	-	-
36 CrNiMo 4	36 CrNiMo 4	36 CrNiMo 4	1.6511	-	(817M37)	-	-	-	-
34 CrNiMo 6	(34 CrNiMo 6)	(34 CrNiMo 6)	(1.6582)	34 CrNiMo 6	(817M40)	-	2541	-	
30 CrNiMo 8	(31 CrNiMo 8)	30 CrNiMo 8	1.6580	-	(823M30)	30 CND 8	-	-	-
36 NiCrMo 16	-	-	-	-	(835M30)	35 CND 16	-	-	-
51 CrV 4	(51 CrV 4)	50 CrV 4	1.815	-	[735A50]	(50 CV 4)	-	51 CrV 4	F1430

(1) The use of steel grade in plain brackets means that its chemical composition is only slightly different from that of EN 10083-1. The indication of a steel grade in square brackets means that its chemical composition differs to a greater extent from that of EN 10083-1. Grades which are neither in plain nor square brackets have virtually the same chemical composition as that of EN 10083-1.

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LIST OF SOME WORKS DEALING WITH FATIGUE PROBLEMS

(1) Niemann, G. "Maschinenelemente"

Band 1
Springer Verlag - Berlin/Göttingen/Heidelberg - 1975

(2) Niemann, G. "Maschinenelemente"

Band 2
Springer Verlag - Berlin/Göttingen/Heidelberg - 1983

(3) Decker, K.-H. "Maschinenelemente"

Carl Hanser Verlag - München - 1982

(4) "Metal Fatigue" by J.A. Pope & Ph. D., D.Sc - Wh. Sch. I Mech.

E. Chapmann & Hall Ltd., 37 Essex street, London WC2

(5) "La fatigue des métaux" by R. Cazaud - Ingénieur Cnam - Doctor of the University of Paris - Lecturer at the Higher Institute for Mechanical Engineering Materials - Consulting engineer

Dunod - 92 rue Bonaparte - Paris

(6) "Fatigue of metals and structures" by H.J. Grover, s.a. Gordon, RL Jackson

Thames & Hudson, London

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Chapter 5

SAFETY REQUIREMENTS

The Design rules developed particularly in chapters 3 and 4 of this standard allow a mobile continuous bulk handling machine to be designed and dimensioned correctly thereby ensuring the safety of the machine itself against the various causes of failure (breaking of major parts, loss of stability, etc).

In addition to these aspects, careful consideration should be given to the safety of persons who will have to work on the machine for its driving, operation and maintenance. This is covered by a number of provisions which, while they often do not affect the strength of the machine, are nevertheless indispensable for safe operation of that machine.

For the health and safety requirements for mobile continuous bulk handling equipment, in Europe, one shall refer to the following Directives of the Council of the European Communities :
- Machinery directive No. 89/392/EEC of 14th June 1989 ; modified on 20th June 1991 (91/368/EEC), modified on 14th June 1993 (93/44/EEC) and on 22nd June 1993 (93/68/EEC).
- Low voltage directive No. 73/23/EEC of 19th February 1973 modified on 22nd July (93/68/EEC).
- Electromagnetic compatibility directive No. 89/336/EEC of the 3rd May 1989 modified on 28th April 1992 (92/31/EEC) and on 22nd July 1993 (93/68/EEC).

For particular applications one can refer to :
- Equipment and Protective systems intended for used in potentillay explosive atmosphere directive No. 94/91/EEC of the 23rd March 1994.
- Single pressure vessel directive No. 87/404/EEC of 25th June 1987 modified on 17th September 1990 (90/488/EEC) and on 22nd July 1993 (93/68/EEC).

On the basis of the Machinery Directive, the Technical Committee 148 of CEN "Continuous handling equipment and systems - safety" has prepared a number of standards applicable to families of products involved in continuous handling. These standards are the following :

prEN 620 (WG 1)	**Continuous handling equipment and systems** Safety requirements for fixed belt conveyors for bulk materials
prEN 619 (WG 2)	**Continuous handling equipment and systems** Safety requirements for equipment for mechanical handling of unit loads
prEN 618 (WG 3)	**Continuous handling equipment and systems** Safety requirements for equipment for mechanical handling of bulk materials except fixed belt conveyors
prEN 617 (WG 4)	**Continuous handling equipment and systems** Safety requirements for equipment for the storage of bulk materials in silos, bunkers, bins and hoppers
prEN 741 (WG 5)	**Continuous handling equipment and systems** Safety requirements for systems and their components for pneumatic handling of bulk materials

In Europe, it is therefore desirable for the design of a mobile continuous bulk handling machine to refer to the standard prEN 618 (WG 3 - mobile mechanical bulk handling machines).

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In the most common case, when the machine includes one or more belt conveyors, there is need to refer also to the standard prEN 620 (WG 1 - equipment for mechanical handling of both unit loads and bulk materials).

It should be noted that the European type C* safety standards mentioned above include numerous references to more general type A, B1, B2 standards* and it may be necessary for a particular type of handling equipment to produce a collection of the parts of standards which are appropriate to it.

