IS/ISO 8686-1 (1989): Specification For Cranes - Design Principles for Loads and Load Combination [MED 14: Cranes, Lifting Chains and Related Equipment]

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IS/ISO 8686-1 : 1989
(Reaffirmed 2006)

भारतीय मानक

भार और भार संयोजन के लिए क्रेन के डिजाईन संबंधी सिद्धान्त

भाग 1 सामान्य

Indian Standard

CRANES — DESIGN PRINCIPLES FOR LOADS AND LOAD COMBINATIONS

PART 1 GENERAL

ICS 53.020.20

BUREAU OF INDIAN STANDARDS
MANAK BHAVAN, 9 BAHADUR SHAH ZAFAR MARG
NEW DELHI 110 002

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Price Group 10

NATIONAL FOREWORD

This Indian Standard which is identical with ISO 8686-1 : 1989 'Cranes — Design principles for loads and load combinations—Part 1 : General', issued by the International Organization for Standardization (ISO), was adopted by the Bureau of Indian Standards on the recommendations of the Cranes, Lifting Chains and Its Related Equipment Sectional Committee and approval of the Heavy Mechanical Engineering Division Council.

This standard is being published in five parts. Other parts of this standard are as follows:

Part 2	Mobile cranes
Part 3	Tower cranes
Part 4	Jib cranes
Part 5	Overhead travelling cranes and portal bridge cranes

The text of ISO standard has been approved for publication as Indian Standard without deviations. Certain terminology and conventions are, however, not identical to those used in Indian Standards. Attention is particularly drawn to the following:

a) Wherever the words 'International Standard' appear referring to this standard, they should be read as 'Indian Standard'.

b) Comma (,) has been used as a decimal marker while in Indian Standards, the current practice is to use full stop (.) as a decimal marker.

In this adopted standard, reference appears to certain International Standards for which Indian Standards also exist. The corresponding Indian Standards which are to be substituted in their place are listed below along with their degree of equivalence for the editions indicated:

International Standard	*Corresponding Indian Standard*	*Degree of Equivalence*
ISO 4306-1 : 1990	IS 13473 (Part 1) : 1992 Cranes — Vocabulary : Part 1 General	Identical
ISO 4306-2 : 1985	IS 13473 (Part 2) : 1992 Cranes — Vocabulary : Part 2 Mobile cranes	Identical
ISO 4306-3 : 1991	IS 13473 (Part 3) : 1993 Cranes — Vocabulary : Part 3 Tower cranes	Identical

The concerned technical committee has reviewed the provisions of ISO 4302 : 1981 and ISO 4310 : 1981 referred in this adopted standard and has decided that they are acceptable for use in conjunction with this standard.

In reporting the results of a test or analysis made in accordance with this standard, if the final value, observed or calculated, is to be rounded off,it shall be done in accordance with IS 2:1960 'Rules for rounding off numerical values (*revised*)'.

Indian Standard

CRANES — DESIGN PRINCIPLES FOR LOADS AND LOAD COMBINATIONS

PART 1 GENERAL

1 Scope

This part of ISO 8686 establishes general methods for calculating loads, and principles to be used to select load combinations for proofs of competence for the structural and mechanical components of cranes as defined in ISO 4306-1.

It is based on rigid-body kinetic analysis and elasto-static analysis but it expressly permits the use of more advanced methods (calculations or tests) to evaluate the effects of loads and load combinations, and the values of dynamic load factors, where it can be demonstrated that these provide at least equivalent levels of competence.

This part of ISO 8686 is intended for two distinct kinds of application :

a) It provides the general form, content and ranges of parameter values for more specific standards to be developed for individual lifting appliance types.

b) It provides a framework for agreement on loads and load combinations between a designer or manufacturer and an appliance purchaser for those types of lifting appliances where specific standards do not exist.

When applying this part of ISO 8686 to different types of lifting appliance, operating in the same service and environmental conditions, equivalent resistance to failure should be sought.

2 Normative references

The following standards contain provisions which, through reference in this text, constitute provisions of this part of ISO 8686. At the time of publication, the editions indicated were valid. All standards are subject to revision, and parties to agreements based on this part of ISO 8686 are encouraged to investigate the possibility of applying the most recent editions of the standards listed below. Members of IEC and ISO maintain registers of currently valid International Standards.

ISO 4302 : 1981, *Cranes — Wind load assessment.*

ISO 4306 (all published parts), *Lifting appliances — Vocabulary.*

ISO 4310 : 1981, *Cranes — Test code and procedures.*

3 Definitions

For the purposes of ISO 8686, the definitions given in ISO 4306, together with the following, apply.

3.1 loads: External or internal actions in the form of forces, displacements or temperature, which cause stresses in the structural or mechanical components of the lifting appliance.

3.2 kinetic analysis of rigid bodies: The study of the movement and the inner forces of systems modelled by elements that are assumed to be non-elastic.

3.3 kinetic analysis for elastic bodies: The study of the relative elastic displacements (distortion), movement and the inner forces of systems modelled by elements that are assumed to be elastic.

4 Symbols

The main symbols used in this part of ISO 8686 are given in table 1.

Table 1 — Main symbols

Symbol	Description	Reference
ϕ	Factors for dynamic effects	Various
ϕ_1	Factors for hoisting and gravity effects acting on the mass of the lifting appliance	6.1.1
a	Term used in determining the value of ϕ_1	6.1.1
ϕ_2	Factor for hoisting a grounded load	6.1.2.1
ϕ_3	Factor for dynamic effects of sudden release of part of load	6.1.2.3
ϕ_4	Factor for dynamic effects of travelling on an uneven surface	6.1.3.2
ϕ_5	Factor for dynamic loads arising from acceleration of crane drives	6.1.4
ϕ_6	Factor for effects of dynamic load tests	6.3.2
ϕ_7	Factor for elastic effects arising from collision with buffers	6.3.3
HC_1 to HC_4	Hoisting classes assigned to lifting appliances	6.1.2.1
β_2	Factor assigned to hoisting class	6.1.2.1
β_3	Term used in determining the value of ϕ_3	6.1.2.3
v_h	Steady hoisting speed, in metres per second	6.1.2.2
F_x, F_{x2}, F_{x4}	Buffer forces	6.3.3
γ_{fA}, γ_{fB}, γ_{fC}	Coefficients for calculating allowable stresses	7.3.2
γ_p	Partial load coefficient	7.3.3
γ_m	Resistance coefficient	Annex A
γ_n	Coefficient for high-risk applications	7.3.6
m	Mass of the load	6.1.2.3, 6.3.1
$\eta m = m - \Delta m$	Mass of that part of the hoist load remaining suspended from the appliance	6.3.1
NOTE — Further symbols are used in the annexes and are defined therein.		

5 General

5.1 The objective of proof of competence calculations carried out in accordance with this part of ISO 8686 is to determine mathematically that a lifting appliance will be competent to perform in practice when operated in compliance with the manufacturer's instructions.

The basis for such proof against failure (by yielding, elastic instability or fatigue, for example) is the comparison between calculated stresses induced by loads and the corresponding calculated strengths of the constituent structural and mechanical components of the lifting appliance.

Proof against failure may also be required in respect of overturning stability. Here, the comparison is made between the calculated overturning moments induced by loads and the calculated resistance to overturning provided by the lifting appliance. In addition, there may be limitations on forces that are necessary to assure the stability and/or to avoid unwanted displacement of portions of the appliance or of the appliance itself, for example the jib support ropes becoming unloaded or the appliance sliding.

The effects of differences between actual and ideal geometry of mechanical and structural systems (for example the effect of tolerances, settlements, etc.) shall be taken into account. However, they shall be included specifically in proof of competence calculations only where, in conjunction with applied loads, they may cause stresses that exceed specified limits.

5.2 There are two general approaches to structural design or proof of competence calculations:

a) The allowable stress method, where the design stresses induced by combined loads are compared with allowable stresses established for the type of member or condition being examined. The assignment of allowable stress is made on the basis of service experience with consideration for protection against failure due, for example, to yielding, elastic instability or fatigue.

b) The limit state method, where partial load factors are used to amplify loads before they are combined and compared with the limit states imposed, for example, by yielding or elastic instability. The partial load coefficient for each load is established on the basis of probability and the degree of accuracy with which the load can be determined. Limit state values comprise the characteristic strength of the member reduced to reflect statistical variations in its strength and geometric parameters.

The limit state method generally permits more efficient design because it takes into account greater certainty in determining appliance mass and lesser certainty in values selected to reflect applied loads.

Annex A gives a more detailed description of the application of the allowable stress method and the limit state method.

5.3 To calculate stresses from applied loads, an appropriate model of the appliance shall be used. Under the provisions of this part of ISO 8686, loads which cause time variant load effects are assessed as equivalent static loads from experience, experiments, or by calculation. A rigid-body kinetic analysis can be used with dynamic factors to estimate the forces necessary to simulate the response of the elastic system. Alternatively, either elasto-kinetic analysis or field measurements can be carried out, but to reflect the operating regime, a realistic model of the actions of the appliance operator may be required.

For both the allowable stress method and the limit state method, and for considerations of stability and displacements, loads, load combinations, load factors, allowable stresses and limit states should be assigned either on the basis of experience, with consideration of other International Standards or, if applicable, on the basis of experimental or statistical data. The parameters used in this part of ISO 8686 are considered to be deterministic.

Where a specific loading cannot occur (for example wind loading on an appliance used indoors) then that loading can be ignored in the proof of competence calculations. Similarly, loadings can be modified when they result from

a) conditions prohibited in the appliance instructions;

b) features not present in the design;

c) conditions which are prevented or suppressed by the design of the appliance.

If a probabilistic proof of competence calculation is used, the relevant conditions, particularly the acceptable probability of failure, shall be stated.

6 Loads and applicable factors

This clause gives loads and ranges of values for the factors used in proof of competence calculations when determining load effects.

Individual values for specific types of appliance, selected from these ranges, will be found in the parts of this International Standard covering those appliances.

The loads acting on a lifting appliance are divided into the categories of regular, occasional, exceptional, and miscellaneous. Individual loads are considered only when and if they are relevant to the appliance under consideration or to its usage:

a) Regular loads, occurring during normal operation, shall be considered in proof of competence calculations against failure by yielding, elastic instability and, when applicable, against fatigue. They result from gravity and from acceleration or deceleration produced by drives and brakes acting on the masses of the lifting appliance and the hoist load, as well as from displacements.

b) Occasional loads and effects which occur infrequently are usually neglected in fatigue evaluations. They include loads induced by in-service wind, snow and ice, temperature and skewing.

c) Exceptional loads and their effects are also infrequent and may likewise usually be excluded from fatigue consideration. They include loads caused by testing, out-of-service wind, buffer forces and tilting, as well as from emergency cut-out, failure of drive components, and external excitation of the lifting appliance foundation.

d) Miscellaneous loads include erection and dismantling loads as well as loads on platforms and means of access.

