2nd edition 09.1982

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

Continued on pages 2 to 5

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/s^2
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/s^2 max. if persons ride on the lifting carriage during vertical motions

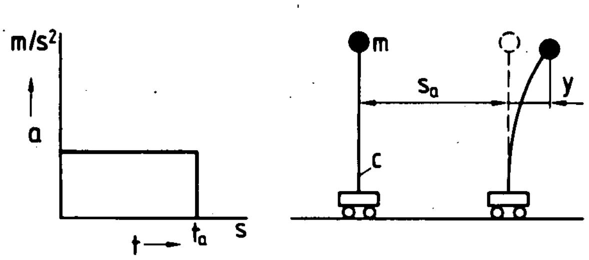
2.1.4 Horizontal forces of inertia produced by drive units

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:

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

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

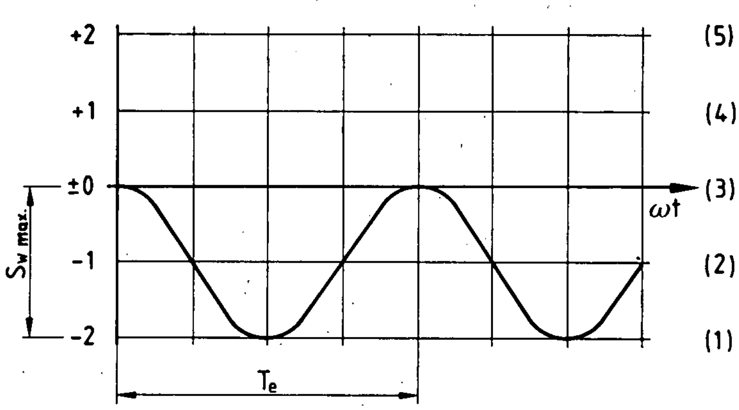
$$y_{(t)} = \frac{m \times a}{c} \times (\cos \omega t - 1)$$

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).

Since the stresses to which the structure is subjected are directly proportional to the dynamic deformations, the oscillation coefficient $S_{w\,max} = 2$ must be used for further calculations of deformations, stresses, and stress resultants.

2.1.4.2 Effect of horizontal forces of inertia

Without the coefficients φ and ψ according to section 2.1.3 but with the oscillation coefficient according to section 2.1.4.1, the dynamic effects of the masses resulting from structure flexibility shall be assumed to be applied to the individual centres of gravity (s_1, s_2, s_3, etc.).

Example:

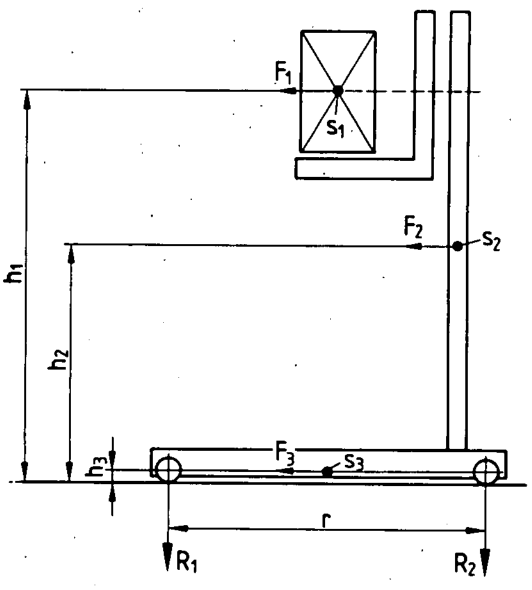
F_1 horizontal dynamic force of acceleration due to the load unit and the weight of the lifting carriage

F_2 horizontal force of dynamic acceleration due to the distributed load of the mast

F_3 horizontal force of dynamic acceleration due to the weight of bottom carriage, travel mechanism and attachments (e. g. control cubicle, hoist mechanism, etc.)

The masses shall be considered according to their distribution in each particular case.

Figure 3.



Depending on the direction of acceleration, the dynamic forces of inertia tend to increase or to decrease the wheel loads R_1 and R_2.

$F_1 = (m_L + m_H) \times 2 \times a_m$, since $S_w = 2$

$F_2 = m_s \times 2 \times a_m$, since $S_w = 2$

$F_3 = (m_B + m_A) \times a_m$, since $S_w = 1$ (unsprung)

where:

m_L mass of the load unit

m_H mass of the lifting carriage

m_S mass of the mast

m_B mass of bottom carriage

m_A mass of the attachments to the bottom carriage

a_m mean acceleration

$$R_{min} = R_{stat} - \frac{F_1 \times h_1 + F_2 \times h_2 + F_3 \times h_3}{r}$$

$$R_{max} = R_{stat} + \frac{F_1 \times h_1 + F_2 \times h_2 + F_3 \times h_3}{r}$$

For S/R machines, the stability is defined as follows:

Stability:

$$\nu = \frac{\Sigma \text{ stabilising moments}}{\Sigma \text{ overturning moments}}$$

$$\nu = \frac{R_{stat} \times r}{F_1 \times h_1 + F_2 \times h_2 + F_3 \times h_3}$$

The stability ν shall be $\geqq$ 1.5 under service conditions; in exceptional situations, e. g. for emergency braking, it shall be at least 1.1 (leaving the safety claws out of consideration).

