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1 Calculation

1.1 General

The calculations must be in compliance with accepted rules of statics, dynamics, and theoretical mechanics.

Provided that identical factors of safety are adhered to, the calculations may also be based on the results of tests carried out for determining the stresses which are produced in a structure under the assumed load conditions.

The data concerning system, dimensions, and cross sections shown in calculations and drawings must be in agreement. Discrepancies are not permissible unless these have the effect of increasing beyond any doubt the safety of all components.

1.2 Calculation method

For the loads assumed as explained in section 2, the three possible causes of failure are taken into account as follows:

A — Overstraining of materials beyond the elastic limit
B — Overstraining of structures beyond the critical buckling stresses
C — Overstraining of materials beyond the fatigue strength

1.3 Permissible stresses

Details on the permissible stresses concerning A, B, and C above can be seen from the future FEM design rules for crane structures.

Relevant national standards apply for the transition period.

2 Assumed loads

Structures are subjected to the following load categories:

main loads,
additional loads and
special loads.

Main loads:
- Deadloads,
- lifted loads (lifting carriage deadweight and weight of the load unit),
- horizontal forces of inertia produced by drive units,
- statical stabilizing forces.

Additional loads:
- Forces due to running askew,
- effects due to temperature,
- loads on walkways, stairs, platforms, and handrails.

Special loads:
- Buffer forces,
- test loads,
- emergency catch loads.

2.1 Categories of main loads

2.1.1 Deadloads

Deadloads are the forces due to the weight of all fixed and moving parts which are permanent operational components of the mechanical and electrical equipment, and of the proportionate share of the load supporting means such as ropes but with the exception of the loads listed in 2.1.2.

2.1.2 Lifted loads

The lifted loads comprise the weight of the load unit and the deadweight of assemblies receiving the load unit such as telescopic load fork and roller table, the deadweight of the lifting carriage, and the proportionate share of the weight of the load supporting means such as ropes, chains, etc.

2.1.3 Effects of vertical forces of inertia

The effects of vertical forces of innertia produced when moving the lifting carriage and the loads listed in sections 2.1.1 and 2.1.2 are taken into account by applying „deadload coefficients" φ and „lifted load coefficients" ψ.

2.1.3.1 Deadload coefficients φ

The deadloads of S/R machines in motion (see section 2.1.1) and the corresponding stress resultants or stresses shall be multiplied by a deadload coefficient φ selected from table 1.

Table 1

Travel speed v_f in m/min, Runways 1: with joints	without joints	Deadload coefficient φ
up to 63	up to 100	1.1
above 63 to 125	above 100 to 200	1.2
above 125	above 200	1.3

In the case of live loads on a S/R machine running on spring-loaded, plastic, etc. wheels φ = 1.1 may be used for calculation independent of the travel speed and the type of runway.

Taking an S/R machine on steel rim travel wheels as an example:

a) Travel speed v_f = 125 m/min, φ = 1.2
b) Travel speed v_f = 50 m/min, φ = 1.1

2.1.3.2 Lifted load coefficient ψ and lifting classes

The lifted loads according to section 2.1.2 or the stress resultants or the corresponding stresses must be multiplied by a lifted-load coefficient ψ according to table 2. Its value depends on the surge to be expected from the load lifting means when lifting starts and thus on the nominal lifting speed v_H; it is the smaller the more flexible hoist unit and structure and the lower and more

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constant acceleration and deceleration are when the vertical motion is reversed.

The S/R machines are therefore classified into lifting classes H1, H2, and H3, each of which has a different load coefficient ψ assigned to it according to table 2.

Table 2

Lifting class	Lifted load coefficient ψ for a lifting speed v_H of up to 90 m/min	Average main hoist acceleration $\pm a_m$ in m/s²
H1	$1.1 + 0.0022 \times v_H$	$\leqq 0.6$
H2	$1.2 + 0.0044 \times v_H$	$\leqq 1.3$
H3	$1.3 + 0.0066 \times v_H$	> 1.3 [1]

1) $\pm a_m$ 1.3 m/s² max. if persons ride on the lifting carriage during vertical motions

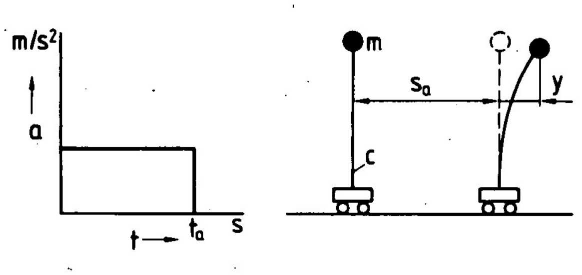
2.1.4 Horizontal forces of inertia produced by drive units

