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RULES FOR THE DESIGN OF HOISTING APPLIANCES

BOOKLET 4

CHECKING FOR FATIGUE AND CHOICE OF
MECHANISM COMPONENTS



The total 3rd Edition comprises booklets 1 to 8

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This booklet is part of the "Rules for the design of hoisting appliances" consisting of 8 booklets :

- Booklet 1 - Object and scope
- Booklet 2 - Classification and loading on structures and mechanisms
- Booklet 3 - Calculating the stresses in structures
- Booklet 4 - Checking for fatigue and choice of mechanism components
- Booklet 5 - Electrical equipment
- Booklet 6 - Stability and safety against movement by the wind
- Booklet 7 - Safety rules
- Booklet 8 - Test loads and tolerances

and must not be used separately.

BOOKLET 4

CHECKING FOR FATIGUE AND CHOICE OF MECHANISM COMPONENTS

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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 overheating or deflection which could interfere with correct functioning of the mechanism.

4.1.1.

CHECKING FOR ULTIMATE STRENGTH (1)

Mechanism components are checked for ultimate strength by verifying that the calculated stress does not exceed a permissible stress 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 case of loading (clause 2.3.)

(1)

It might seem logical to check against the elastic limit, in line with the structures booklet, as this figure is, in principle, the limit not to be exceeded in the use of a material. The steels normally used for structures have a wide gap between the yield strength and the ultimate strength and this gap affords protection against sudden failure even when the yield strength is considerably exceeded.

On the other hand, the use in mechanisms of certain steels with a very high, elastic limit as compared to the ultimate strength would result in fragile parts being produced if the permissible limit stress were to be based on the elastic limit, and any accidental overstepping of this limit would lead to immediate failure. This explains why the ultimate strength is chosen as the criterion for verification.

4.1.1.2. VALUES OF THE COEFFICIENT ν_R

The values to be adopted for ν_R are given in table T.4.1.1.2.

Table T.4.1.1.2.

Values of ν_R

Cases of loading	I and II	III
Value of ν_R	2.2	1.8

In the case of grey cast iron, the values of ν_R are to be amplified by 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 are designed in compliance with clause 3.3. for booklet 3, checking that the calculated stress does not exceed a limit stress determined as a function of the critical stress above which there is a risk of crippling occurring.

For this check, the coefficient γ_m is taken into account, its value depending on the group in which the mechanism is classified (see table T.2.6.)

Some general considerations relating to the checking of parts for crippling are given in appendix A-3.3.

4.1.3.

CHECKING FOR FATIGUE

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 concentration ;
- the ratio k 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 mechanical component is known only in exceptional cases. Generally speaking, it is to be derived from the characteristics of the material and of the component and from accepted laws concerning their behaviour.

The starting point is provided by the endurance limit under alternating tensile fatigue loading ($r = -1$) of a polished specimen, made from the material under consideration. The diminution 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 ratios κ between extreme stresses is obtained with the aid of a SMITH diagram, in which certain hypotheses are made as to the shape of the strength curve.

The endurance limit thus determined for the actual component, and with respect to a given ratio κ between extreme stresses, is taken as the basis for the plotting of the WÖHLER curve, concerning which certain hypotheses are also made. From this WÖHLER curve (fatigue strength under the effect solely of stress cycles, all having the same ratio κ between extreme stresses), the PALMGREN-MINER hypothesis on fatigue damage accumulation can be used to determine the fatigue strength of a component according to a group in which the component is classified.

The method described in 4.1.3. 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, case-hardening). In such cases the fatigue strength can be derived from the WÖHLER curve only if the latter has itself been determined with respect to 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 case of loading I.

Where the number of stress cycles is less than 8 000, such checking is not necessary.

4.1.3.2. ENDURANCE LIMIT UNDER ALTERNATING LOADING OF A POLISHED SPECIMEN

The specialized works on the subject (see also appendix A-4.1.3.) provide the endurance limit value σ_{bw} under alternating rotational bending of a polished specimen in the case of materials used regularly in construction of mechanisms.

By approximation, the same values of σ_{bw} may be accepted for the endurance limit under alternating rotary bending.

To obtain the endurance limit under alternating axial tension and compression, the values of σ_{bw} have to be decreased by 20 % (1).

-
- (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.

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 in common use in mechanisms, it is permissible to adopt :

$$\sigma_{bw} = 0,5 \sigma_R$$

σ_R being the minimum ultimate strength.

4.1.3.3. INFLUENCE OF THE SHAPE, SIZE, SURFACE CONDITION AND CORROSION

The shape, size, surface condition (machining) and state of corrosion of the component under consideration entail a decrease in the endurance limit under alternating loading in relation to the ideal case of a polished specimen.

This influence is allowed for by introducing factors k_s , k_d , k_u and k_c respectively, concerning the determination of which, directions will be found in appendix A-4.1.3.