For S/R machines which are positively prevented from tilting (e. g. by positively guided travel wheels), the stability analysis is omitted if the negative wheel loads are completely absorbed by the guiding members and transmitted to the rack or building structure.

2.1.4.3 Friction force coefficient μ

The driving forces on the travel wheel periphery are limited by the friction coefficient applicable to the travel wheel/rail pair.

The driving force which can be continuously transmitted is thus $\leqq R_{min} \times \mu$. For steel-on-steel pairs, $\mu = 0.2$ shall be used as a maximum.

2.1.4.4 Lateral forces S_z

If, due to forces of inertia, lateral forces occur transversely to the runway centre line, these shall be transmitted to the rails by form and friction grip in line with the guiding members and the systems of the structure and travel mechanism.

2.2 Categories of additional loads

2.2.1 Forces due to running askew

Depending on the systems of travel mechanism and structure, form grip forces are produced which act on the guiding members (wheel flanges or guide rollers) of an S/R machine by running askew at an angle α, and from these frictional rubbing forces result which act on the contact surfaces of the travel wheels.

For the generally accepted tolerances of travel wheel diameter, parallelism of the wheel bore axes and runway alignment, a linear law of friction forces applies to the longitudinal and transverse slip of two travel wheels of ferrous material on a steel rail:

$$f = 0.30 \times (1 - e^{-0.25 \times \alpha})$$

where:

e = base of the natural logarithms 2.71828

α = angle of skew in ‰

Table 3. **Friction force coefficient f as depending on the angle of skew α**

α ‰	0.5	1.0	1.5	2.0	2.5	3.0	3.5	4.0	4.5	5.0	6.0	7.0	8.0	9.0	10.0	15.0
f	0.035	0.066	0.094	0.118	0.139	0.158	0.175	0.190	0.203	0.214	0.233	0.248	0.259	0.268	0.275	0.300

$S = f \cdot R_{stat}$ (force due to running askew)

where:

R_{stat} wheel load due to deadloads and lifted load without the coefficients according to sections 2.1.3 and 2.1.4

α $\alpha_F + \alpha_v + \alpha_o + \alpha_s \leqq 10$ ‰

angle of skew in relation to the spacing of the guide members which give rise to the frictional forces, the sum total of all possible misalignments transverse to the runway when the S/R machine takes up a skewed position.

α_F angle of skew due to the clearance between straight rail and form grip guide members, i. e. at least 5 mm for wheel flanges and 3 mm for guide rollers.

α_v angle of skew due to wear. At least 1.5 % of the rail head width in the case of guide rollers and at least 5 % of the rail head width in the case of wheel flanges.

$\alpha_o \leqq 1$ ‰ angle of skew due to S/R machine tolerances

$\alpha_s \leqq 1$ ‰ angle of skew due to tolerances of the rail laid on the floor

2.2.2 Temperature effects

Temperature effects must be taken into consideration only in special cases. Based on an assumed temperature of + 10 °C during erection, variations in temperature of ± 35 K or, in the case of an uneven temperature rise in individual sub-assemblies, variations of ± 15 K shall be assumed for an S/R machine operating in an open building.

These values must be adequate for local conditions in the case of S/R machines operating in plants with a hot atmosphere or in low-temperature warehouses.

A coefficient of elongation

$\alpha_t = 12 \cdot 10^{-6}$ mm/mm x K (for steel)

shall be used for the calculations.

2.2.3 Loads on platforms and handrails

In addition to the deadloads, a single live load shall be assumed for platforms, i. e.

300 kg for persons walking on it and carrying a load
150 kg for persons walking on it without a load.

A concentrated live load applied horizontally to the outside or to the inside shall be assumed for handrails, i. e.

30 kg for persons carrying a load and
15 kg for persons without a load.

These concentrated loads need not be taken into account in the case of structural members subjected to loads lifted according to section 2.1.2.

2.3 Categories of special loads

2.3.1 Buffer forces

For this special case of loading it is assumed that during regular service collisions of S/R machines do not occur.