The category in which a load is placed is not an indication of the importance or criticality of that load. For example, erection and dismantling loads, although in the last category, shall be given particular attention as a substantial portion of accidents occur during those phases of operation.

6.1 Regular loads

6.1.1 Hoisting and gravity effects acting on the mass of the lifting appliance

The mass of the appliance includes those components which are always in place during operation, except for the payload itself (see 6.1.2). For some appliances or applications, it may be necessary to add mass to account for encrustation of materials, such as coal or similar dust, which build up on the appliance or its parts.

The gravitational force induced by the mass of the appliance (dead weight) shall be multiplied by the factor ϕ_1, where $\phi_1 = 1 \pm a$, $0 < a < 0{,}1$. In this way the vibrational excitement of the lifting appliance structure, when lifting the gross load off the ground, is taken into account. There are always two values for the factor in order to reflect both the upper and lower reaches of the vibrational pulses.

The factor ϕ_1 shall be used in the design of the appliance structure and its supports; in some cases, both values of the factor shall be applied in order to find the most critical loadings in members and components.

Annex C gives a general comment on the application of ϕ factors.

6.1.2 Inertial and gravity effects acting vertically on the gross load

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

6.1.2.1 Hoisting class

For the purposes of this clause, lifting appliances are assigned to hoisting classes HC_1 to HC_4 according to their dynamic characteristics. The hoisting classes of appliances are given in table 2 and shall be selected on the basis of experience. Corresponding values of β_2 and ϕ_2 are also given in table 2 and illustrated in figure 1.

The selection of the hoisting class depends on the particular type of lifting appliance and is dealt with in the other parts of this International Standard.

Equally, values of ϕ_2 can be determined by experiment or analysis without reference to hoisting class.

Table 2 — Values of β_2 and ϕ_2

Hoisting class of appliance	β_2	ϕ_2	
		$\phi_{2,\,min}$	$\phi_{2,\,max}$
HC_1	0,2	1	1,3
HC_2	0,4	1,05	1,6
HC_3	0,6	1,1	1,9
HC_4	0,8	1,15	2,2

6.1.2.2 Hoisting an unrestrained grounded load

In the case of hoisting an unrestrained grounded load, the dynamic effects of transferring the load from the ground to the lifting appliance shall be taken into account by multiplying the gravitational force due to the mass of the gross load by a factor ϕ_2. (See figure 1.)

NOTE — The dynamic effects covered by this clause occur when the drive comes up to speed before the lifting attachment engages the load and are the result of the build-up of kinetic energy and the drive torque.

The factor ϕ_2 shall be taken as follows:

$$\phi_2 = \phi_{2,\,min}, \text{ for } v_h < 0{,}2 \text{ m/s}$$

$$\phi_2 = \phi_{2,\,min} + \beta_2 (v_h - 0{,}2), \text{ for } v_h > 0{,}2 \text{ m/s}$$

where

v_h is the steady hoisting speed, in metres per second, related to the lifting attachment, derived from the steady rotational speed of the unloaded motor or engine;

β_2 is a factor assigned to the hoisting class (see table 2);

$\phi_{2,\,min}$ is given in table 2 for the hoisting class.

Where the hoist drive control system ensures the use of a steady creep speed, this speed only shall be taken into account to cover normal operation in determining the value of ϕ_2.

Where this is not the case, two conditions shall be considered by taking a value of ϕ_2 to cover normal operation, as in 6.1.2.2.1, and a value of $\phi_{2,\,max}$ to cover exceptional occurrences, as in 6.1.2.2.2.

6.1.2.2.1 For normal operation

a) Where a steady creep speed can be selected by the crane driver, this speed shall be used in determining the value of ϕ_2.

b) Where a stepless variable speed control is provided or such control can be exercised by the crane driver, the value of $\phi_{2,\,min}$ for the appropriate hoisting class shall be selected from figure 1.

6.1.2.2.2 For exceptional occurrences

For appliances with control of type a) as in 6.1.2.2.1, the value of $\phi_{2,\,max}$ shall be based on a value of v_h derived from the maximum nominal speed of the unloaded motor or engine.

For appliances with control of type b) as in 6.1.2.2.1, the value of $\phi_{2,\,max}$ for the hoisting class shall be based on a value of v_h derived from a value of not less than 0,5 times the maximum nominal speed of the unloaded motor or engine.

Annex C gives a general comment on the application of ϕ factors.

6.1.2.3 Effects of sudden release of part of payload

For lifting appliances that release or drop part of the payload as a normal working procedure, such as when grabs or magnets are used, the peak dynamic effect on the appliance can be simulated by multiplying the payload by the factor ϕ_3 (see figure 2).

The value of ϕ_3 is given by

$$\phi_3 = 1 - \frac{\Delta m}{m} (1 + \beta_3)$$

where

Δm is the released or dropped part of the payload;

m is the mass of the payload;

β_3 = 0,5 for appliances equipped with grabs or similar slow-release devices,

= 1 for appliances equipped with magnets or similar rapid-release devices.

Annex C gives a general comment on the application of ϕ factors.

6.1.3 Loads caused by travelling on an uneven surface

6.1.3.1 Lifting appliances travelling on or off roadways

The effects of travelling, with or without load, on or off roadways, depend on the appliance configuration (mass distribution), the elasticity of the appliance and/or its suspension, the travel speed and on the nature and condition of the travel surface. The dynamic effects shall be estimated from experience, experiment, or by calculation using an appropriate model for the appliance and the travel surface.

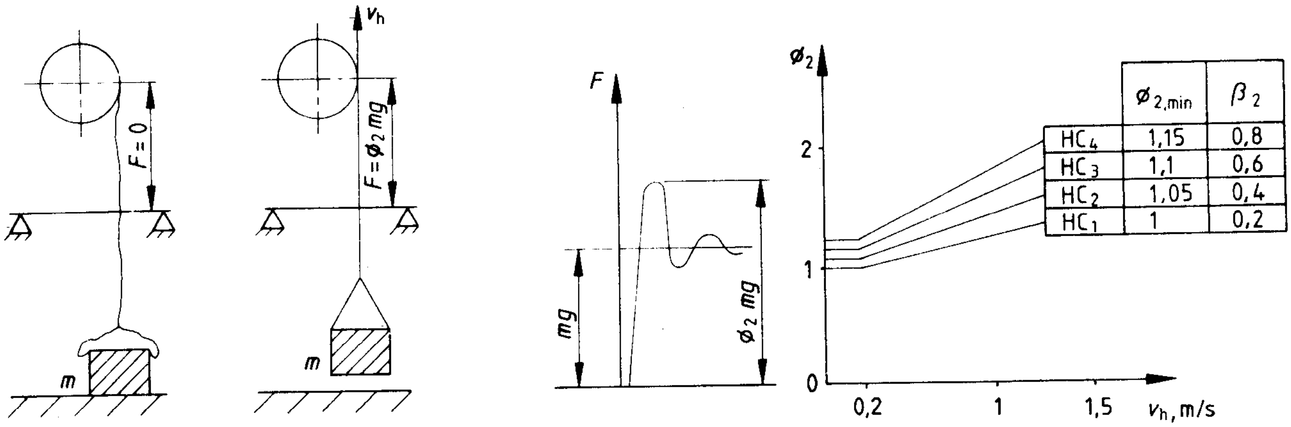


Figure 1 — Factor ϕ_2

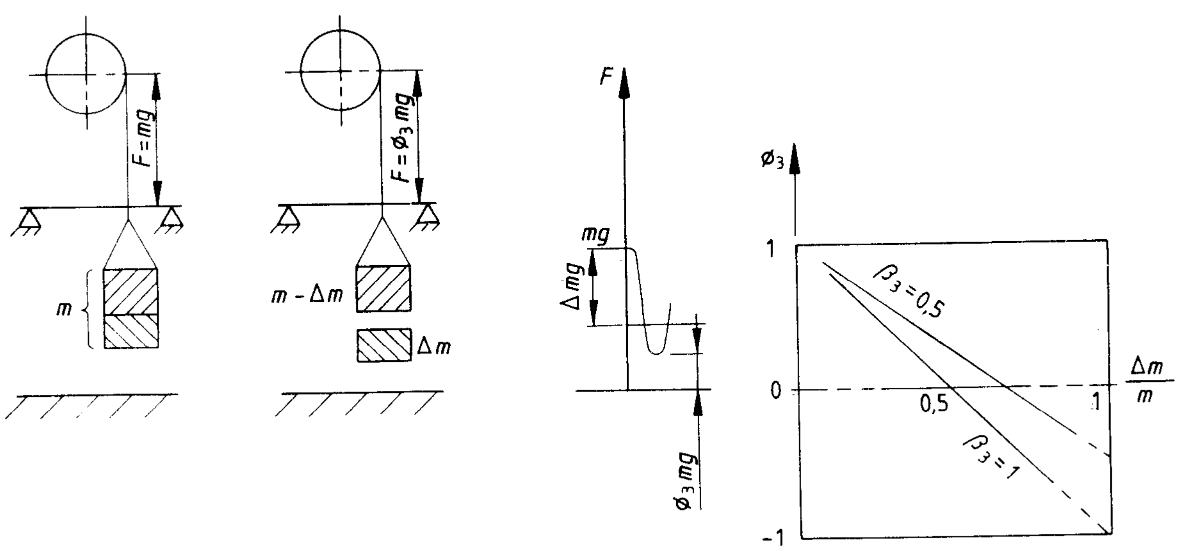


Figure 2 — Factor ϕ_3

6.1.3.2 Lifting appliances travelling on rails

The effects of travelling with or without load on rail tracks having geometric or elastic characteristics that induce accelerations at the wheels of the appliances depend on the appliance configuration (mass distribution, elasticity of the appliance and/or its suspension), travel speed and wheel diameter. They shall be estimated from experience, experiment, or by calculation using an appropriate model for the appliance and the track.

The induced accelerations may be taken into account by multiplying the gravitational forces due to the masses of the lifting appliance and gross load by a factor ϕ_4. International Standards for individual types of appliance may specify tolerances for rail tracks and indicate conditions within which the value of ϕ_4 may be taken as 1.