* *Safety standards are classified as follows :*

Type A standards *(basic safety standards) giving basic concepts, principles for design, and general in the same or a similar manner for all machinery.*

Type B standards *(group safety standards) dealing with one safety aspect or one type of safety related device in the same or a similar manner for a range of machinery*
- *type B1 standards on particular safety aspects (e.g. safety distances, surface temperature, noise)*
- *type B2 standards on safety related devices (e.g. two-hand controls, interlocking devices, pressure sensitive devices, guards).*

Type C standards *(machine safety standards) giving detailed safety requirements for a particular type of machinery or group of machines defined in the scope of the standard.*

(extract from the standard EN 292-1 "Safety of machinery - Basic concepts, general principles for design)

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Chapter 6

TESTS AND TOLERANCES

CHAPTER 6

TESTS AND TOLERANCES

CONTENTS

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6-1 TESTS

6-1.1 GENERAL

The variety of mechanical handling equipment available and the diversity of its application precludes the specification of any single, universal acceptance test.

For this reason, it is essential, from the earliest pre-tender enquiries if possible, that the end user, his engineer and the manufacturer/supplier of the equipment are all anticipating the same outcome.

Consequently, before a contract is awarded, the end user of the equipment should define and agree with the supplier, the criteria that have to be met before the equipment can be handed over.

The specification for these acceptance tests should be as straightforward and as simple as possible, often involving a "representative" test cycle to verify the performance of a single item of equipment in isolation from the rest of the plant.

The conditions for these acceptance tests should also be defined, for example, the following data should be specified :

- Weather conditions (wind, temperature, snow...),
- Whether the tests will be carried out in the daytime or at night,
- Characteristics of the materials to be handled (i.e. moisture, grain size, bulk density...),
- Flow rates (mass flow, volume flow) to be achieved for the various bulk materials, and the corresponding duration for each flow rate,
- Environmental conditions, in particular any restrictions affecting the movement of the tested appliances. For instance, information should be given on the type of ship on which a loader or unloader will operate, the type of stockpile (cross section, length) on which a bucket-wheel reclaimer or a stacker will operate, etc,
- Methods for measuring handled quantities, actual operating times, etc,
- Qualification of the operator(s),
- ...

6-1.2 TYPICAL ACCEPTANCE TESTS

Typical acceptance tests might be :

a) Demonstrate that a stockyard equipment is able to store or to reclaim the anticipated material quantities in the required number of piles.

   This requires to agree especially on the general arrangement of the piles and the travel distances of the mobile machines.

b) Demonstrate that the equipment can handle the working load(s) in normal operating conditions.

   This requires an agreed definition of the working load(s) and how it will be measured

It should be noted that for bulk materials which density is variable, it could be necessary to define for example one tonnage capacity and one corresponding volumetric capacity, the performance being considered acceptable as soon as one of these capacities (tonnage or volume) is reached.

c) Demonstrate that the equipment will improve the material blend by the required amount.

This requires an agreed sampling technique and method of analysis.

d) Demonstrate that the plant can load/unload a ship of a given size and type in the required time.

"Whole plant" tests of this nature are difficult to achieve especially where parts of the plant are supplied by different manufacturers. The management and organisation of the plant also has a significant effect and this is often outside the supplier's control.

e) Demonstrate that the equipment can handle a nominated surge load without excess of spillage.

This requires a definition of the surge load, both in magnitude and duration, also an agreement on what is "acceptable" spillage.

f) Demonstrate that the system can restart under load after an emergency (non sequence) stop at a normal working throughput.

This requires an agreed definition of the corresponding working load and how it will be measured.

g) Demonstrate that the continuous output from the equipment does not fluctuate by more than an agreed tolerance.

The fluctuation tolerance should be quoted for a corresponding throughput and a method of assessment agreed.

6-1.3 DEFINITIONS

.1 Nominal or design rate : it is the throughput received or delivered by the machine under normal working conditions.

.2 Peak instantaneous rate : it is the maximum practical throughput used for sizing equipment where flow surges are expected. It is the maximum rate that the machine will meet in operation.

For reclaiming equipment this is normally defined by the manufacturer. In this case, the rate depends on the flow characteristics of the equipment and the required maximum average rate.

For equipment which receives material from other up stream plant (e.g. a stacker), the peak instantaneous rate is normally given by the customer.

.3 Maximum average rate / 1 hour average rate : it is the maximum operational throughput for short-term*, uninterrupted operation (Excludes such as stockpile ends, machine re-positioning, hatch changes on ship loaders, etc).

This rate is often used for machine acceptance tests.

* typical time scale, 1 hour, may vary slightly.

.4 Daily average rate / Shift average rate / Through-ship rate : "whole plant" rate, greatly affected by plant management practices, availability and location of materials, up and downstream restrictions, stockpile layout, hold size and efficiency of clean-up aids for ship unloaders, operator skills, etc.

It is sometimes required for plant acceptance tests but difficulties can be encountered where the supplier does not have full control of the plant and incoming and outgoing materials.

The relationship between 1, 2 and 3 above is largely dependent on equipment type and it's method of operation (especially for reclaiming equipment), characteristics of material or materials, method of storage, weather conditions (low temperature, for instance), qualification of drivers and operators, etc, and must be established from experience.

Note : In the case of multi-material machines, it is necessary to agree on the performance (both mass and volumetric capacities) for each material, especially excluding, if necessary, the possibility of using the machine to handle the maximum volumetric capacity of material of the heavier bulk density.