The endurance limit under alternating loading σ_{wk} or τ_{wk} of the component under consideration is given for tension, compression, bending and torsional shear by the relation :

$$\sigma_{wk} = \frac{\sigma_{bw}}{k_s k_d k_u k_c}$$

or

$$\tau_{wk} = \frac{\tau_w}{k_s k_d k_u k_c}$$

In the case of pure shear we take :

$$\tau_{wk} = \tau_w$$

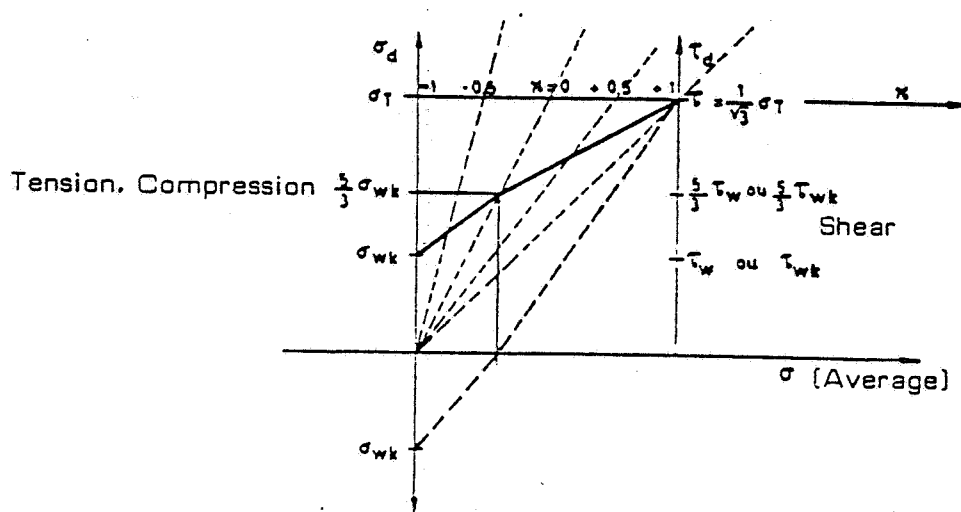
4.1.3.4. ENDURANCE LIMIT AS A FUNCTION OF κ , σ_R AND σ_{wk} (or τ_{wk})

Fig. 4.1.3.4. expresses, in the form of a SMITH diagram, the hypotheses made concerning the relations between the endurance limit σ_d (or τ_d), the ratio κ between extreme stresses, the tensile strength σ_R and the endurance limit under alternating loading σ_{wk} (or τ_{wk}).

This gives the following relations :

Normal stresses	$-1 \leq \kappa < 0$	$\sigma_d = \frac{5}{3-2\kappa} \sigma_{wk}$	Alternating stresses
	$0 \leq \kappa \leq 1$	$\sigma_d = \frac{\frac{5}{3} \sigma_{wk}}{1 - \left(1 - \frac{5}{3} \frac{\sigma_{wk}}{\sigma_R}\right) \cdot \kappa}$	Pulsating stresses
Shear stresses	$-1 \leq \kappa < 0$	$\tau_d = \frac{5}{3-2\kappa} \tau_{wk}$	Alternating stresses
	$0 \leq \kappa \leq 1$	$\tau_d = \frac{\frac{5}{3} \tau_{wk}}{1 - \left(1 - \frac{5}{3} \frac{\tau_{wk}}{\sigma_R}\right) \cdot \kappa}$	Pulsating stresses

Figure 4.1.3.4.



4.1.3.5. WÜHLER CURVE

"WÜHLER curve" in this context means an endurance curve representing the number n of stress cycles which can be withstood before fatigue failure, as a function of the maximum stress σ (τ), when all stress cycles present the same amplitude and the same ratio k between extreme values.

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

- for $n = 8 \cdot 10^3$:

$$\sigma = \sigma_R$$

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 logarithmic scale axes (figure 4.1.3.5.)

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

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

or

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

- for $n = 2 \cdot 10^6$:

$$\sigma = \sigma_d$$

or

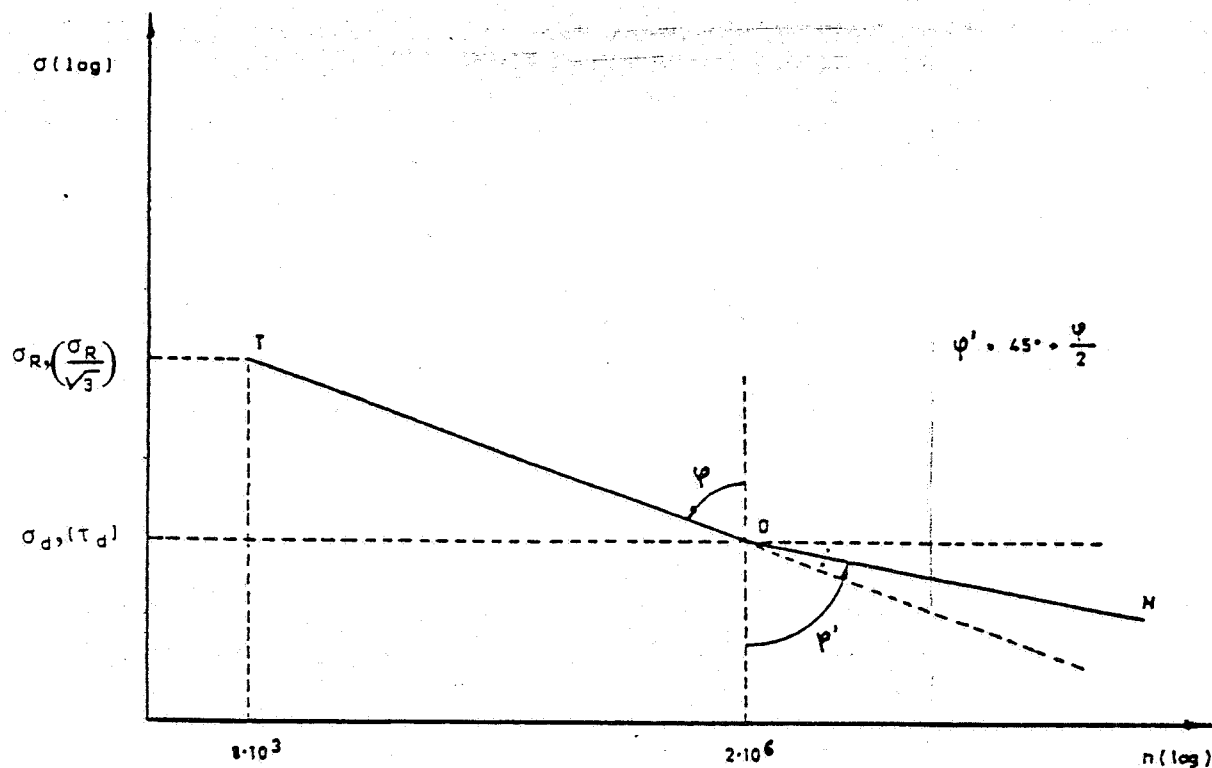
$$\tau = \tau_d$$

- for $n > 2 \cdot 10^6$, the so-called region of endurance limit, the function is represented, in the same reference system as above, by the straight line DN, bisector of the angle formed by the extension of TD and a line parallel to the axis of the n values, passing through D.

The slope of the WÜHLER curve for $n > 2 \cdot 10^6$ is characterized by the factor :

$$c' = \tan \varphi' = c + \sqrt{c^2 + 1}$$

Figure 4.1.3.5.



The spectrum factor k_{sp} of the component is determined by means of the above mentioned value of c . In the case of certain components in group E8 (see 4.1.3.6.), the calculation must also be performed in exactly the same way, but after replacing c by c' . To distinguish between the two spectrum factors thus found, the second will be designated k'_{sp} .

A c value below 2.5 is an indication of faulty design of the component concerned. Such a component must not be put into service.

4.1.3.6. FATIGUE STRENGTH OF A MECHANICAL COMPONENT

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

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

or

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

where j is the component's group number.

In the case of group E8 components, of which the total duration of use n and the spectrum factor k'_{sp} (see 4.1.3.5.) satisfy the inequality :

$$n k'_{sp} > 2 \cdot 10^6$$

σ_k or τ_k must, however, be determined by the expression :

$$\sigma_k = \left(\frac{2 \cdot 10^6}{n} \cdot \frac{1}{k'_{sp}} \right)^{1/c'} \cdot \sigma_d$$

or

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

The group classification of components, 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, are illustrated in figure 4.1.3.6. where σ_{jk} represents the stress applying to group E_j . 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 ψ_k .

One takes :

$$\psi_k = 3.2^{\frac{1}{c'}}$$

or

$$\psi_k = 3.2^{\frac{1}{c'}}$$

for group E8 components satisfying the inequality in the penultimate paragraph of 4.1.3.6.

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

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

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

and one verifies that :