Annex C gives a general comment on the application of ϕ factors.

Annex D gives an example of a model for estimating the value of ϕ_4 to take account of the vertical accelerations induced at the wheels of an appliance travelling on rail tracks with non-welded steps or gaps.

6.1.4 Loads caused by acceleration of all crane drives including hoist drives

Loads induced in a lifting appliance by accelerations or decelerations caused by drive forces may be calculated using rigid-body kinetic models that take into account the geometric properties and mass distribution of the lifting appliance drive and, where applicable, resulting inner frictional losses. For this purpose, the gross load is taken to be fixed at the top of the jib or immediately below the crab.

A rigid-body analysis does not directly reflect elastic effects. To allow for these, the change in drive force (ΔF), inducing either the acceleration or deceleration, may be multiplied by a factor ϕ_5 and algebraically added to the force present before the acceleration or deceleration takes place. This amplified force is then applied to the components exposed to the drive force and, where applicable, to the appliance and the gross load as well. (See figure 3.)

The range of values for ϕ_5 is $1 < \phi_5 < 2$. The value used depends on the rate of change of the drive or braking force and on the mass distribution and elastic properties of the system. In general, lower values correspond to systems in which forces change smoothly and higher values to those in which sudden changes occur.

For centrifugal forces, ϕ_5 may be taken as 1.

Where a force that can be transmitted is limited by friction or by the nature of the drive mechanism, the limited force and a factor ϕ_5 appropriate to that system shall be used.

Annex C gives a general comment on the application of ϕ factors.

Annex E gives an example of a determination of the loads caused by acceleration of a bridge crane having unsynchronized travel gear and non-symmetrical load distribution.

6.1.5 Loads induced by displacements

Account shall be taken of loads arising from displacements included in the design such as those resulting from pre-stressing and those within the limits necessary to initiate response of skewing and other compensating control systems.

Other loads to be considered include those that can arise from displacements that are within defined limits such as those set for the variation in the gauge between rails or uneven settlement of supports.

6.2 Occasional loads

6.2.1 Climatic effects

6.2.1.1 In-service wind

Loads due to in-service wind shall be calculated in accordance with ISO 4302.

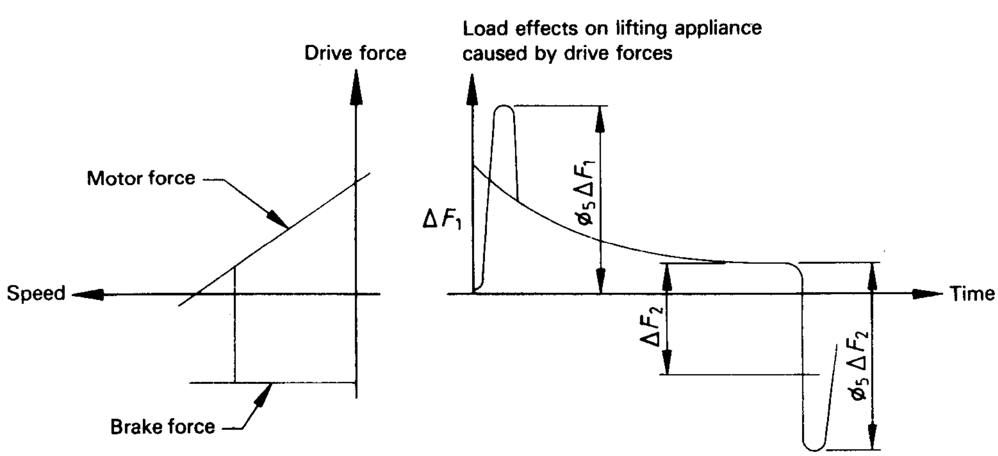


Figure 3 — Factor ϕ_5

6.2.1.2 Snow and ice loads

Where relevant, snow and ice loads shall be taken into account. The increased wind exposure surfaces due to encrustation shall be considered.

6.2.1.3 Loads due to temperature variation

Loads caused by the restraint of expansion or contraction of a component due to local temperature variation shall be taken into account.

6.2.2 Loads caused by skewing

This sub-clause covers skewing loads that occur at the guidance means (such as guide rollers or wheel flanges) of a guided, wheel-mounted appliance while it is travelling or traversing in steady-state motion. These loads are induced by guidance reactions which force the wheels to deviate from their free-rolling, natural travelling direction. Similar loads induced by acceleration acting on asymmetrical mass distribution and which can also cause the appliance to skew are taken into account under 6.1.4.

Skewing loads as defined above are usually taken as occasional loads but their frequency of occurrence varies with the type, configuration and service of the appliance. In individual cases, the frequency of occurrence will determine whether they are taken as occasional or regular loads. Guidance for establishing the magnitude of skewing loads and the category into which they are placed is given in the parts of this International Standard covering those individual appliance types.

Annex F gives an example of a method for analysing skewing loads on a rigid lifting appliance structure travelling at a constant speed. For appliances with structures that are not rigid in respect of applied skewing forces or that have specially controlled travel guidance, appropriate models shall be used which take the system properties into account.

6.3 Exceptional loads

6.3.1 Out-of-service wind conditions

When considering out-of-service wind conditions, the gravitational force on that part of the mass of the hoist load remaining suspended from the appliance, ηm, shall be taken into account:

$$\eta m = m - \Delta m$$

where

$m - \Delta m$ is that part of the gross load remaining suspended from the appliance,

m is the mass of the gross load.

Wind loads shall be calculated in accordance with ISO 4302.

6.3.2 Test loads

The values of test loads shall be in accordance with ISO 4310.

Where values for dynamic or static test loads are required that are above the minimum given in ISO 4310, proof of competence calculations for these test conditions may be necessary. In this case the dynamic test load shall be multiplied by a factor ϕ_6, given by

$$\phi_6 = 0,5\,(1 + \phi_2)$$

where ϕ_2 is calculated in accordance with 6.1.2.

Annex C gives a general comment on the application of ϕ factors.

6.3.3 Buffer forces

Where buffers are used, the forces on the crane structure arising from collision with them shall be calculated from the kinetic energy of all relevant parts of the appliance moving in general at 0,7 to 1 times the nominal speed. Lower values may be used where they are justified by special considerations such as the existence of an automatic control system of demonstrable reliability for retarding the motion or where there would be limited consequences in the event of a buffer impact.

The calculation may be based on a rigid body model. The actual behaviour of the crane and buffer system shall be taken into account.

Where the lifting appliance or component is restrained against rotation, for example by guide rails, the buffer deformations may be assumed to be equal, in which case if the buffer characteristics are similar, the buffer forces will be equal. This case is illustrated in figure 4 a) in which

$$F_{x2} = F_{x4} = \hat{F}_x/2$$

Where the appliance or component is not restrained against rotation, the buffer forces shall be calculated taking into account the distribution of the relevant masses and the buffer characteristics. This case is illustrated in figure 4 b).

The resulting forces as well as the horizontal inertia forces in balance with the buffer forces shall be multiplied by a factor ϕ_7 to account for elastic effects which cannot be evaluated using a rigid body analysis. ϕ_7 shall be taken as 1,25 in the case of buffers with linear characteristics (for example springs) and as 1,6 in the case of buffers with rectangular characteristics (for example hydraulic constant force buffers). For buffers with other characteristics other values justified by calculation or by test shall be used. (See note 2 and figure 5.)

NOTES

1 In calculating buffer forces, the effects of suspended loads that are unrestrained horizontally (free to swing) should not be taken into account.

2 Intermediate values of ϕ_7 can be estimated as follows:

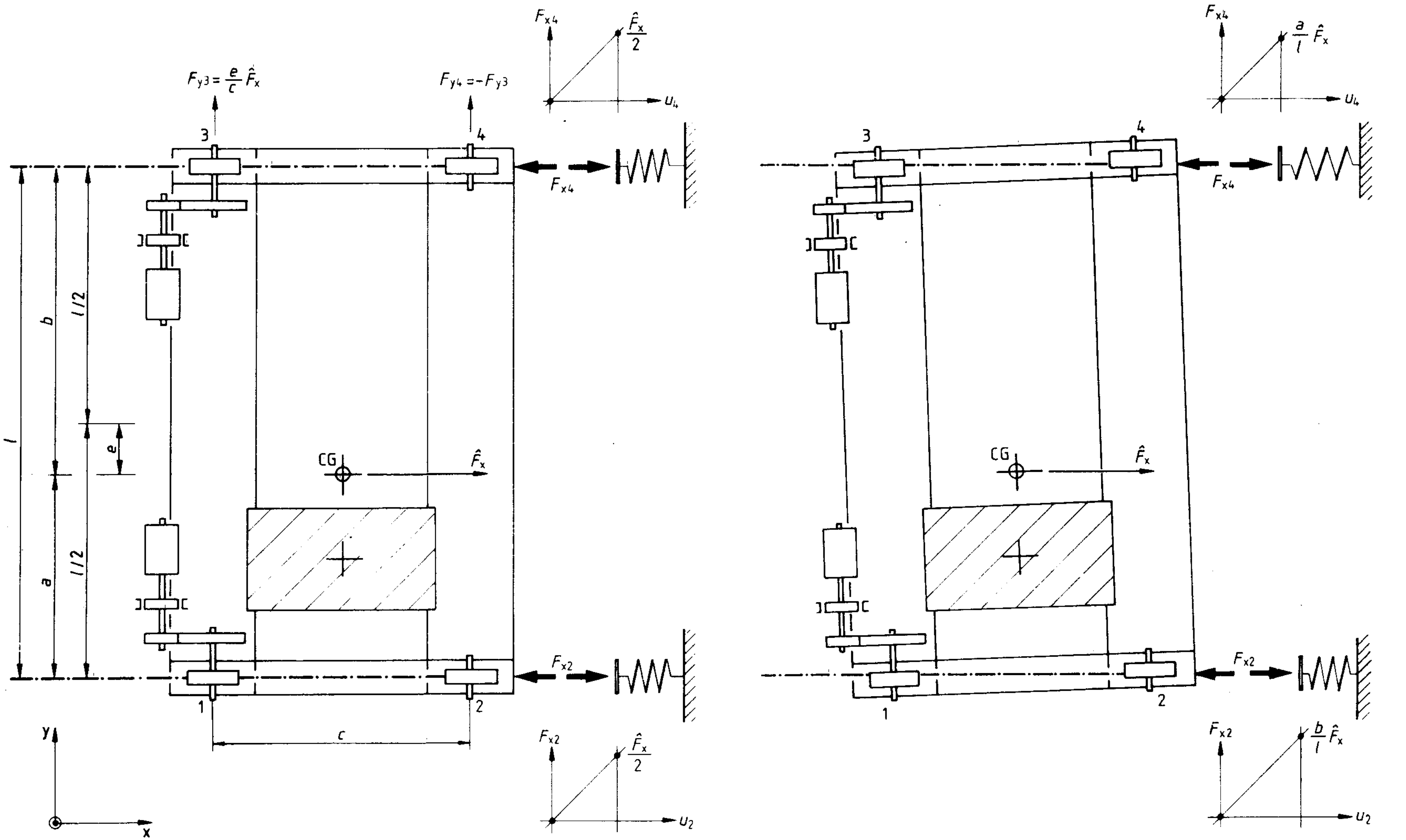
$\phi_7 = 1,25$ if $0 < \xi < 0,5$

$\phi_7 = 1,25 + 0,7\,(\xi - 0,5)$ if $0,5 < \xi < 1$

6.3.4 Tilting forces

If an appliance with horizontally restrained load can tilt when it, its load or lifting attachment collides with an obstacle, the resulting static forces shall be determined.