In this case, it is obviously necessary to install efficient "throughput limiting devices" either on the machine itself (reclaiming machine) or on an other part of the plant, upstream of the machine.

6-1.4 CATEGORIES OF TESTS

In general, acceptance tests fall into three categories :

1) Verification before starting and no load functional testing,

2) Operational tests at normal working loads,

3) Overload/surge tests, only where appropriate.

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6-1.4.1 NO LOAD FUNCTIONAL TESTING

Before no load functional testing begins, it has to be verified that :

1) All relevant adjustments and settings have been correctly made.

2) Conveyor belts have been adequately jointed, tensioned and correctly aligned and that belt cleaners, scrapers and guards have been properly installed.

3) All relevant limit switches, level probes and interlocks, sequencing and control systems have been separately verified.

4) Special attention has been paid to the verification and adjustment of force, torque, pressure limiting devices as well as the stability of the machine in all conditions.

5) All ancillary equipment (such as cleaning devices, belt weighers, sampling equipment, metal detectors and separators, dust extractors and vibratory feeders, normally supplied by specialist sub-contractors) have been correctly installed and no-load commissioned.

6) Warning notices have been erected where necessary.

7) Where operational interlocks have to be bypassed for the duration of the test, all operators and test personnel have been informed.

The no-load functional test should include all modes of operation : manual and automatic operation where appropriate and both normal and emergency stops under all operating conditions.

6-1.4.2 TESTING UNDER NOMINAL LOAD

The commencement of operational tests assumes, among others :

1) The satisfactory completion of relevant no-load functional tests.

2) The availability of an agreed minimum quantity of suitable material with acceptable properties such as moisture content, size and bulk density. The responsability for the supply of this material and its delivery must be agreed at an early stage.

3) An adequate liaison with other affected parties and suppliers on the site.

4) Representative operating conditions.

5) Where ancillary equipment such as cleaning devices, belt weighers and sampling equipment are to be used to assess the performance of the plant, these must have been separately commissioned on

load and certified before the main tests commence.

At an early stage in the contract, the end user and supplier of the equipment should agree an operational test schedule, the details of what schedule should contain and in what form any test results should be presented.

6-1.4.3 OVERLOAD TESTING

Where surge load or overload tests are required, the supplier and user must agree on method of quantifying the transient loads involved and where some spillage or carry-over is anticipated, determine beforehand, what will be acceptable.

6-1.5 LIST OF TYPICAL CHECKS AND TESTS

Listed below are some of the items that may appear in an operational test schedule. This is by no means comprehensive, nor does the inclusion of an item on this list imply that this must be included. Items should only be included where these are particularly relevant to the equipment involved.

- Verification of safety for personnel,
- Verification of working clearances on adjacent buildings and other machines,
- Drive speeds and absorbed power.

Note : Records of ambient temperatures, wind speeds, etc, during test may be required so that for instance maximum power requirements, maximum forces, etc, can be got by back-calculation.

- Dust,
- Noise,
- Stockpile capacity and size, live and dead storage areas,
- Safety devices : force, torque, pressure limiting devices, ...,
- Carry-over and spillage,
- Level probes, limit switches, sequencing and control systems,
- Operation of chutes and diverter flaps, etc,
- Start under nominal load,
- Emergency and sequence stops,
- Achievement of nominal or design rating.

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6-2 TOLERANCES FOR TRAVELLING DEVICES AND TRACKS

6-2.1 GENERAL

The use of these design rules pre-supposes that the tolerances specified hereafter for travelling devices and tracks shall be maintained.

These tolerances apply unless other conditions have been agreed with the user and take no account of elastic deformation during operation.

Section 6-2.2 defines the manufacturing tolerances for travelling devices of machines.

Section 6-2.3 specifies the tolerances for tracks on rigid support.

Section 6-2.4 specifies the tolerances for tracks on ballast.

Under all circumstances, sufficient lateral play between rail and guiding devices must be provided to accomodate the maximum allowable variation in machine span and track gauge.

It is then essential to ensure that tolerances for travelling devices and tolerances for track are compatible.

Where the design of a machine is such that the following tolerance categories for tracks do not obviously apply, the supplier of the machine must clearly indicate the initial tolerances to be followed for the tracks and the maximum deviation allowed before resetting of tracks is required.

6-2.1.1 MEASURING PROCEDURE

The measuring procedures used must be selected to provide the required accuracy and repeatability over the lengths involved.

When measuring tapes are used, calibrated steel tapes are recommended. The readings obtained should be corrected for the sag in the tape as well as for the difference between the ambient and reference temperatures. All related measurements should be taken with the same tape and tension force.