The buffer forces F_p, which occur when S/R machines impact against runway stops, shall be limited by providing adequate buffers or similar means of dissipating energy. The required working capacity for energy dissipation and the maximum buffer forces F_p shall be determined for S/R machines on the basis of 100 % nominal travel speed.

If automatic means [2] ensure a reduction of the travel speed, the required working capacity of the buffers for energy dissipation and the maximum buffer forces F_p may be calculated taking the then fastest possible travel speed, but at least 70 % of the nominal speed, as a basis.

The kinetic energy shall be assumed to be

$$W_{kin} = \frac{m \times v^2}{2}.$$

For an analysis of the buffers and the strength of the structure the forces due to the moving masses of the deadloads and of the guided lifted loads [3] shall be assumed to be effective in their respective least favourable position, but without the coefficients according to sections 2.1.3 and 2.1.4.

A corresponding equivalent mass has to be introduced into the calculation for rotating parts of the travel mechanism. The buffer forces shall be distributed according to the buffer characteristics and the freedom of movement of the structure.

If energy converting buffers are provided, an amount of 10 % may be deducted from the total energy for the free dying down of structure oscillations.

In the case of a simplified analysis of the stresses in the structure, the buffer end forces shall be multiplied by an oscillation coefficient selected from table 4 in accordance with the shape and area below the buffer characteristic.

Table 4. **Oscillation coefficients for simplified calculation**

The area below the buffer characteristic resembles approx.	Oscillation coefficient S_p at S/R machine impact
Triangle	1.25
Rectangle	1.50

2) In the case of electric means at least 2 safety devices shall be provided which monitor each other.

3) The maximum possible friction force between load unit and load handling accessory shall be assumed for the load unit as a non-guided lifted load.

The travel wheels of S/R machines with or without a payload shall not lift off the rail due to the 110 % buffer force and the above-mentioned deadloads and lifted loads. Reaction pressure rollers or safety claws shall be provided as required for absorbing negative wheel loads.

The buffer stops shall be dimensioned for absorbing at least the actual buffer end force S_p.

2.3.2 Test loads

2.3.2.1 General function test

The functions of load pick-up, lifting, lowering, and travelling of S/R machines are tested by applying the following load:

test load: $P_k = 1.25$ payload

For the analysis of stresses (overstraining beyond the elastic limit) and of stability the test load P_k shall be multiplied by the reduced lifted load coefficient $\psi' = \frac{1 + \psi}{2}$.

The assumed load makes it necessary to proceed as follows for applying the test load:

With the test load applied, all motions shall be individually tested with reasonable care and with the load in the least favourable positions. A motion may not be initiated before the oscillations caused by the previous motion have died down.

2.3.3 Emergency catch loads

2.3.3.1 Function test of the catching device

On S/R machines equipped with a speed controlled catching device, the function test of this device is carried out by applying the catch load $P_k = 1.25 \cdot$ payload.

The catch load shall be positioned centrally on the load pick-up device.

The following tests shall be carried out:

1) function test of the overspeed governor by lowering the lifting carriage at excessive speed or by a simulation of this condition;
2) function test of the catching device by manual locking of the overspeed governor during the lowering motion;
3) free-fall stop test of the lifting carriage lifted up together with the catch load.

The free-fall stop test according to 3) above can be omitted if the catching device is built up of function-tested components for which an official certificate has been issued.

For the stress analyses according to section 1.2 A „Overstraining of materials beyond the elastic limit" and B „Overstraining of structures beyond the critical buckling stresses" the catch load shall be multiplied by the increased lifted load coefficient ψ''.

For specifying the increased lifted load values ψ'' it has been taken into account that the actual speed for tripping the catching device v_{ab} considerably exceeds the nominal lifting speed v_H, which is espressed by the following equation::

$$v_{ab} = \sqrt{v_g^2 + 2 \times g \times S_R + 2 \times g \times S_F}$$

where:

v_g preset tripping speed of the overspeed governor $\approx 1.4 \times v_H$

S_R spacing of the notches of the overspeed governoor in relation to the vertical path of the lifting carriage

S_F idle path of the catching mechanism until braking starts

g 9.81 m/s²

The energy equation indicates the mean deceleration during the catching operation:

$$\frac{m \times v_{ab}^2}{2} + m \times g \times h - m \times a_m \times h = o$$

$$\text{hence:} \quad a_m = \frac{v_{ab}^2}{2h} + g$$

where:

a_m mean deceleration

h catching or braking path of the lifting carriage

m mass of the test load and of the lifting carriage dead weight

The maximum deceleration a_{max} in the event of catching results from the characteristic curve of the working diagram for the catching device and by taking into account an oscillation coefficient according to table 4.