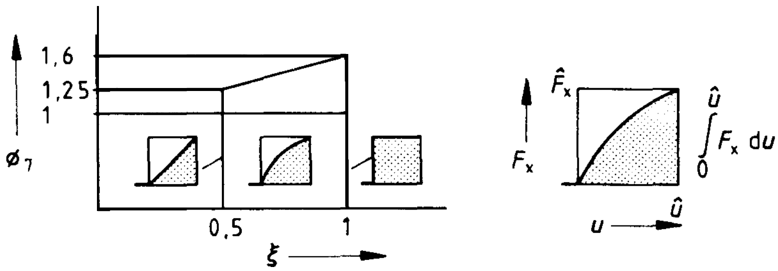
If a tilted appliance can fall back into its normal position uncontrolled, the resulting impact on the supporting structure shall be taken into account.



a) **Appliance horizontally guided by rails** ($u_2 = u_4$)

b) **Appliance not restrained against rotation** ($F_{y3} = F_{y4} = 0$)

Figure 4 — Examples of buffer forces and buffer deformation (four-wheel bridge crane shown)



$$\xi = \frac{1}{\hat{P}a}\int_0^{u} F_x \, du \text{ = relative buffer energy}$$

For a buffer with linear characteristics: $\xi = 0.5$

For a buffer with rectangular characteristics : $\xi = 1$

Figure 5 — Factor ϕ_7

6.3.5 Loads caused by emergency cut-out

Loads caused by emergency cut-out shall be evaluated in accordance with 6.1.4, taking into account the most unfavourable state of drive (i.e. the most unfavourable combination of acceleration and loading) at the time of cut-out. The value of the factor ϕ_5 shall be chosen from the range $1.5 < \phi_5 < 2$.

6.3.6 Loads caused by failure of mechanism or components

Where protection is provided by emergency brakes in addition to service brakes, failure and emergency brake activation shall be assumed to occur under the most unfavourable condition.

Where mechanisms are duplicated for safety reasons, failure shall be assumed to occur in any part of either system.

In both these cases, resulting loads shall be evaluated in accordance with 6.1.4, taking into account any impacts resulting from the transfer of forces.

6.3.7 External excitation of the lifting appliance foundation

Examples of lifting appliance foundation excitation are earthquakes or wave-induced movements.

Loads caused by such excitations shall be considered only when they constitute a significant risk.

NOTE — Special requirements given in regulations or specifications may apply.

8.4 Miscellaneous loads

6.4.1 Loads due to erection, dismantling and transport

The loads acting at each stage of the erection and dismantling process shall be taken into account, including those arising from a wind speed of 8,3 m/s or greater. Higher values may be specified for individual types of crane covered by the other parts of this International Standard. Theyshall be combined in accordance with 7.2.

In some cases it may also be necessary to take account of loads occurring during transport.

6.4.2 Loads on platforms and other means provided for access

The loads are considered to be local, acting only on the facilities themselves and on their immediate supporting members.

The following loads shall be taken into account:

3 000 N where materials can be deposited;

1 500 N on meansprovided for access only;

not less than 300 N horizontally on railings, depending on location and use.

7 Principles of choice of load combinations

7.1 Basic considerations

Loads shall be combined to determine the stresses an appliance will experience, during normal operation, as simulated by an elastostatic calculation. To achieve this,

a) the appliance is taken in its most unfavourable attitude and configuration while the loads are assumed to act in magnitude, position and direction causing unfavourable stresses at the critical points selected for evaluation on the basis of engineering considerations, and

b) conservatively, loads can be combined at the values defined in this part of ISO 8686 or, when appropriate, they can be combined with some loads factored to more closely reflect loading conditions actually found in practice.

The load combinations appropriate to individual types of appliances shall be in accordance with the principles set out in 7.1.1 to 7.2 and in table 3.

Table 3 — Loads and load combinations

1	2			3					4						5									6
Categories of loads	Loads, f_i			Load combinations A					Load combinations B						Load combinations C									Line No.
				Partial load factors γ_p	A1	A2	A3	A4	Partial load factors γ_p	B1	B2	B3	B4	B5	Partial load factors γ_p	C1	C2	C3	C4	C5	C6	C7	C8	
Regular (see 6.1)	Gravitation, acceleration, impacts	1) Mass of the lifting appliance		γ_{pA1}	ϕ_1	ϕ_1	1	—	γ_{pB1}	ϕ_1	ϕ_1	1	—	—	γ_{pC1}	ϕ_1	1	ϕ_1	1	1	1	1	1	1
		2) Mass of the gross load		γ_{pA2}	ϕ_2	ϕ_3	—	—	γ_{pB2}	ϕ_2	ϕ_3	—	—	—	γ_{pC2}	ϕ_2	η	—	1	1	1	1	1	2
		3) Masses of lifting appliance and hoist load, travelling on an uneven surface		γ_{pA3}	—	—	—	ϕ_4	γ_{pB3}	—	—	—	ϕ_4	ϕ_4	γ_{pC3}	—	—	—	—	—	—	—	—	3
	Acceleration from drives	4) Masses of lifting appliance and gross load	a) Hoist drives excluded	γ_{pA4}	ϕ_5	ϕ_5	—	—	γ_{pB4}	ϕ_5	ϕ_5	—	—	—	γ_{pC4}	—	—	ϕ_5	—	—	—	—	—	4
			b) Hoist drives included		—	—	ϕ_5	ϕ_5		—	—	ϕ_5	ϕ_5	—		—	—	—	—	—	—	—	—	5
	Displacements	5) See 6.1.5		γ_{pA5}	1	1	1	1	γ_{pB5}	1	1	1	1	1	γ_{pC5}	1	1	1	1	1	1	1	1	6
Occasional (see 6.2)	Effects of climate	1) In-service wind loads							γ_{pB6}	1	1	1	1	1	γ_{pC6}	—	—	—	—	—	—	—	—	7
		2) Snow and ice loads							γ_{pB7}	1	1	1	1	1	γ_{pC7}	—	1	—	—	—	—	—	—	8
		3) Temperature variations							γ_{pB8}	1	1	1	1	1	γ_{pC8}	—	1	—	—	—	—	—	—	9
	Skewing	4) See 6.2.2							γ_{pB9}	—	—	—	—	1	γ_{pC9}	—	—	—	—	—	—	—	—	10
Exceptional (see 6.3)	1) Hoisting a grounded load														γ_{pC10}	ϕ_2	—	—	—	—	—	—	—	11
	2) Out-of-service wind loads														γ_{pC11}	—	1	—	—	—	—	—	—	12
	3) Test loads														γ_{pC12}	—	—	ϕ_6	—	—	—	—	—	13
	4) Buffer forces														γ_{pC13}	—	—	—	ϕ_7	—	—	—	—	14
	5) Tilting forces														γ_{pC14}	—	—	—	—	1	—	—	—	15
	6) Emergency cut-out														γ_{pC15}	—	—	—	—	—	ϕ_5	—	—	16
	7) Failure of mechanism														γ_{pC16}	—	—	—	—	—	—	ϕ_5	—	17
	8) Excitation of the lifting appliance foundation														γ_{pC17}	—	—	—	—	—	—	—	1	18
	Strength coefficient γ_r			γ_{fA}					γ_{fB}						γ_{fC}									19

Load combinations

A1 and B1	Lifting appliances under normal service conditions, hoisting and depositing loads, without in-service wind and loads from other climatic effects (A1), and with in-service wind and loads from other climatic effects (B1). In general, hoisting, travelling, slewing and luffing movements are possible simultaneously. The various loads resulting from these movements shall be combined to correspond with the specified working conditions.
A2 and B2	Lifting appliances under normal service conditions, sudden releasing of a part of the hoist load, without in-service wind and loads from other climatic effects (A2), and with in-service wind and loads from other climatic effects (B2). Drive forces shall be combined as in A1 and B1.
A3 and B3	Lifting appliances under normal service conditions, accelerating the suspended load, without in-service wind and loads from other climatic effects (A3), and with in-service wind and loads from other climatic effects (B3). Other drive forces shall be combined as in A1 and B1.
A4 and B4	Lifting appliances under normal service conditions, travelling on an uneven surface or track, without in-service wind and loads from other climatic effects (A4), and with in-service wind and loads from other climatic effects (B4). Drive forces shall be combined as in A1 and B1.
B5	Lifting appliances under normal service condition, travelling on an uneven surface at constant speed and skewing, with in-service wind and loads from other climatic effects.
C1	Lifting appliances under in-service conditions hoisting a grounded load under the exceptional circumstance applying to ϕ_2 in 6.1.2.2.2.
C2	Lifting appliances under out-of-service conditions, including out-of-service wind and loads from other climatic effects.
C3	Lifting appliances under test conditions. Drive forces shall be combined as in A1 and B1.
C4 to C8	Lifting appliances with gross load in combination with loads such as buffer forces (C4), tilting forces (C5), emergency cut-out (C6), failure of mechanism (C7), excitation of the lifting appliance foundation (C8).

NOTE — For erection and dismantling loads, see 7.2.

7.1.1 Basic load combinations

Basic load combinations are given in table 3. In general, load combinations A cover regular loads, load combinations B cover regular loads combined with occasional loads, and load combinations C cover regular loads combined with occasional and exceptional loads.

7.2 Load combinations during erection, dismantling and transport

Each stage of the erection and dismantling process shall be considered, taking into account the appropriate loads and load combinations which shall be as specified in the parts of this International Standard covering each type of crane. Proof of competence calculations shall be carried out for each instance of significant loading of a member or component.