2.1.4.1 Dynamic oscillation coefficient

The forces of inertia affecting the structure of an S/R machine due to acceleration and deceleration of motions such as horizontal travel, lifting, and telescoping shall be determined on the basis of the maximum forces produced by the respective drive units in regular service. To simplify the calculation for taking into account the dynamic effects, the „quasistatic" forces affecting the structure may be multiplied by the coefficient S_w, the „quasistatic" forces being those which result when considering the system's centre-of-mass motion under the effects of the drive forces, of the resistances to motion and of the forces of inertia. Application of the coefficient S_w is subject to the condition that the driving forces act on the S/R machine practically without any play.

For dimensioning the S/R machines it is necessary to know exactly the stresses due to oscillations caused by travel motions. Deflections, stresses, and stress resultans shall be multiplied by the dynamic oscillation coefficient S_w.

Figure 1. **Dynamic equivalent system**



$$m\ddot{y} + c \times y + m \times a = o$$

$$\ddot{y} + \frac{c}{m}\, y + a = o$$

$$\ddot{y} + \omega^2 \times y + a = o \qquad \omega^2 = \frac{c}{m}$$

$$\ddot{y} + \omega^2 \times y = -a$$

The integral function of the above differential equation is:

$$y_{(t)} = c_1 \times \sin \omega t + c_2 \times \cos \omega t - \frac{a}{\omega^2}$$

The integration constants c_1 and c_2 are determined from the initial conditions:

$$y_{(0)} = o = c_1 \times o + c_2 - \frac{a}{\omega^2} \qquad c_2 = \frac{a}{\omega^2}$$

$$\dot{y}_{(0)} = o = \omega \times c_1 - c_2 \times o \qquad c_1 = o$$

Thus, the oscillation equation is:

$$y_{(t)} = o \times \sin \omega t + \frac{a}{\omega^2} \times \cos \omega t - \frac{a}{\omega^2}$$

$$y_{(t)} = \frac{a}{\omega^2} \times (\cos \omega t - 1) \qquad \text{or}$$

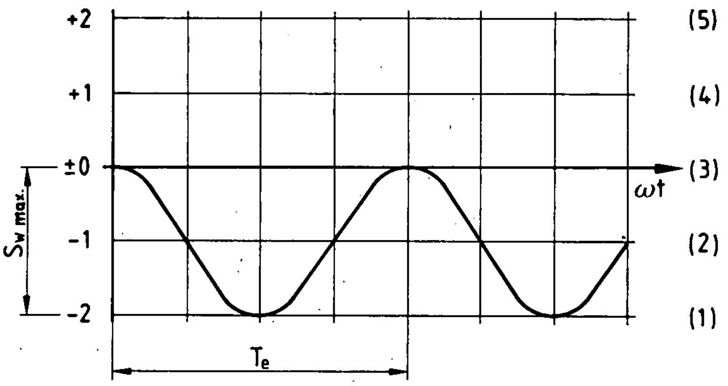
$$y_{(t)} = \frac{m \times a}{c} \times (\cos \omega t - 1)$$

dynamic deformation = static deformation x oscillation coefficient S_w

where:

- m = dynamic equivalent mass of the flexible masses
- c = spring constant of the structure
- a = mean deceleration (acceleration) of the horizontal motion
- s_a = braking path (path of acceleration)
- t_a = period of deceleration (period of acceleration)
- $y_{(t)}$ = dynamic deformation

Figure 2. **Oscillation coefficient** $S_w = \cos \omega t - 1$



(1) maximum amplitude of oscillation during deceleration (–) $S_{w\ max}$
(2) „quasistatic" mean position during deceleration
(3) position of rest
(4) „quasistatic" mean position during acceleration
(5) maximum amplitude of oscillation during aceleration (+)

Assumptions:

- damping is ignored
- constant deceleration (or acceleration)
- $t_a > T_e$
  (T_e = natural oscillating period of the structure)

The oscillation function

$$y_{(t)} = \frac{m \times a}{c} \times (\cos \omega t - 1)$$

reaches its maximum if the expression $(\cos \omega t - 1)$ assumes the value (–2).