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6-2.2 MANUFACTURING TOLERANCES FOR TRAVELLING DEVICES

6-2.2.1 DISTANCE FROM CENTERLINE TO CENTERLINE (MACHINE SPAN)

The greatest divergence Δs of the machine span s from the nominal drawing dimension must not exceed the following values :

for $s \leq 15$ m : $\Delta s = \pm 2$ mm
for $s > 15$ m : $\Delta s = \pm [\, 2 + 0.15 \,.\, (s-15) \,]$ mm (<u>max. ± 15 mm</u>)

(s is to be expressed in m)

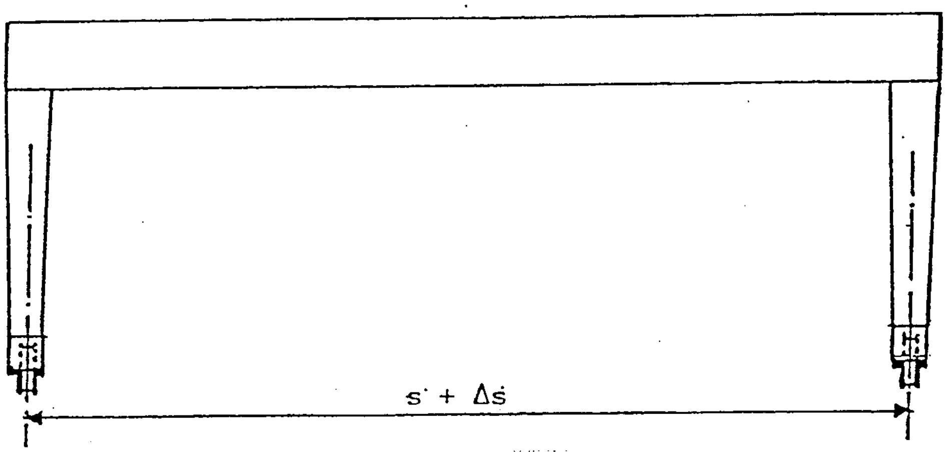

Figure 6-2.2.1

6-2.2.2 SAG OF GANTRY

Gantry type structures which are designed to carry a moving trolley (as tripper for instance) and are freely supported at their ends must have no sag, under dead loads. This means that the track for the trolley, without the trolley and in an unloaded condition, must have no downward deviation from the horizontal.

6-2.2.3 INCLINATION OF THE WHEELS

For all machines, where the transversal axis of the top of the rail is flat, the inclination of the wheel axis from the horizontal, for the travelling devices in an unloaded condition, must be between + 0.2 % and - 0.05 % (see figure 6-2.2.3).

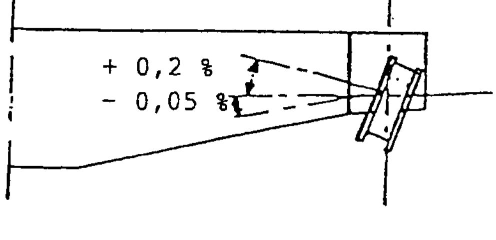

Figure 6-2.2.3

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6-2.2.4 TROLLEY RAIL CENTER DISTANCE

The trolley rail center distance must not differ from the nominal dimension s by more than ± 3 mm (see figure 6-2.2.4).

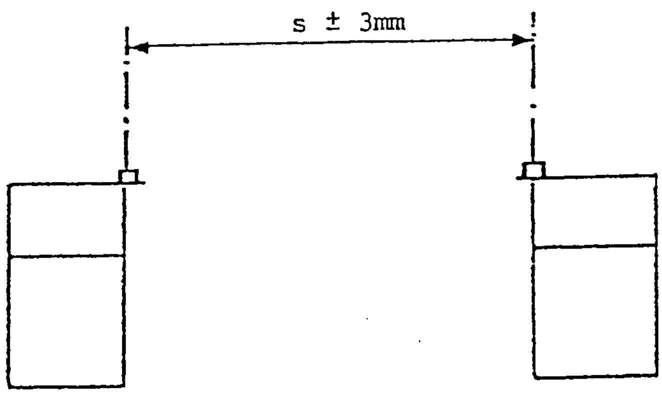


Figure 6-2.2.4

6-2.2.5 LEVEL OF RAILS FOR TROLLEY

In a plane perpendicular to the travel direction of the trolley, the difference in height of two opposite points of the trolley track shall not exceed 0.15 % of the trolley rail centre distance s, with a maximum of 10 mm (see figure 6-2.2.5).

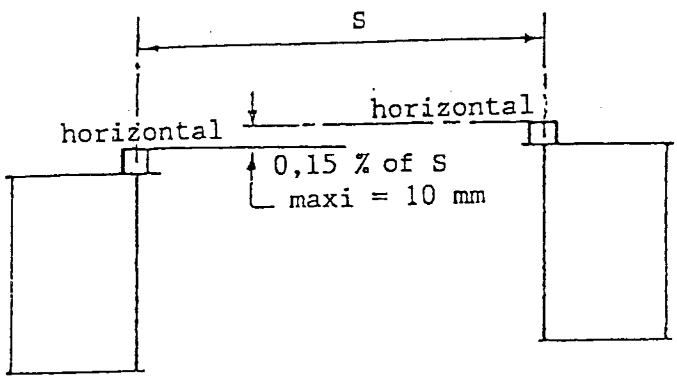


Figure 6-2.2.5

6-2.2.6 HEIGHT TOLERANCE ON FOUR POINTS

Trolley rails shall be laid in such a way that the running surface is horizontal and that the greatest unevenness of the bearing surface is no more than ± 3 mm for rail centres up to 3 m and no more than ± 0.1 % of the trolley wheel centre distance if it exceeds 3 m (see figure 6-2.2.6).