In some cases it may also be necessary to take account of load occurring during transport.

7.3 Application of table 3

7.3.1 General

The masses in column 2, lines 1 to 3, shall be multiplied by gravitational acceleration g, and masses in column 2, lines 4 and 5, by the appropriate accelerations. The resulting or given loads shall be multiplied by the corresponding factors or by 1.

Each combination of loads shall be applied in accordance with 7.1.

7.3.2 Allowable stress method

The allowable stresses for load combinations A, B and C shall be determined by dividing the appropriate specified strength of the material, element, component or connection (for example the stress at yielding, buckling or limit of elastic stability) by γ_{fA}, γ_{fB} or γ_{fC}.

Values for coefficients γ_{fA}, γ_{fB} and γ_{fC} for this method are given in table B.1 (in annex B).

7.3.3 Limit state method

The various loads shall be multiplied by the partial load coefficients γ_p depending on the type of load and load combinations A, B or C before being applied to the model.

The partial load coefficients γ_p to be selected are listed in columns 3, 4 and 5.

Ranges of values of partial load coefficient γ_p are given in table B.1.

7.3.4 Elastic displacements

In some instances, elastic displacements can render an appliance unfit to perform its intended duties, can affect stability, or may interfere with the proper functioning of mechanisms. In such instances, consideration of displacements shall be part of the proof of competence calculations and, where appropriate, calculated displacements shall be compared with established limits.

7.3.5 Proofs of fatigue strength

The effects of fatigue shall be considered. Where proofs of fatigue strength are found to be necessary they shall be carried out in accordance with the principles set down in 7.1. In general, load combinations A1, A2, A3 and A4 (regular loads) shall be taken into account.

In some applications it may be necessary to consider also occasional loads such as in-service wind, skewing and exceptional loads such as test loads and excitation of the lifting appliance foundation (for example wave effects).

7.3.6 High-risk applications

In special cases where the human or economic consequences of failure are exceptionally severe (for example ladle cranes or cranes for nuclear applications), increased reliability shall be obtained by the use of a risk coefficient $\gamma_n > 1$ the value of which shall be selected according to the requirements of the particular application.

Using the allowable stress method, the allowable stresses shall be divided by the coefficient. Using the limit state method, the loads shall be multiplied by γ_n. See annex A.

Annex A
(normative)

Application to the allowable stress method and the limit state method of design

(see clause 5)

A.1 Introduction

The principles set out in this part of ISO 8686 for determining the loads and load combinations to be taken into account in proof of competence calculations are applicable to both the allowable stress method and the limit state method of design. This annex describes their application in general terms.

A.2 Allowable stress method

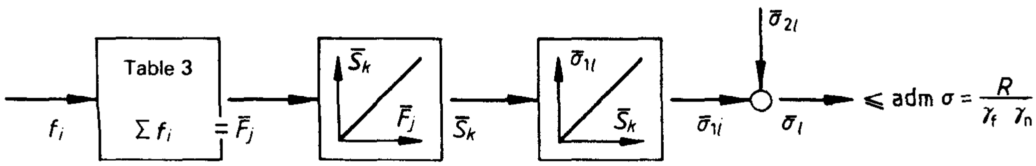
Individual specified loads, f_i, are calculated and amplified where necessary using the applicable factors ϕ. They are then combined according to the load combination under consideration from table 3. The combined load, $\bar{F}_j$, is used to determine the resulting load effects, $\bar{S}_k$, i.e. the inner forces and moments in members or the forces on supports.

The stresses, $\bar{\sigma}_{1l}$, due to the action of the load effects on a particular element or component are calculated and combined with any stresses, $\bar{\sigma}_{2l}$, resulting from local effects. The resulting design stress $\bar{\sigma}_l$ should be compared with an appropriate allowable value of adm σ.

Admissible stresses are obtained by dividing the specified strengths R of the material, such as the stresses corresponding to the yield point, limit of elastic stability or fatigue strength, by a coefficient γ_f, specified in table 3 according to the basic load combination (see 7.1.1), and, where appropriate, by a risk coefficient γ_n (see 7.3.5).

Special care is required to ensure a valid proof of competence when the allowable stress method is applied to cases where internal forces are not linearly proportional to the loads producing them or critical values of stress result from the combination of independently varying loads which give stresses of opposite signs.

A flow chart illustrating the allowable stress method of design is shown in figure A.1.



f_i	is the load i on the element or component
$\bar{F}_j$	is the load combination j
$\bar{S}_k$	are the load effects in section k of members or supporting parts, such as inner forces and moments resulting from load combination $\bar{F}_j$
$\bar{\sigma}_{1l}$	are the stresses in the particular element l as result of load effects $\bar{S}_k$
$\bar{\sigma}_{2l}$	are the stresses in the particular element l arising from local effects
$\bar{\sigma}_l$	is the resulting design stress in the particular element l
R	is the specified strength or characteristic resistance of the material, particular element or connection, such as the stress corresponding to the yield point, limit of elastic-stability or fatigue strength (limit states)
adm σ	are the allowable stresses
γ_f	are the coefficients applied to the specified strength according to the load combination under consideration
γ_n	is the risk coefficient, where applicable

Figure A.1 — Typical flow chart of the allowable stress method

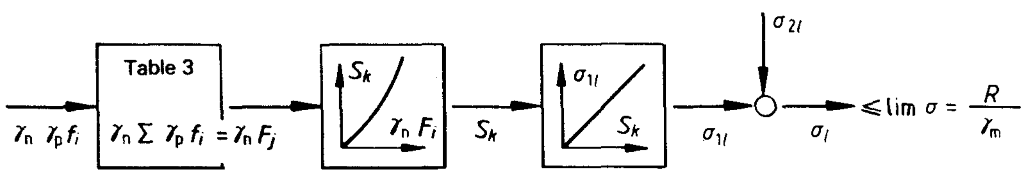
A.3 Limit state method

Individual specified or characteristic loads, f_i, are calculated and amplified where necessary using the factors ϕ, multiplied by the appropriate partial load coefficients γ_p. They are then combined according to the load combination under consideration to give the combined load $\bar{F}_j$. Factors ϕ and partial load coefficients γ_p for individual loads are given in table~B.1.

Where appropriate, the risk coefficient γ_n is applied to the combined load $\bar{F}_j$ (see 7.3.6) to give the design load, $\gamma_n \bar{F}_j$. Design load effects, $\bar{S}_k$, are determined from the design load. The stresses, σ_{1l}, due to the action of the load effects on a particular element or component are calculated and combined with any stresses, σ_{2l}, resulting from local effects which have also been calculated using the appropriate load coefficients.

The resulting design stress $\bar{\sigma}_l$ should be compared with an appropriate limit value, lim $\bar{\sigma}$.

A flow chart illustrating the limit state method of design is shown in figure A.2.



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

NOTES

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

2 A general description of the limit state, method of design is given in ISO 23%: 1986, *General principles on reliability for structures.*

Figure A.2 — Typical flow chart of the limit state method

Annex B
(normative)

Values of coefficients γ_f, γ_m and γ_p

Table B.1 gives values of γ_f, γ_m and γ_p to be used in proof of competence calculations for load combinations A, B and C.

Table B.1 — Values of coefficients[1)] γ_f, γ_m and γ_p

Load combinations	Allowable stress method	Limit state method										
	Coefficient γ_f	Resistance coefficient γ_m	Partial load coefficient γ_p									
A	1,48	1,1	1,16	1,22	1,28	1,34[2)]	1,41	1,48	1,55	1,63	1,71	1,8
B	1,34	1,05	1,1	1,16	1,22	1,28[2)]	1,34	1,41	1,48	1,55	1,63	1,71
C	1,22	1	1,05	1,1	1,16	1,22[2)]	1,28	1,34	1,41	1,48	1,55	1,63

1) The coefficients are calculated from the formula $\gamma = 1{,}05^{\nu}$, where $0 < \nu < 12$.

2) These values apply to the mass of the payload.

For individual types of appliance, values of γ_p for each load should be selected from those given in table B.1 and stated in the part of this International Standard covering the type of crane under consideration. Where the same load occurs in more than one load combination, the values of γ_p applying to that load shall be taken from the same column.

The value of γ_p is selected according to the accuracy with which the relevant load can be determined. There may be special cases where the effect of a load is to reduce stress and values of $\gamma_p < 1$ may be appropriate. These will be covered in the other parts of this International Standard for individual types of appliances.

In proof of fatigue strength, instead of γ coefficients, strength values shall be used that provide an appropriate probability of survival. Fatigue design will be covered in a future International Standard.

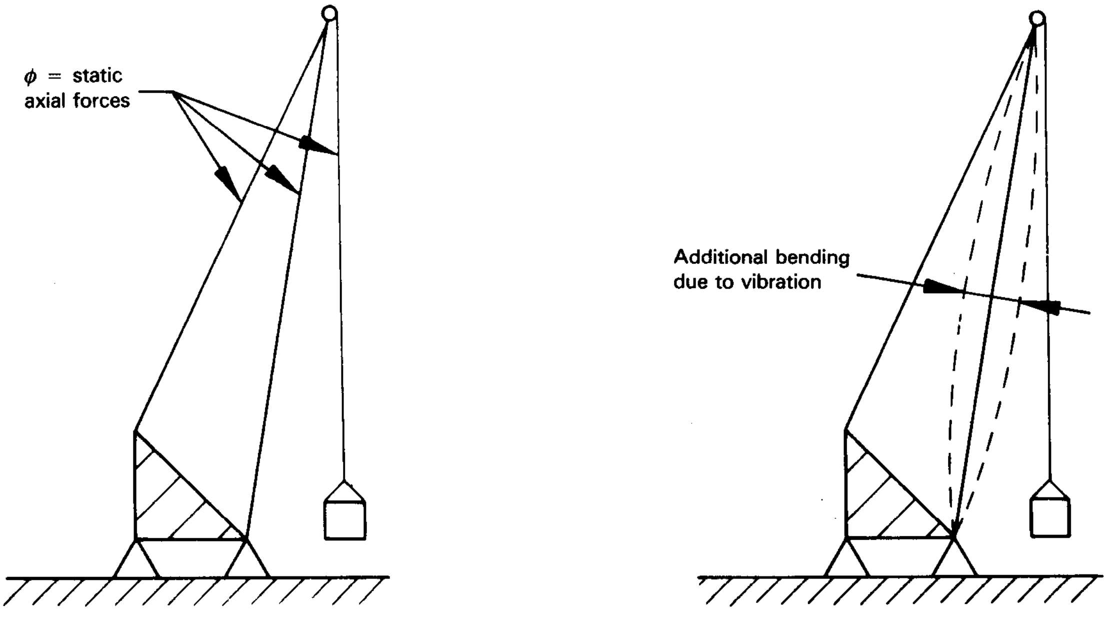
Annex C
(informative)

General comment on the application of ϕ factors

C.1 Dynamic effects

In general, the dynamic responses induced by different loads (see clause 6) are taken into account by the use of dynamic factors ϕ, by which gravitational forces due to the masses and inertia forces due to rigid body movements are multiplied (see figure C.1).