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

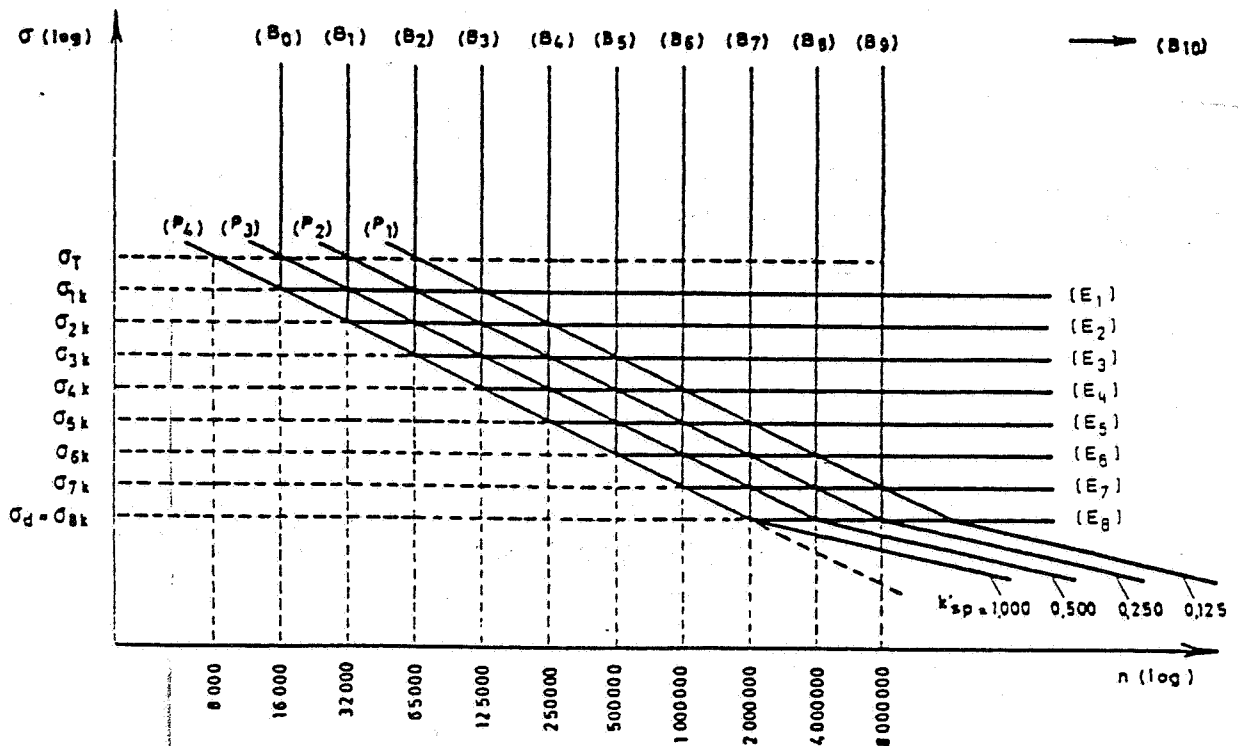
$$\tau \leq \tau_{af}$$

with :

σ maximum calculated normal stress ;

τ maximum calculated shear stress.

Figure 4.1.3.6.



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

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

in which :

σ_x, σ_y = maximum normal stresses in the directions x and y respectively ;

τ = maximum shear stress ;

σ_{kx}, σ_{ky} = fatigue strength for normal stresses, in the directions x and y respectively ;

τ_k = shear fatigue strength.

If it is not possible to determine the most unfavourable case of the foregoing relation from the corresponding stresses σ_x, σ_y and τ , calculations must be performed separately for the loads $\sigma_{x \max}, \sigma_{y \max}$ and $\tau_{\max}$ and the most unfavourable corresponding stresses.

It should be noted that the checks described above do not guarantee safety against brittle fracture. Such safety can be ensured only by a suitable choice of material quality.

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 circumferential velocity must be determined. The figures must be such that, on the basis of present experience, they will not lead to excessive wear.

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 whichever of loading cases I, II or III is the most unfavourable, and
- the maximum dynamic load in the more unfavourable of loading cases 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.) as a function of the class of operation of the mechanism under a constant mean load as defined in clauses 4.2.1.2. and 4.2.1.3. below.

4.2.1.2.

MEAN LOADING OF BEARINGS SUBJECTED TO TYPE S_M LOADS

In order to allow for variations in the loads of type S_M during the cycles of operation, an equivalent mean loading $S_{M \text{ mean}}$ is determined which is supposed to be applied constantly during the theoretical life determined by clause 4.2.1.1.

$S_{M \text{ mean}}$ is obtained by multiplying $S_{M \text{ max II}}$ (1), defined by clause 2.6.4.1. and 2.6.4.2., by the cube root of the spectrum factor k_m defined in 2.1.3.3.

$$S_{M \text{ mean}} = k_m S_{M \text{ max II}} \quad (1)$$

(1)

or $S_{M \text{ max I}}$ for components not subjected to wind.

4.2.1.2.1. Determination of the mean load S_M mean on anti-friction bearings for combined motions

In the case of motions which combine an elevation of the centre of gravity of the moving masses with a horizontal displacement (e.g. unbalanced luffing), the mean load S_M mean is determined by combining :

- the mean load due to the accelerations and the effect of the wind, as determined by applying clause 4.2.1.2. with,
- the mean load due to the vertical displacement of the centre of gravity of the moving masses, as determined from the expression :

$$S_{M \text{ mean}} = \frac{2 S_{M \text{ max}} + S_{M \text{ min}}}{3}$$

where $S_{M \text{ max}}$ and $S_{M \text{ min}}$ are the maximum and minimum values of the corresponding loads.

4.2.1.3. MEAN LOADING OF BEARINGS SUBJECTED TO TYPE S_R LOADS

The extreme loads S_R max and S_R min developed in loading case I for appliances not subjected to wind or loading case II for appliances subjected to wind (see clause 2.6.) are considered and the bearing is designed for a constant mean load given by the expression :

$$S_{R \text{ mean}} = \frac{2 S_{R \text{ max}} + S_{R \text{ min}}}{3}$$

and applied for the theoretical life in accordance with clause 4.2.1.1.

4.2.1.4. MEAN LOADING OF BEARINGS SIMULTANEOUSLY SUBJECTED TO TYPE S_M AND TYPE S_R LOADS

On the basis indicated above the equivalent mean loads are determined for each of the type S_M and S_R loads, assumed to be acting alone and the bearing is selected for an equivalent mean load resulting from combination of the two mean loads S_M and S_R .