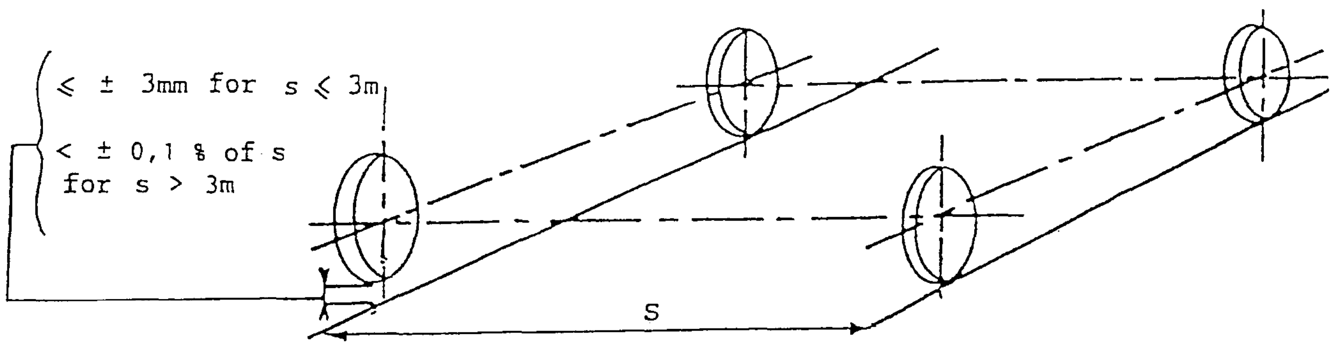


Figure 6-2.2.6

6-2.2.7 RAIL ON GIRDER WEB

The vertical axis of the trolley rail must not diverge from the vertical axis of the rail girder web by more than half the thickness of the rail girder web (see figure 6-2.2.7).

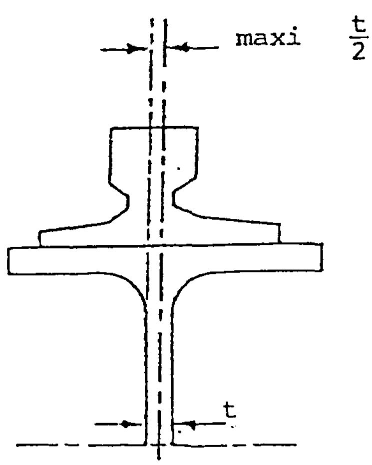


Figure 6-2.2.7

6-2.2.8 TROLLEY RAILS STRAIGHTNESS

The axes of the trolley rails must not diverge from their theoretical axis by more than ± 1.0 mm in a rail length of 2 m (see figure 6-2.2.8). There should be no misalignments at rail joints.

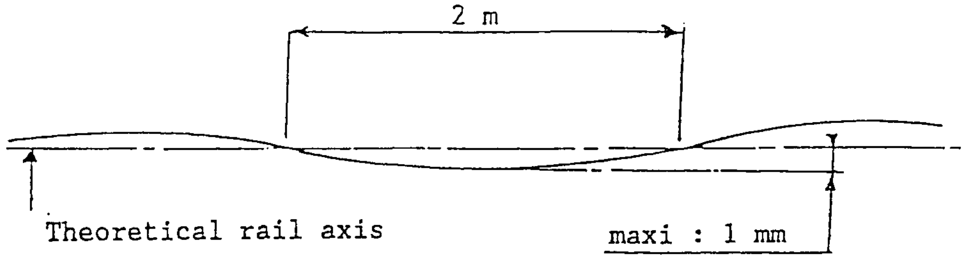


Figure 6-2.2.8

6-2.2.9 ANGULAR ALIGNMENT OF WHEELS

The tangent of the angles between the axis of the wheel bores and the theoretical axis must not be greater than ± 0.04 % in the horizontal plane (see figure 6-2.2.9).

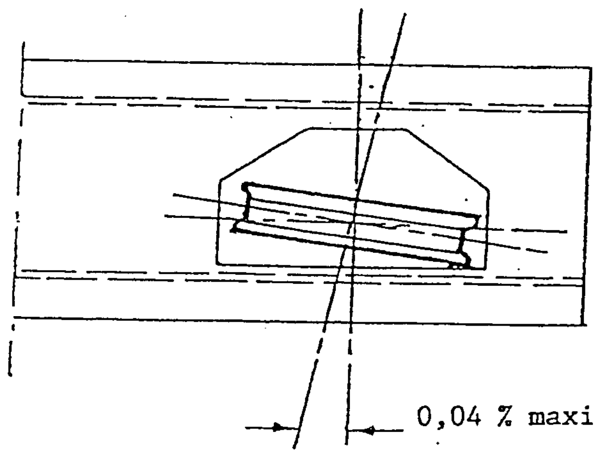


Figure 6-2.2.9

6-2.2.10 LEVEL OF WHEEL AXLES

The axle bores of wheels opposite to each other at each side of the tracks, or, if wheels are mounted in bogies the axis of the bogie pins shall have an alignment divergence in the vertical plane, less than 0.15 %, of the wheel centre distance (maximum 2 mm - see figure 6-2.2.10).