Table 5.

a_{max}	$2 \times a_m$	triangular diagram
	$1.75 \times a_m$	trapezoidal diagram
	$1.5 \times a_m$	rectangular diagram

The increased lifted load coefficient ψ'' results from the relationship between the maximum deceleration due to braking and acceleration due to gravity.

$$\psi'' = \frac{a_{max}}{g}$$

If an exact calculation of the increased lifted load coefficient is omitted, the calculation may be simplified by taking ψ'' from table 6.

Table 6. **Approximate figures for ψ''**

Increased lifted load coefficient ψ'' 4)		Lifting speed v_H (m/min)			
for catching by	approx. working diagram	$\leqq 20$	$\leqq 31.5$	$\leqq 50$	> 50
Clamping roller catching device	triangular	5	6	–	–
Brake type catching device	trapezoidal	2.5	3	3.5	–
Cushioning catching device	rectangular	2	2	2	2

4) The above values for ψ'' can be influenced by varying the design of the catching device (angle of the sloping surface in the catching block, width and diameter of the catching roller)

Depending on particular applications, additional calculations regarding oscillation dynamics may be advisable if due to its flexibility the structure is definitely capable of absorbing a certain part of the energy produced by catching.

Erstellt durch den technischen Unterausschuß „Regalbediengeräte und Stapelkrane" der Sektion IX der
Prepared by the Technical Sub-Committee „Storage and Retrieval Machines and Stacker Cranes" of Section IX of the
Etabli par le sous-comité technique „Transtockeurs et ponts gerbeurs" de la section IX de la
Emesso della sottocommissione tecnica „Trasloelevatori e gru impilatrici" della sezione IX della

Fédération Européenne de la Manutention (FEM)

Sekretariat:	Sekretariat der FEM Sektion IX
Secretariat:	c/o VDMA
Secrétariat:	Fachgemeinschaft Fördertechnik
Segretariato:	Postfach 71 01 09
	D–6000 Frankfurt 71

Zu beziehen durch das oben angegebene Sekretariat oder durch die folgenden Nationalkomitees der FEM
Available from the above secretariat or from the following national committees of the FEM
En vente auprès du secrétariat ou des comités nationaux suivants de la FEM
Da richiedere attraverso il su citato segretariato o i comitati nazionali della FEM

Belgique

Comité National Belge de la FEM
Fabrimétal
Rue des Drapiers 21
B – 1050 Bruxelles

Danmark

Dansk National Komite FEM
Transportmaterial Foreningen
Lille Kirkestraede 3
DK – 1072 Kopenhagen K

Deutschland

Deutsches Nationalkomitee der FEM
VDMA
Fachgemeinschaft Fördertechnik
Postfach 71 01 09
D – 6000 Frankfurt 71

España

Comité Nacional Español de la FEM
Associación Nacional de Ingenieros Industriales
Via Layetana 39 (bajos)
E – Barcelona 3

France

Comité National Français de la FEM
SIMMA – Syndicat des industries de materiels de manutention
10, avenue Hoche
F – 75 382 Paris cedex 08

Great Britain

British National Committee of FEM
Messrs. Peat, Marwick, Mitchell & Co.
7 Ludgate Broadway
GB – London EC 4V 6DX

Ireland

Irish National Committee of FEM
Institute for Industrial Research and Standards (IIRS)
Ballymun Road
IRL – Dublin 9

Italia

Comitato Nazionale Italiano de la FEM
Associazione Nazionale Industria
Meccanica Varia et Affine (ANIMA)
Piazza Diaz 2
I – 20 123 Milano

Luxembourg

Comité National Luxembourgeois de la FEM
Groupement des Constructeurs et Fondeurs du Grand-Duché de Luxembourg
Rue Alcide de Gasperi 7
Plateau de Kirchberg
Boîte postale No. 1304
L – Luxembourg

Nederland

Nederlands National Comité bij de FEM
GKT-Vereniging van Metaal industrieën
Postbus 190
NL – 2700 Ad Zoetermeer

Norge

Norwegian FEM Groups
Norsk Verkstedsindustris
Standardiseringssentral NVS
Box 7072 H
N – Oslo 3

Portugal

Commissao Nacional Portuguesa da FEM
CEMUL – Prof. Eng. L. O. Faria
Avenida António José de Almeida – I.S.T.
P – 1000 Lisboa

Schweiz / Suisse / Svizzera

Comité National Suisse de la FEM
Verein Schweizerischer Maschinen-Industrieller
Kirchenweg 4
CH – 8032 Zürich

Suomi

Finnish National Committee of FEM
Federation of Finnish Metal and Engineering Industries
Eteläranta 10
SF – 00 130 Helsinki 13

Sverige

Swedish National Committee of FEM
Materialhanteringsgruppen Inom Sveriges Mekanförbund
Box 5506
S – 11 485 Stockholm