4.2.2. CHOICE OF ROPES

The following rules aim at defining reasonable minimum requirements for the choice of ropes used on hoisting appliances covered by these Design Rules.

They do not purport to resolve all the problems nor to serve as a substitute for the dialogue which is essential between the rope manufacturer and the manufacturer of hoisting 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 such ropes, 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 conformity 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 hoisting mechanism. However, for appliances which require frequent dismantling (such as builders tower cranes), in which ropes have to be changed more frequently, it is permissible to select a hoist rope from the group immediately below that of the hoisting mechanism but not inferior to group M 3.

Whenever hoisting appliances are used for dangerous handling operations (e.g. molten metal, highly radioactive or corrosive products, etc.) the choice of the ropes and pulleys must take account of the mechanism group next above that resulting from the normal classification of the hoisting appliances.

Group M 5 is the minimum group to be used for the handling of dangerous loads for the choice of rope and pulley diameters.

4.2.2.1. CHOICE OF ROPE DIAMETER

Two methods can be used at the choice of the manufacturer :

- the method using the minimum practical factor of safety Z_p (see 4.2.2.1.2.) which is valid for running ropes and static ropes (guy ropes, stays, etc.).
- the C factor method (4.2.2.1.3.) applicable to running ropes only.

4.2.2.1.1. Common bases for the two methods

4.2.2.1.1.1. Definition of the maximum tensile force S in the hoist rope (grab ropes excepted)

This is obtained by taking account of the following factors :

- maximum safe working load of the appliance,
- weight of the pulley block and the hoist accessories, the dead weights of which are added to the load effect so as to increase the rope tension,
- mechanical demultiplication due to the rope reeving,
- efficiency of the rope reeving,
- loads due to accelerations if they exceed 10 % of the vertical loads,
- rope inclination at the upper extreme position if the angle of the rope with the hoist axis exceeds 22.5° .

4.2.2.1.1.2. Definition of the maximum tensile force S in ropes other than hoist ropes

The determination of the maximum tensile force S in the various ropes which are not exclusively used for the vertical hoisting of loads is based on the loads determined in load cases I or II taking account of the most unfavourable case which can occur repeatedly in normal use.

For ropes which produce horizontal movement of loads, account must be taken of the loading resulting from rolling motion and friction, together with the maximum inclination that the support, on which the load is moved, can assume locally under the influence of the normal loading.

4.2.2.1.1.3. Determination of the maximum tensile force S in the ropes of multi-rope grabs (holding and closing)

In the case of appliances with grabs, where the weight of the load is not always equally distributed between the closing ropes and the holding ropes during the whole of a cycle, the value of S to be applied should be determined as follows :

- 1) If the system used automatically ensures an equal distribution of the hoisted load between the closing and holding ropes, or any difference between the loads carried by the ropes is limited to a short period at the end of closing or at the beginning of opening, S should be determined as follows :

- a) closing ropes : $S = 66\%$ of the weight of the loaded grab, divided by the number of closing ropes.

b) holding ropes : same pourcentage.

2) If the system used does not automatically secure an equal distribution of the load between the closing ropes and the holding ropes during the hoisting motion and, in practice, almost all the load is applied to the closing ropes, S should be determined as follows :

a) closing ropes : $S = \text{total weight of the loaded grab divided by the number of closing ropes.}$

b) holding ropes : $S = 66 \% \text{ of the total weight of the loaded grab divided by the number of holding ropes.}$

4.2.2.1.2. Method using the minimum practical factor of safety Z_p

Definitions

The minimum 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

$$Z_p = \frac{F_0}{S}$$

4.2.2.1.2.1. Rope selection

The chosen rope must have a minimum 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.

Factor of safety Z_p

Group of mechanism	Minimum value Z_p	
	Running ropes	Static ropes
M 1	3,15	2,5
M 2	3,35	2,5
M 3	3,55	3
M 4	4	3,5
M 5	4,5	4
M 6	5,6	4,5
M 7	7,1	5
M 8	9	5

4.2.2.1.3. C-factor method

Definitions :

- C = rope selection factor,
- S = maximum tensile force exerted on the rope when in use,
- 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,
- R₀ = minimum ultimate tensile stress of the wire composing the rope,
- K' = empirical factor for the minimum breaking load for a given rope construction such that

$$K' = \frac{\pi}{4} \cdot f \cdot k$$

4.2.2.1.3.1. Rope 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 :

$$C = \sqrt{\frac{Z_p}{K \cdot f \cdot \frac{\pi}{4} R_0}} = \sqrt{\frac{Z_p}{K' R_0}}$$

where Z_p is the minimum value for running ropes in table T.4.2.2.1.2., corresponding to the mechanism group chosen for the rope.

The nominal diameter must be such that :

$$d \geq C \sqrt{S}$$

4.2.2.1.3.2. Calculation of the factor C - guarantees

The values of C are calculated taking account of :

- the factor Z_p corresponding to the mechanism group,
- the breaking strength under tension of the steel of the rope wires,
- the factor K' (or factors k and f) which can either :
 - . be taken from ISO Recommendation 2048 for normal ropes covered therein (see Appendix),
 - . or be guaranteed by the rope manufacturer if the rope is of a special construction. In this case, the certificate supplied by the rope manufacturer must clearly state the guaranteed values of K'.