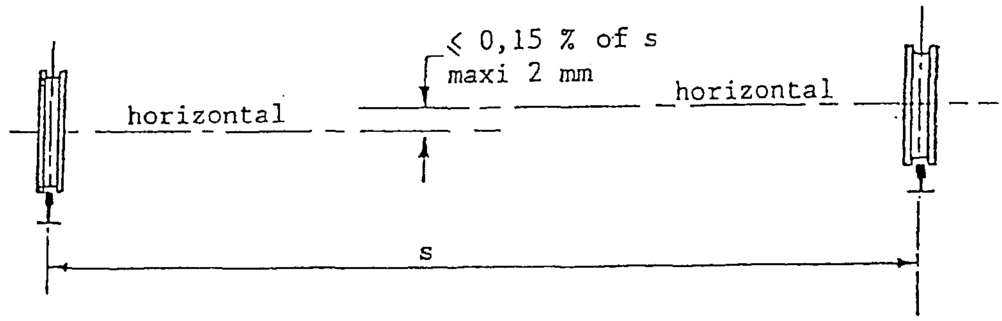


Figure 6-2.2.10

6-2.2.11 TRANSVERSE ALIGNMENT OF WHEELS

The centre planes of wheels rolling on a common rail must not diverge by more than ± 1 mm from the theoretical rail axis (see figure 6-2.2.11).

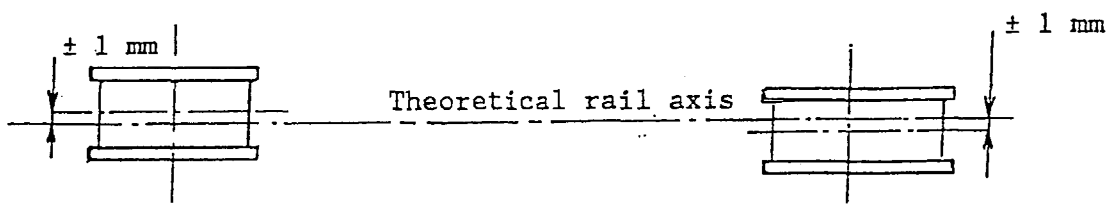


Figure 6-2.2.11

For bushed wheels, the above tolerances apply with the wheel in a central position between the contact surfaces at either side of the wheel.

6-2.2.12 HORIZONTAL GUIDE ROLLERS

If horizontal guide rollers are used, the centre of the distance between guide rollers at one corner must not deviate by more than ± 1 mm from the theoretical axis of the rail (see figure 6-2.2.12).

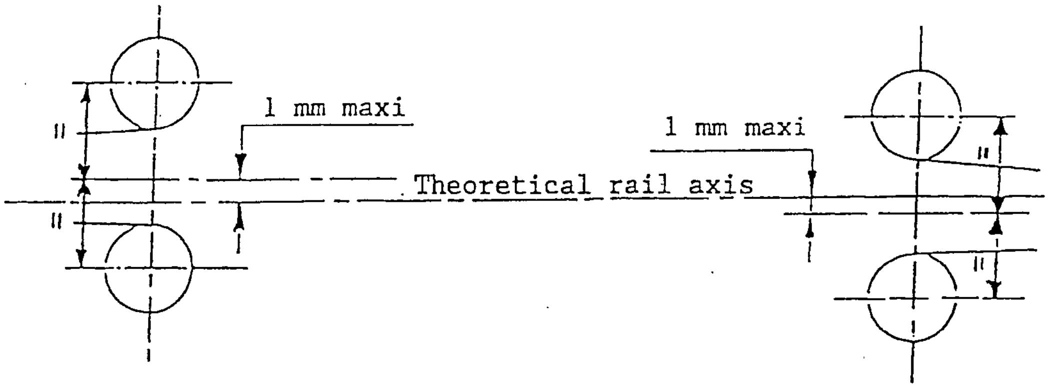


Figure 6-2.2.12

6-2.2.13 WHEEL DIAMETER

The diametral tolerance of driving wheels should correspond to the ISO tolerance classification h9. If runner wheel speeds are synchronised electrically or where two wheels in a bogie are driven from a common pinion, tighter tolerances may be required.

These tighter tolerances apply also to non driven wheels where the wheels are interchangeable.

Wheel tolerances are more critical and may have to be tighter on machines which are constantly moving and not operating in a "step-advance" mode.

6-2.3 TOLERANCES FOR TRACKS ON RIGID SUPPORT

6-2.3.1 ALLOWABLE DEVIATION BEFORE RE-ARRANGEMENT

The tolerances specified in this section apply to new fixed ground-bearing tracks.

Tolerances for machine mounted tracks are specified in section 6-2.2.

If, in the course of use, the admissible deviations for the new installation are exceeded by 5 mm or 20 %, the track must be re-arranged.

In certain circumstances, it may be necessary to re-arrange the track before the 20 % limit is reached, if the travelling behaviour is noticeably deteriorating.

6-2.3.2 TRACK GAUGE TOLERANCES

The greatest admissible divergence Δs from the nominal span is :

for $s < 15$ m : $\Delta s = \pm 3$ mm

for $s > 15$ m : $\Delta s = \pm [3 + 0.25 \cdot (s-15)]$ mm (with a maximum of $\pm$ 25 mm)

(s is expressed in metres)

If the machine is guided on one rail only, or if the machine has a pivoted support or is of high elasticity, the tolerance Δs may be increased to three times the above value but must not exceed 25 mm.