In cases where the load effect and dynamic response are not covered by these factors, elasto-kinetic analyses or experiments shall be carried out, unless it is known from experience that these effects are sufficiently small to be ignored.



a) Example of the load effects covered by dynamic factors ϕ

b) Example of the load effects not covered by dynamic factors ϕ

Figure C.1 — Application of dynamic factors ϕ

Annex D
(informative)

Example of a model for estimating the value of ϕ_4 for a lifting appliance travelling on rails

(see 6.1.3.2)

D.1 General

The dynamic loads caused by travelling or traversing on rails with steps or gaps may be estimated by using appropriate elasto-kinetic models. Unevenness functions may be used to represent the steps or gaps in the rails.

D.2 Elasto-kinetic model

In this example, the dynamic loads on the appliance caused by excitation of the system are estimated using a simple model.

A single mass m, in kilograms, moving horizontally at constant speed v, in metres per second, is supported by a linear elastic spring with a spring constant c, in newtons per metre, and is guided by a rail (see figure D.1).

With the unevenness function $h(t)$, in metres, and the coordinate $z(t)$, in metres, describing the position of the spring-supported mass, the dynamic force in the spring follows the expression $F(t) = c[h(t) - z(t)]$, in newtons.

The maximum force F_{max} is given by the maximum value of the expression $F(t)$ during the period of response. This may occur during or after the period of excitation.

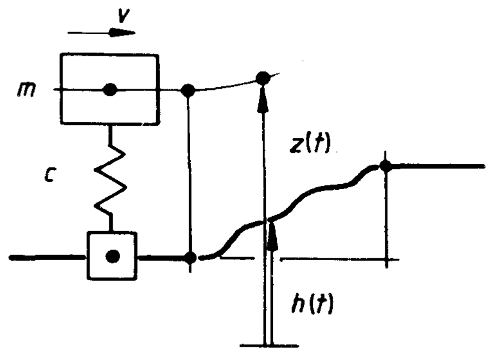
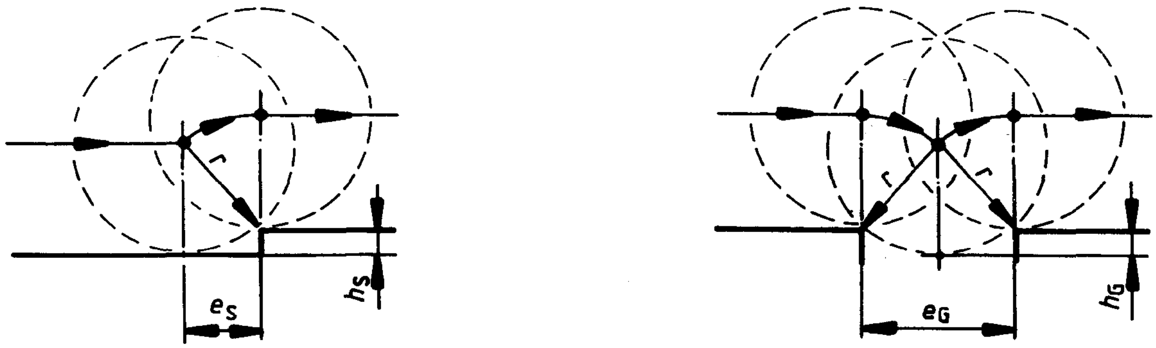


Figure D.1 — Model for determining dynamic factor ϕ_4

D.2.1 Movement of the wheel centre when passing over a step or gap

The movement of the wheel centre when passing over a step or gap and the corresponding formulae are shown in figure D.2.



$e_S \approx \sqrt{2rh_S} \quad (h_S \ll r)$

a) Passing over a step

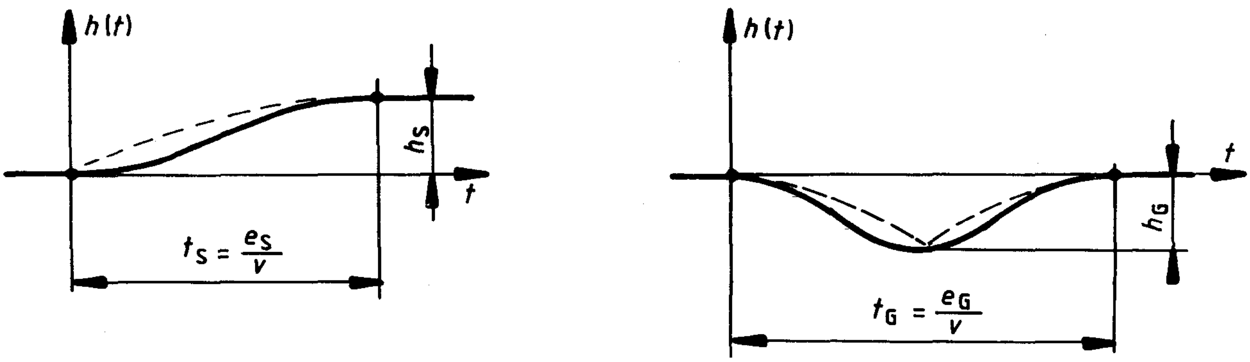
$h_G \approx \frac{e_G^2}{8r} \quad (e_G \approx r)$

b) Passing over a gap

Figure D.2 — Movement of wheel centre

D.2.2 Approximate unevenness functions for exciting the elasto-kinetic model

The approximate unevenness functions $h(t)$ for exciting the elasto-kinetic model are shown in figure D.3 and in the corresponding formulae in D.2.3.



$$h(t) = \frac{h_S}{2}(1 - \cos \Omega t)$$

where $\Omega t_S = \pi$

a) Passing over a step

$$h(t) = \frac{h_G}{2}(1 - \cos \Omega t)$$

where $\Omega t_G = 2\pi$

b) Passing over a gap

Figure D.3 — Unevenness functions $h(t)$

D.2.3 Maximum vertical accelerations

D.2.3.1 Lower end of spring

The maximum vertical acceleration of the lower end of the spring, $\hat{\ddot{h}}$, when passing over a step or a gap at constant speed v, is given by

$$\hat{\ddot{h}} = \frac{h_S}{2}\Omega^2 = \frac{h_G}{2}\Omega^2$$

$$= \left(\frac{\pi}{2}\right)^2 \frac{v^2}{r}$$

where h_S, h_G, Ω, v and r are as shown in figures D.2 and D.3.

D.2.3.2 Mass passing a step

The maximum vertical acceleration $\hat{\ddot{z}}$ for a mass m passing a step is given by

$$\hat{\ddot{z}} = \hat{\ddot{h}}\, \xi_S(\alpha_S)$$

where

$$\alpha_S = \frac{\omega h}{\pi v} \sqrt{\frac{2r}{h_S}}$$

in which $\omega = \sqrt{c/m}$ is the natural circular frequency of the elasto-kinetic model.

D.2.3.3 Mass passing a gap

The maximum vertical acceleration $\hat{\ddot{z}}$ for a mass m passing a gap is given by

$$\hat{\ddot{z}} = \hat{\ddot{h}}\, \xi_G(\alpha_G)$$

where

$$\alpha_G = \frac{\omega e_G}{2\pi v}$$

D.2.4 Factors ξ_S and ξ_G

In figure D.4, the curves for factors $\xi_S(\alpha_S)$ and $\xi_G(\alpha_G)$ for a parabolic (par) unevenness function are compared with those for the approximate cosine (cos) unevenness function previously introduced. The numbers in brackets [(1) or (2)] indicate the periods for which the factors ξ are valid. Period (1) covers times t_S and t_G and period (2) is the response time thereafter.

For both excitations (step or gap) the maximum values of ξ_S or ξ_G for $\alpha \approx < 1{,}3$ have been found to occur in period (2), i.e. after the time the wheel has passed the unevenness, and with the cosine unevenness function [cos(2)].

In this case, the values of the factors may be determined analytically by

$$\xi_S = \frac{\alpha_S^2}{1-\alpha_S^2}\sqrt{2+2\cos(\pi\alpha_S)}$$

or

$$\xi_G = \frac{\alpha_G^2}{1-\alpha_G^2}\sqrt{2-2\cos(2\pi\alpha_G)}$$

D.2.5 Dynamic factor, ϕ_4

ϕ_4 is defined as follows:

$$\phi_4 = \frac{mg + m\hat{\ddot{z}}}{mg}$$

$$= 1 + \frac{\hat{\ddot{h}}}{g}\xi$$

For the two cases and the assumptions made, including $\alpha < 1{,}3$, using the formulae for ξ_S, α_S and ξ_G, α_G, the factors ϕ_4 may be calculated as follows:

for a step:

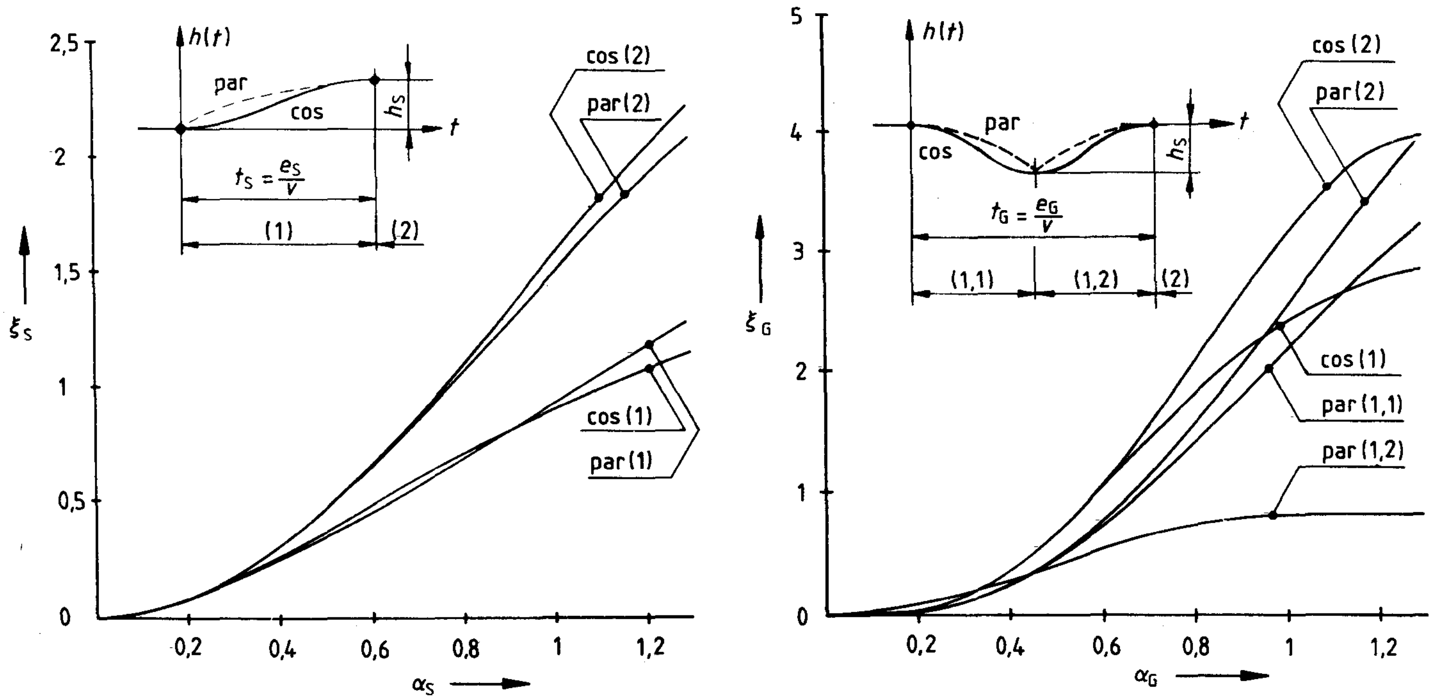
$$\phi_4 = 1 + \left(\frac{\pi}{2}\right)^2 \frac{v^2}{gr}\xi_S(\alpha_S)$$

for a gap:

$$\phi_4 = 1 + \left(\frac{\pi}{2}\right)^2 \frac{v^2}{gr}\xi_G(\alpha_G)$$

D.2.6 Comments

The use of this simple elasto-kinetic model is restricted to appliances whose actual dynamic behaviour corresponds to that of the model and which are excited in the manner shown by passing over steps or gaps in the rails. If more than one natural mode contributes a significant response and/or rotation occurs, the designer should estimate the dynamic loads using an appropriate model for the circumstances.



a) For a step

b) For a gap

Figure D.4 — Curves of unevenness function

Annex E
(informative)

Example of determination of loads caused by acceleration

(see 6.1.4)

E.1 Rigid body kinetic model

The example considered is that of a rigid lifting appliance (i.e. an overhead travelling crane) consisting of a double girder crane bridge supported by four crane travel wheels and travelling at a constant speed. One wheel on each side is driven by a simplified independent drive. A traversing loaded trolley is supported by the crane bridge (see figure E.1).

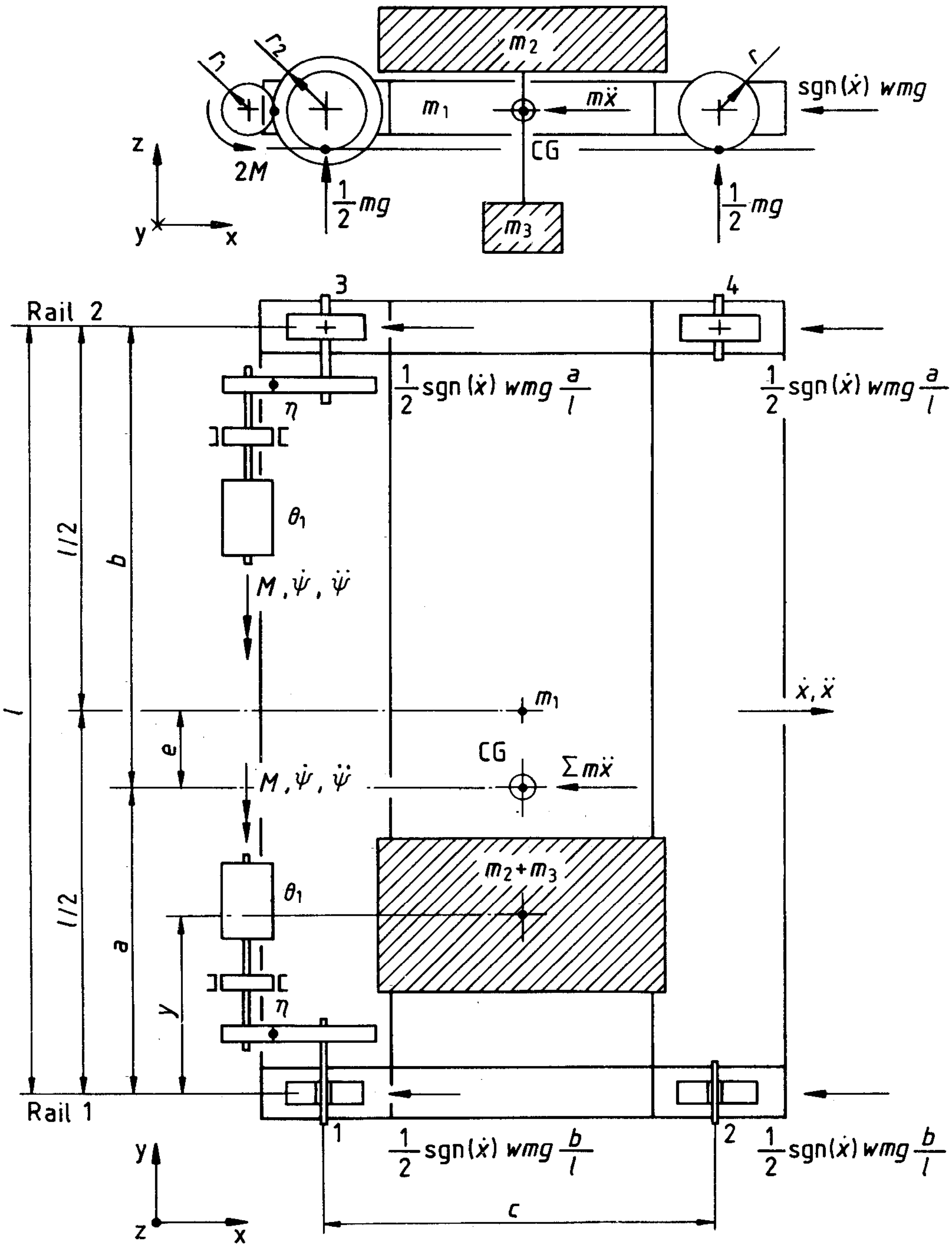


Figure E.1 — Loads acting on an overhead travelling crane (see table E.1)

The drive forces developed by the motors and brakes are transferred through one-step gears to the crane travel wheels. The travel wheels are supported in the end carriages, those on one side being laterally fixed and those on the other side being laterally movable.

E.2 Symbols

The symbols used in this annex are given in table E.1.

Table E.1 — Symbols used in annex E

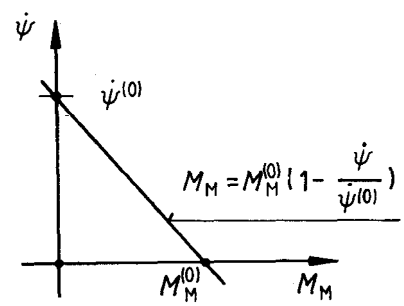
Symbol	Description
	Geometric parameters (in metres)
l	Span of the appliance
y	Distance of centre of mass of loaded trolley from rail 1
a	Distance of centre of gravity (CG) from rail 1
b	Distance of centre of gravity (CG) from rail 2
c	Wheel base
r_1	Radius of gear wheel 1
r_2	Radius of gear wheel 2
r	Radius of crane travel wheels
	Masses (in kilograms)
m_1	Mass of crane bridge with travel drives
m_2	Mass of crab
m_3	Gross load
m	Mass of the loaded lifting appliance ($m = m_1 + m_2 + m_3$)
	Mass moments of inertia (in kilograms metre squared)
θ_1	Mass moment of inertia of motor, coupling, brake drum and gear wheel 1
θ_2	Mass moment of inertia of gear wheel 2 and crane travel wheels (neglected in this example)
	Internal friction losses
η	Ratio of output power of gearing to input power of gearing
	Speeds and accelerations (in radians or metres per second or second squared)
$\dot{\psi}$, $\ddot{\psi}$	Rotational speed and acceleration, respectively, of motor, coupling, brake drum and gear wheel 1
$\dot{x}$, $\ddot{x}$	Travel speed and acceleration, respectively, of the lifting appliance
	Torques (in newton metres)
M	Drive torque acting on the first shaft of the crane travel gear
M_M	Torque due to the stationary characteristics of the motor
M_B	Torque of the mechanical brake

E.3 Forces

E.3.1 Drive forces and external forces

The motion of the appliance $[x(t)]$ and load effects depend on drive forces which are in balance with the internal frictional forces, the inertia forces and the external forces. The external forces include the frictional forces due to mechanical resistance (losses) at the wheels, wind load and, in the case of an inclined track, gravitational forces.

The torques $M = M_M$ or $M = M_B$ may be defined by the motor or brake characteristics and these are illustrated by the two examples given in figures E.2 and E.3.