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 :

$$D \geq H \cdot d$$

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.

Note - Refer to 4.2.2. for the mechanism group in which the rope should be classified.

4.2.3.1.1. Values of H

The minimum values of the coefficient H, depending upon the group in which the mechanism is classified, are given in table T.4.2.3.1.1. for drums, pulleys and compensating pulleys.

They correspond to ropes currently used and known and are based on experience concerning their working conditions.

They do not however serve as a substitute for the dialogue which is indispensable between the rope manufacturer and the manufacturer of hoisting appliances, especially when the use of new ropes with varying flexibility characteristics is being considered.

Table T.4.2.3.1.1.

Values of H

Mechanism group	Drums	Pulleys	Compensating pulleys
M 1	11,2	12,5	11,2
M 2	12,5	14	12,5
M 3	14	16	12,5
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

4.2.3.1.2. 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 attachments must be so designed as to withstand a tensile force 2.5 times the maximum tensile 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 of the turns which remain around the drum, the sum of the frictional and fixing forces withstands a tensile force 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 completely unwound from the drum, 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 is determined by considering :

- the load on the wheel,
- the quality of the metal from which it is made,
- the type of rail on which it runs,
- the speed of rotation of the wheel,
- the group classification of the mechanism.

4.2.4.1. RAIL WHEEL SIZE

To determine the 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 :

$$\frac{P_{\text{mean III}}}{b \cdot D} \leq P_L \cdot c_{1\text{max}} \cdot c_{2\text{max}} < 1,38 P_L \approx 1,4 P_L$$

taking $c_{1\text{max}} = 1,2$ and $c_{2\text{max}} = 1,15$

$$\text{and } \frac{P_{\text{mean I,II}}}{b \cdot D} \leq P_L \cdot c_1 \cdot c_2$$

where :

D is the wheel diameter in mm

b the useful width of the rail in mm

P_L a limiting pressure dependent upon the metal used for the wheel, in N/mm^2

c_1 a coefficient depending on the speed of rotation of the wheel

c_2 is a coefficient depending on the group of the mechanism

$P_{\text{mean III}}$ is the mean load to be withstood by the wheel, in loading case III, in N, calculated according to the formulae in clause 4.2.4.1.1.

$P_{\text{mean I,II}}$ is the mean load in case I or II.

4.2.4.1.1. Determining the mean load

In order to determine the mean loads, the procedure is to consider the maximum and minimum loads withstood by the wheel in the loading cases considered, i.e. with the appliance in normal duty but omitting the dynamic coefficient Ψ when determining $P_{\text{mean I,II}}$ and with the appliance not in use for $P_{\text{mean III}}$. The values of P_{mean} are determined by the formula below in the three cases of loading I, II and III :

$$P_{\text{mean I,II,III}} = \frac{P_{\text{min I,II,III}} + 2 P_{\text{max I,II,III}}}{3}$$

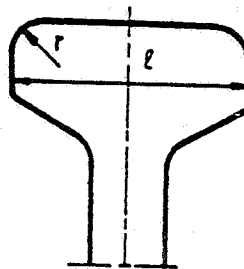
4.2.4.1.2. Determining the useful rail width b

For rails having a flat bearing surface and a total width ℓ with rounded corners of radius r at each side, we have :

$$b = \ell - 2r$$

for rails with a convex bearing surface, we have :

$$b = \ell - \frac{4}{3}r \quad (1)$$



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 metal of which the rail wheel is made ;

Table T.4.2.4.1.3.

Values of P_L

Ultimate strength for metal used for rail wheel	P_L
	in N/mm^2
$\sigma_R > 500 \text{ N/mm}^2$	5,0
$\sigma_R > 600 \text{ N/mm}^2$	5,6
$\sigma_R > 700 \text{ N/mm}^2$	6,5
$\sigma_R > 800 \text{ N/mm}^2$	7,2

(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.

The qualities of metal refer to cast, forged or rolled steels, and spheroidal graphite cast iron.

In the case of rail wheels with tyres, consideration must obviously be given to the quality of the tyre, 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 in the case of 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 to be feared.

When these are used their diameter is determined by taking P_L equal to 5 N/mm^2 .

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/min.

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/min														
	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	0,96	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	0,87
1 250	-	-	-	1,17	1,15	1,14	1,13	1,11	1,09	1,06	1,03	1	0,97	0,94	0,87

4.2.4.1.5. Determining the coefficient c_2

The coefficient c_2 depends on the group classification 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

Group classification of mechanism	c_2
M 1 to M 4	1,12
M 5	1
M 6	0,9
M 7 - M 8	0,8

4.2.4.2. NOTES

Note 1

The formulae apply only to wheels whose diameters do not exceed 1,25 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 method of calculating set out above is derived from 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 N/mm^2

E the modulus of elasticity of the metal in N/mm^2

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 dimension of a pressure in N/mm^2 , the relation may be written :

$$K_L = \frac{P}{b \cdot D}$$

and characterizes the wheel pressure on the rail. The formula of clause 4.2.4.1. is obtained by putting :

$$K_L = P_L \cdot c_1 \cdot c_2$$

4.2.5.