6-2.3.3 SAG OF RAILS

It is assumed that under normal working conditions, the deflection of both rail tracks under load is approximately equal and does not significantly effect the performance of the machine.

6-2.3.4 OVERALL RAIL TOLERANCES

. Straightness
For each rail, the overall centre-line measured at the running surface, shall not deviate by more than $\pm$ 10 mm from the theoretical line in both the horizontal and vertical planes.

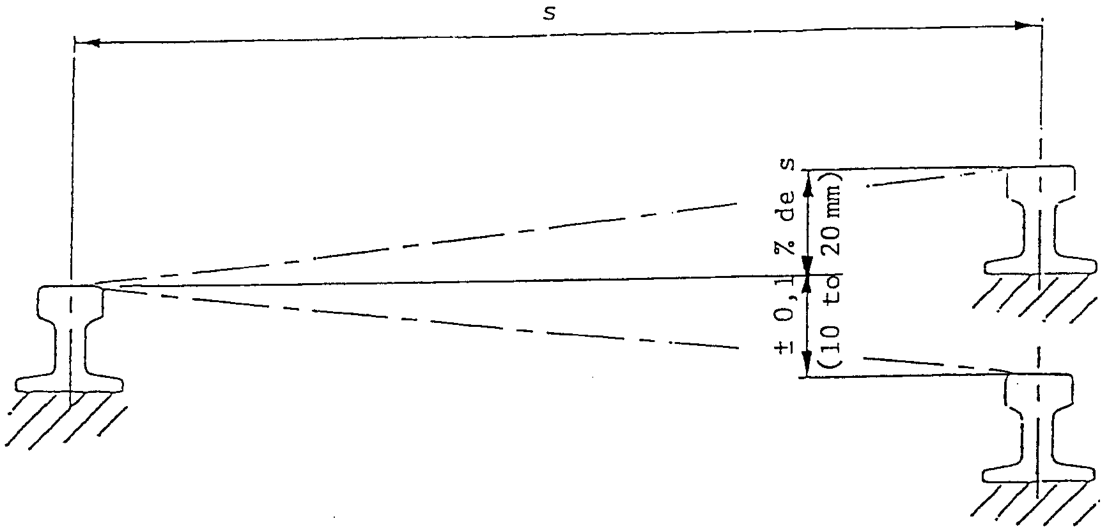
For machines guided on one rail only, the requirement for lateral straightness of the non-guiding rail only may be lowered, in agreement with the manufacturer of the machine.

. Rail joints
It is recommended that welded rail joints are used.

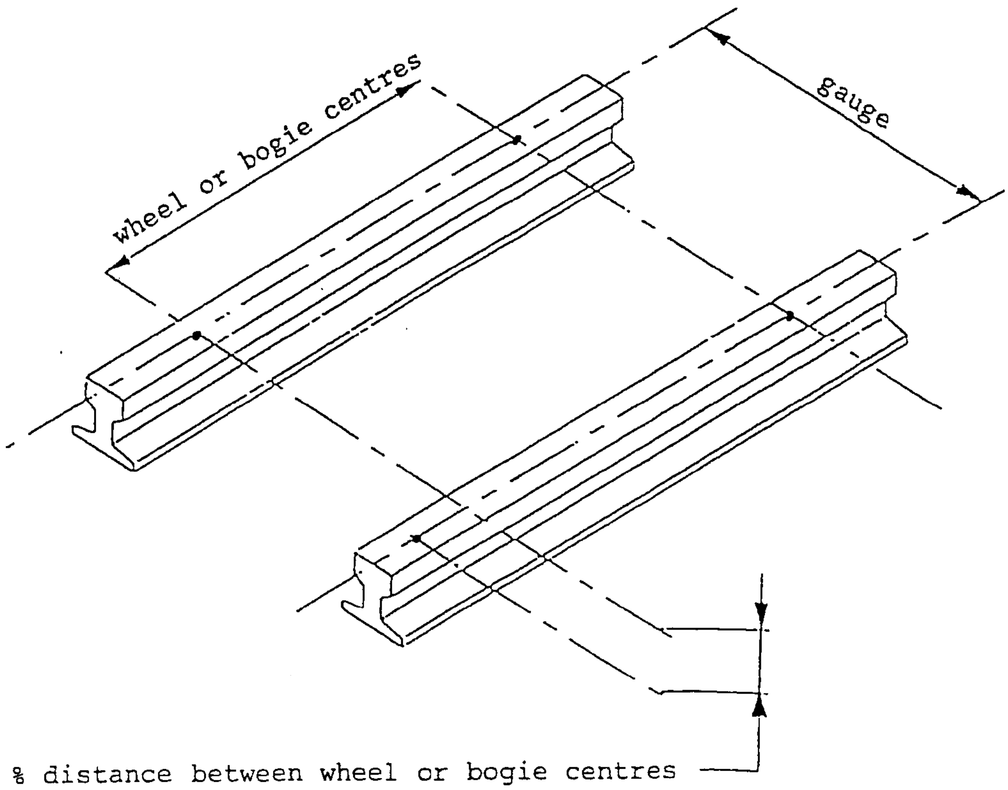
Misalignment at the rail joints is, therefore, not expected and need not be taken into account.

6-2.3.5 RAIL LEVEL TOLERANCES

. <u>Relative rail levels</u>
The greatest divergence in level between the two rails, perpendicular to the track axis, shall be less than 10 mm for tracks up to 10 m centres, and less than 0.1 % of s above with a maximum of 20 mm.



. <u>Height tolerance on 4 points</u>
The rails shall be laid in such a way that the greatest unevenness is the bearing surface is not more than ± 0.1 % of the distance between wheel or bogie centres.

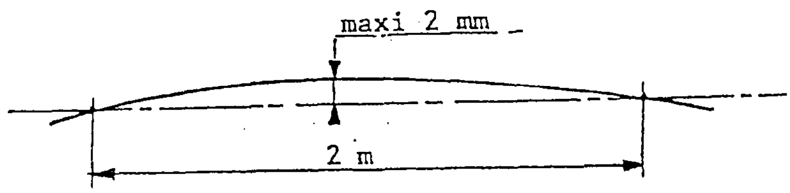


. Local rail curvature (vertical)
The vertical curvature in the longitudinal axis shall not exceed ± 2 mm in any 2 m length taken at random.

. Rail inclination
The longitudinal inclination of the rail rolling surface must not deviate from the theoretical value by more than 0.3 %.

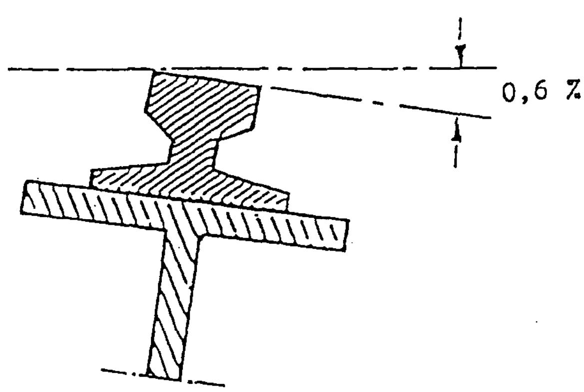
6-2.3.6 LATERAL RAIL TOLERANCES

. Local rail curvature (lateral)
The lateral curvature shall not exceed ± 2 mm in any 2 m length taken at random.



random sampling

. Rail inclination
The lateral inclination of the rail rolling surface must not deviate from the theoretical value by more than 0.6 %.



6-2.4 TOLERANCES FOR TRACKS ON BALLAST

6-2.4.1 ALLOWABLE DEVIATION BEFORE RE-ARRAGEMENT

The tolerances specified in this section apply to new fixed ground-bearing tracks.

The maximum allowable deviation, before the rails must be reset, is shown in brackets.

In certain circumstances, it may be necessary to re-arrange the track before the quoted limit is reached, if the travelling behaviour is noticeably deteriorating.

6-2.4.2 TRACK GAUGE TOLERANCES

The greatest admissible divergence Δs from the nominal span is ± 10 mm (with a maximum of ± 40 mm before re-alignment).

If the machine is guided on one rail only, or if the machine has a pivoted support or is of high elasticity, the initial tolerance may be increased from the above value with the agreement of the machine manufacturer.

6-2.4.3 SAG OF RAILS

It is assumed that under normal working conditions, the deflection of both rail tracks under load is approximately equal and does not significantly effect the performance of the machine.

6-2.4.4 OVERALL RAIL TOLERANCES

. Straightness
For each rail, the overall centre-line, measured at the running surface, shall not deviate by more than ± 6 mm from the theoretical line in both the horizontal and vertical planes, when measured over any 30 m length (with a maximum of ± 12 mm before re-alignment).

For machines guided on one rail only, the requirement for lateral straightness of the non-guiding rail only may be lowered, in agreement with the manufacturer of the machine.

. Rail joints
It is recommended that welded rail joints are used.

Misalignment at the rail joints is, therefore, not expected and need not be taken into account.

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6-2.4.5 RAIL LEVEL TOLERANCES

. Relative rail levels
The greatest divergence in level between the two rails, perpendicular to the track axis, shall be less than 0.1 % of the theoretical track centres (with a maximum of 0.3 % before re-alignment).

. Height tolerance on 4 points
The rails shall be laid in such a way that the greatest unevenness in the bearing surface is no more than ± 0.1 % wheel or bogie centres (with a maximum of 0.3 % before re-alignment) [see scheme in 6-2.3.5].

. Local rail curvature (vertical)
The vertical curvature in the longitudinal axis shall not exceed ± 6 mm in any 30 m length taken at random (with a maximum of ± 12 mm before re-alignment).

. Rail inclination
The average longitudinal inclination of the rail rolling surfaces must not deviate from the theoretical value by more than ± 0.1 % of wheel or bogie centres (with a maximum of ± 0.3 % before re-alignment).

6-2.4.6 LATERAL RAIL TOLERANCES

. Local rail curvature (lateral)
The lateral curvature in the longitudinal axis shall not exceed ± 6 mm in any 30 m length taken at random (with a maximum of ± 12 mm before re-alignment).

. Rail inclination
It is assumed that the track is mounted on resilient pads and that lateral deviation in the level of the rail rolling surface will correct itself under load.

6-2.4.7 DEPTH OF BALLAST

The allowable deviation on the depth of ballast material under the rail sleeper shall be + 150 mm or - 100 mm.

There must be a 300 mm minimum depth of ballast under the sleepers.