DESIGN OF GEARS

The choice of the method of making 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 2.6.

In the case of a calculation which takes account of the operating time the conventional hours determined in 2.1.3.2. should be used.

APPENDIX A-4.2.2.

COMMENTS ON THE CHOICE OF ROPES AND ON THE PROBLEM OF THE FACTOR OF SAFETY

The useful life of a hoisting rope depends on factors inherent both in the construction of the rope (internal factors) and in the characteristics of the hoisting appliance and the way in which the rope is reeved (external factors).

The main external factors are the tensile load, the pulley diameters and the type and number of working cycles.

These factors will in principle determine the rope winding diameters.

Figure F.A.4.2.2. shows the relationships between the tensile stress, the pulley diameters and the useful life (number of bending reversals causing failure) for a given rope (of diameter 16 mm).

According to this diagram, the larger the pulley diameter and the smaller the tensile stress, the longer the useful life of the rope.

winding diameters must be determined so as to ensure a reasonable useful life of the rope before its replacement.

The old method of calculation defined fixed minimum factors of safety and minimum winding diameters (as a function of the rope diameter) for certain uses, such as for example in hoisting appliances.

This method, which is still used and sometimes even stipulated by law in many countries, does not enable the conditions for a reasonable useful life to be met. The loads and the type of utilisation of hoisting appliance mechanisms differ considerably from one appliance to another with the result that such minimum values may be too high in certain cases (e.g. for a power station overhead travelling crane) or too low in other cases (e.g. for a grabbing crane on arduous duty).

Even if the safety factors were chosen according to the groups of the mechanisms, the calculation of winding diameters on the basis of these safety factors could not result in a good design for the following reasons :

The manufacturer wishes to use ropes of the smallest possible diameter in order to obtain the smallest possible diameters for the pulleys and drums. For a given safety factor, he achieves this by using wire of the greatest possible strength and using a rope with the greatest possible fill factor.

Yet the useful life of a rope whose diameter has been calculated in this way is not always optimum.

A rope of identical diameter, of which the wire is of lower strength (e.g. 1600 N/mm² instead of 2200 N/mm²) and the fill factor is lower (e.g. eight strands instead of six) may have a much longer useful life in spite of the lower factor of safety.

Another difficulty stems from the fact that the safety factor is related to breaking strengths whose definitions differ from one country to another, and safety factors do not have the same meaning if they relate to a breaking strength defined in different ways.

Four definitions exist for the breaking strength of ropes :

- Calculated breaking strength : this is the cross-section of the rope multiplied by the strength of the wire of which it is composed ;
- Theoretical breaking strength : given by the sum of the breaking strengths of the wires used in the rope ;
- Actual breaking strength : this is the load obtained by an ultimate tensile strength test on the rope ;
- Nominal breaking strength : this is the minimum breaking strength guaranteed by the rope manufacturer.

When a rope is determined by using a safety factor which is related either to the actual breaking strength or to the nominal breaking strength, the manufacturer tends to adopt ropes in which the spinning loss (difference between the theoretical breaking strength and the actual breaking strength) is as small as possible in order to obtain a smaller rope diameter. The spinning loss, however, is not related to the resistance of a rope to repeated bending. A satisfactory useful life for ropes cannot therefore be obtained with this method of calculation.

This shows that the safety factor is not an adequate basis for determining the winding diameters required to ensure satisfactory rope life under bending reversals ; indeed this method will often prevent the optimum solution from being obtained.

Because it is difficult for the manufacturer to allow for the influence of these different factors, it is preferable to determine the rope diameter d_{min} simply as a function of the tensile load S , from the formula :

$$d_{min} = C \sqrt{S}$$

where C is a coefficient depending solely on the group of the mechanism.

In cases where non rotating ropes are used (e.g. tower cranes in which the load is suspended from a single part) and for dangerous handling operations (e.g. molten materials), the values of C are increased above the normal values in order to compensate for the more unfavourable rope construction or the greater risk.

The value of C , the safety factor Z_p referred to the theoretical breaking strength and the rope fill factor f (ratio of the metal cross section of the rope to the area of the circle circumscribing the rope) are linked by the following relation :

$$C = \sqrt{\frac{Z_D}{k \cdot f \cdot \frac{\pi}{4} R_0}}$$

where R_0 is the ultimate strength (in N/mm^2) of the wire used in the rope.

The values of C apply to ropes made with wire having a strength of 1600, 1800, 2000 or 2200 N/mm^2 .

Where, exceptionally, use is made of a rope composed of wires with a strength of 1400 N/mm^2 , the rope diameter must be increased accordingly.

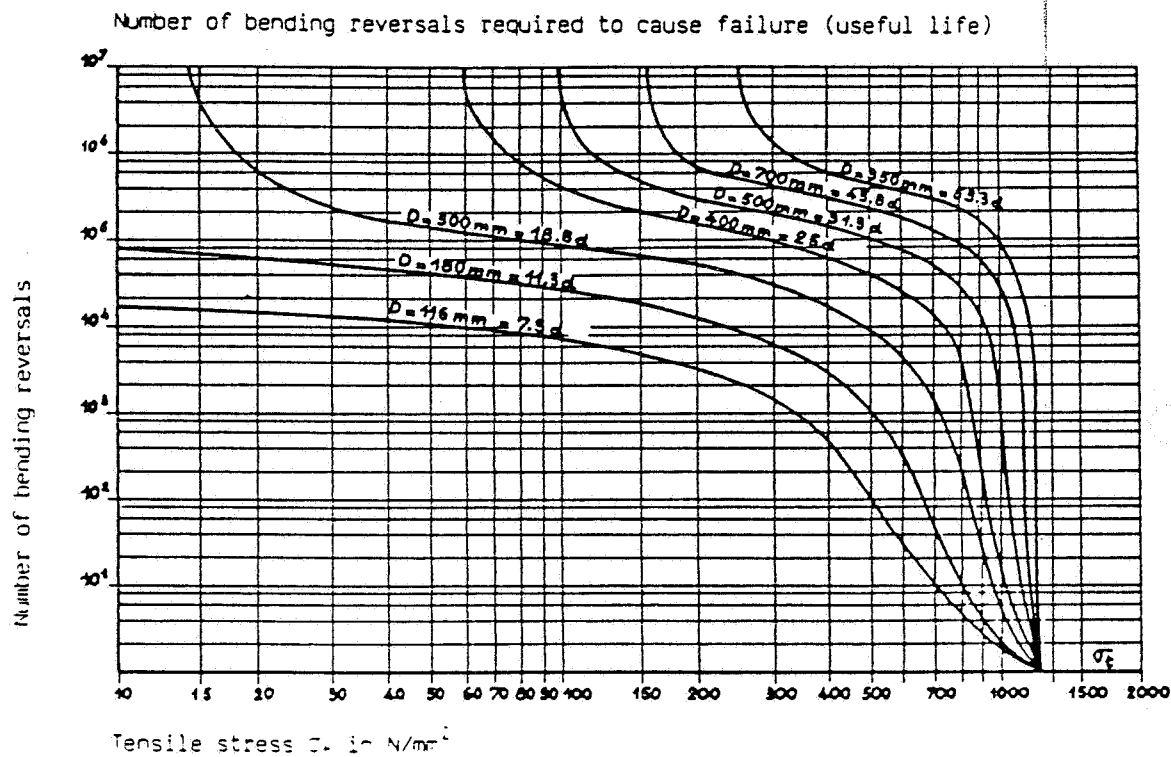
The rope manufacturer or crane maker must choose the composition and cross section of the rope, for the calculated minimum diameter d , to suit the reeving conditions of the particular rope and in the light of the latest technical progress.

Figure A.4.2.2.

Influence of pulley diameter D and tensile stress σ_t on the useful life of a rope :

Langs lay rope, diameter 16 mm, six strands of nineteen 1 mm dia. wires,
 $R_0 = 1400 N/mm^2$

Cast-iron pulleys with machined groove of radius $r = 8,5$ mm.



APPENDIX A-4.2.3.

CONSIDERATIONS ON THE DETERMINATION OF MINIMUM WINDING DIAMETERS FOR ROPES

There are no absolute minimum pulley and drum diameters below which a rope could no longer operate. Nor is there a required absolute minimum diameter for different types of rope.

The useful life of a rope is progressively reduced with decreasing pulley and drum diameters, if other conditions remain unchanged.

Figure A.4.2.3. shows the pattern of behaviour of a particular rope.

In order to ensure an adequate useful rope life, the minimum winding diameters D must be determined, as a function of the group classification of the mechanism involved, from the formula :

$$\frac{D}{d} > H$$

where d is the nominal rope diameter and H is a coefficient which is chosen according to the group in which the mechanism is classified and which becomes higher when the duty is more arduous.

The coefficient H is higher for pulleys than for drums because, in the course of a cycle of operations, the rope is subjected to twice as many bending reversals over a pulley (rope straight, rope bent, rope straight) as on a drum (rope straight, rope bent).

The coefficient H is lower for equalizing pulleys because the rope undergoes fewer bending reversals and the movement is normally very limited. Such pulleys must nonetheless be dimensioned with reference to the number of bending reversals.

Unfavourable winding conditions, e.g. reeving around several pulleys, reverse bends, or the use of non-rotating ropes whose construction is less favourable for bending reversals, must be compensated for by a suitable increase to ensure a useful rope life commensurate with the group classification of the mechanism.

Figure A.4.2.3.

Influence of pulley diameter D and tensile stress σ_t on the useful life of a rope :

Langs lay rope, diameter 16 mm, six strands of nineteen 1 mm dia. wires,
 $\sigma_R = 1400 \text{ N/mm}^2$

Cast-iron pulleys with machined groove radius $r = 8,5 \text{ mm}$.