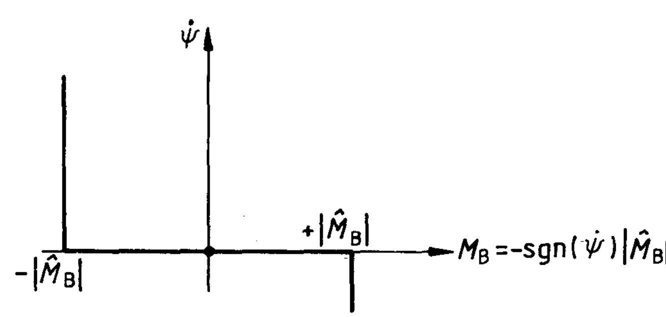


M_M is the steady-state output torque of the motor at a motor speed of $\dot{\psi}$

$M_M^{(0)}$ is the motor starting torque ($\dot{\psi} = 0$)

$\dot{\psi}^{(0)}$ is the synchronous rotational speed of the motor ($M_M = 0$)

Figure E.2 — Resistor-controlled slip-ring motor — Simplified presentation of motor characteristics

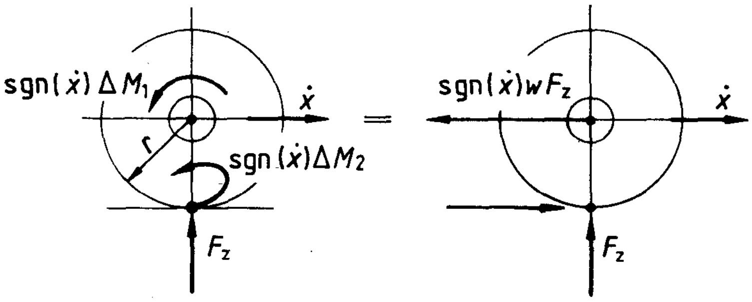


M_B is the brake torque whose direction is opposite to that of $\dot{\psi}$. For simplification purposes its magnitude, $|\hat{M}_B|$, is taken as constant. Mathematically it is expressed as $M_B = -\mathrm{sgn}(\dot{\psi})|\hat{M}_B|$.

Figure E.3 — Mechanical brake — Formal presentation of brake torque

E.3.2 Frictional losses at a wheel

Figure E.4 illustrates frictional losses at a wheel.



ΔM_1 is the torque loss due to friction in the wheel bearing

ΔM_2 is the torque representing the losses due to rolling friction in the contact zone of the rolling wheel

F_z is the wheel load

w is the equivalent friction coefficient ($\Delta M_1 + \Delta M_2 = wF_z r$)

Figure E.4 — Frictional losses at a wheel

E.4 Drive accelerations

The drive model shown in figure E.5 is used for estimating drive accelerations. This representation combines the two drives acting to balance forces and includes all significant effects.

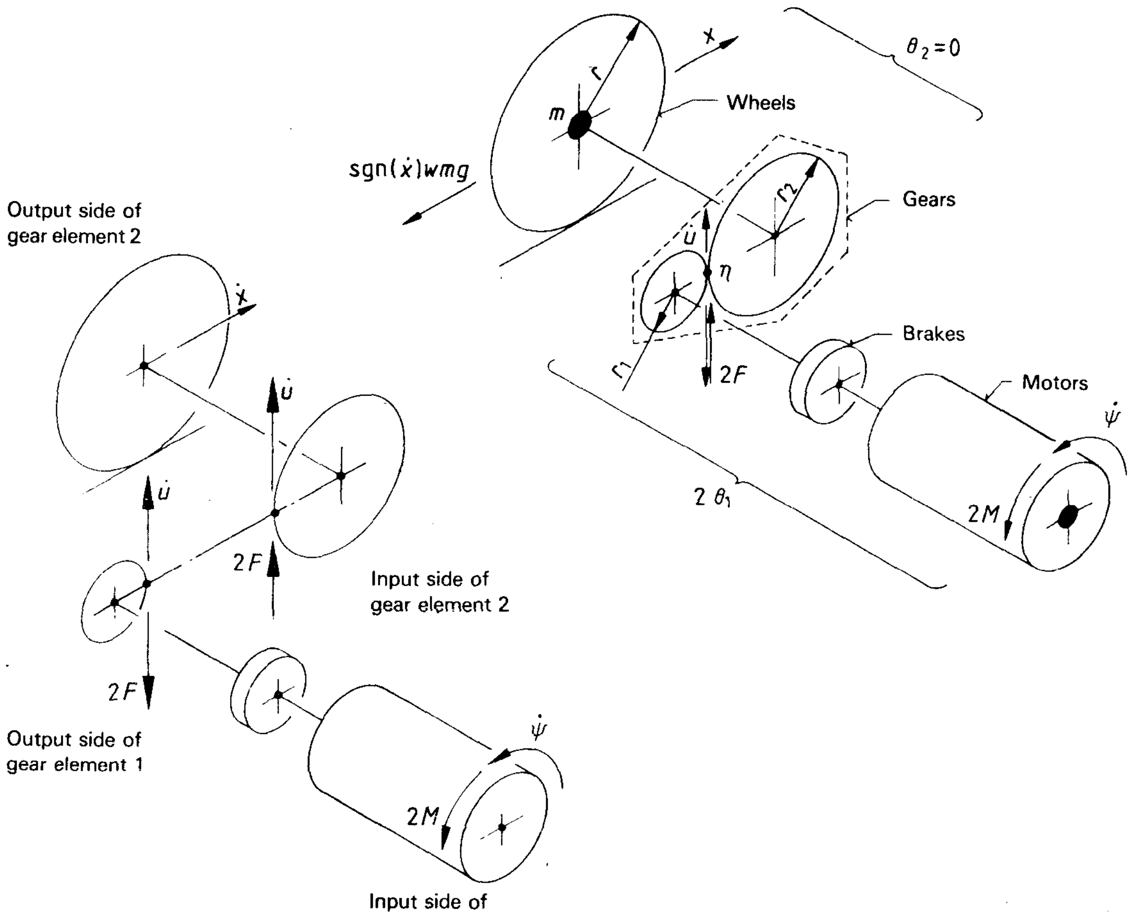


Figure E.5 — Crane drive model (sign convention)

From rigid body kinetic analysis, neglecting the mass moments of inertia, θ_2, the acceleration, $\ddot{x}$, of an appliance not affected by wind forces can be expressed as

$$\ddot{x} = \frac{2M\, r_1^{-1}\, r_2\, r^{-1}\, \eta^{\lambda} - \operatorname{sgn}(\dot{x})\, wmg}{2\theta_1 (r_1^{-1}\, r_2\, r^{-1})^2\, \eta^{\lambda} + m}$$

where

$\lambda = \operatorname{sgn}(\dot{u}F)$;

$\dot{u}$ is the tangential speed of gear wheels;

F is the tangential force to be transferred by the gear wheels.

NOTE — **Sign convention of speed and internal forces**

Internal forces of a gear element are assumed to be positive when acting at the *input* side *in* the direction of the positive speed and at the *output* side *opposite* to the direction of the positive speed. The speeds of gear elements are chosen positive if acting in the positive direction of the movements of the lifting appliance considering the kinematic interactions of the mechanical parts.

E.5 Loads and load effects

The loads and load effects caused by appliance drives during regular use can be taken into account considering the relevant events, for example:

a) Event I

Accelerating the appliance from rest ($\dot{\psi} = 0$) by applying a starting torque $M_{\mathrm{M}}^{(0)}$ to each travel drive.

b) Event II

Decelerating the crane from steady-state motion ($\ddot{\psi} = \ddot{x} = 0$) by mechanical braking whereby the torque on each travel drive is changed from a motor torque M_{M} ($\ddot{\psi} = 0$) to a braking torque of $-|\hat{M}_{\mathrm{B}}|$.

For the purposes of the example in this annex, events I and II are taken to be instantaneous changes in torque. The events are illustrated in figure E.6.

E.6 Accelerations

Before the design load effects arising from changes in torque can be calculated, such as those of events I and II of clause E.5, it is necessary to estimate the initial acceleration $\ddot{x}_{(i)}$ and the final acceleration $\ddot{x}_{(f)}$ bounding the event. These can be estimated as follows:

a) For event I

$$\ddot{x}_{(i)} = 0$$

$$\ddot{x}_{(f)} = \frac{2M_{\mathrm{M}}^{(0)}\, r_1^{-1}\, r_2\, r^{-1}\, \eta - wmg}{2\theta_1 (r_1^{-1}\, r_2\, r^{-1})^2\, \eta + m}$$

since $\lambda = +1$ (as $\dot{u} > 0$ and $F > 0$)

b) For event II

$$\ddot{x}_{(i)} = 0 = \frac{2M_{\mathrm{M}}(\ddot{\psi} = 0)\, r_1^{-1}\, r_2\, r^{-1}\, \eta - wmg}{2\theta_1 (r_1^{-1}\, r_2\, r^{-1})^2\, \eta + m}$$

since $\lambda = +1$ (as $\dot{u} > 0$ and $F > 0$)

$$\ddot{x}_{(f)} = -\frac{2|\hat{M}_{\mathrm{B}}|\, r_1^{-1}\, r_2\, r^{-1}\, \eta^{-1} + wmg}{2\theta_1 (r_1^{-1}\, r_2\, r^{-1})^2\, \eta^{-1} + m}$$

since $\lambda = -1$ (as $\dot{u} > 0$ and $F < 0$)

From these results it can be seen that if $M_{\mathrm{M}}^{(0)} = |\hat{M}_{\mathrm{B}}|$, the acceleration $\ddot{x}_{(f)}$ for event I is less than the deceleration $\ddot{x}_{(f)}$ for event II.

E.7 Design load effects in the mechanical components

As an example, the tangential force to be transferred by the gears and to be considered in design, F, is estimated as follows (see clause E.4 and figure E.5):

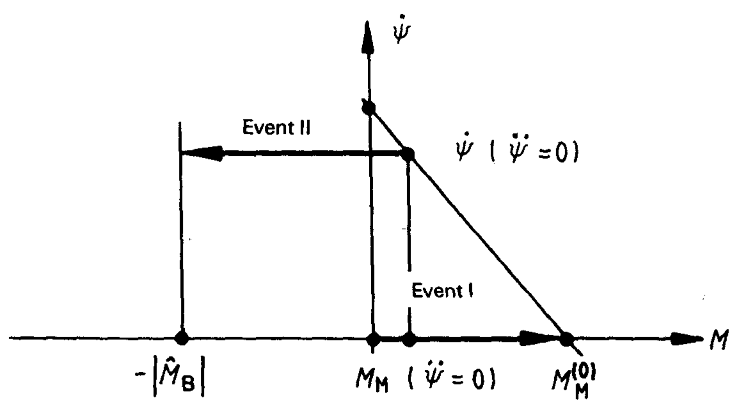


Figure E.6 — Illustration of events I and II

$$F = (M - \theta_1 \ddot{\psi})r_1^{-1}$$

where $\ddot{\psi} = r_1^{-1} r_2 r^{-1} \ddot{x}$

$$\hat{F} = F_{(i)} + \phi_5 \Delta F$$

where $\Delta F = F_{(f)} - F_{(i)}$

a) For event I

$$F_{(i)} = 0$$

$$F_{(f)} = [M_M^{(0)} - \theta_1 r_1^{-1} r_2 r^{-1} \ddot{x}_{(f)}]r_1^{-1}$$

b) For event II

$$F_{(i)} = M_M(\ddot{\psi} = 0)r_1^{-1}$$

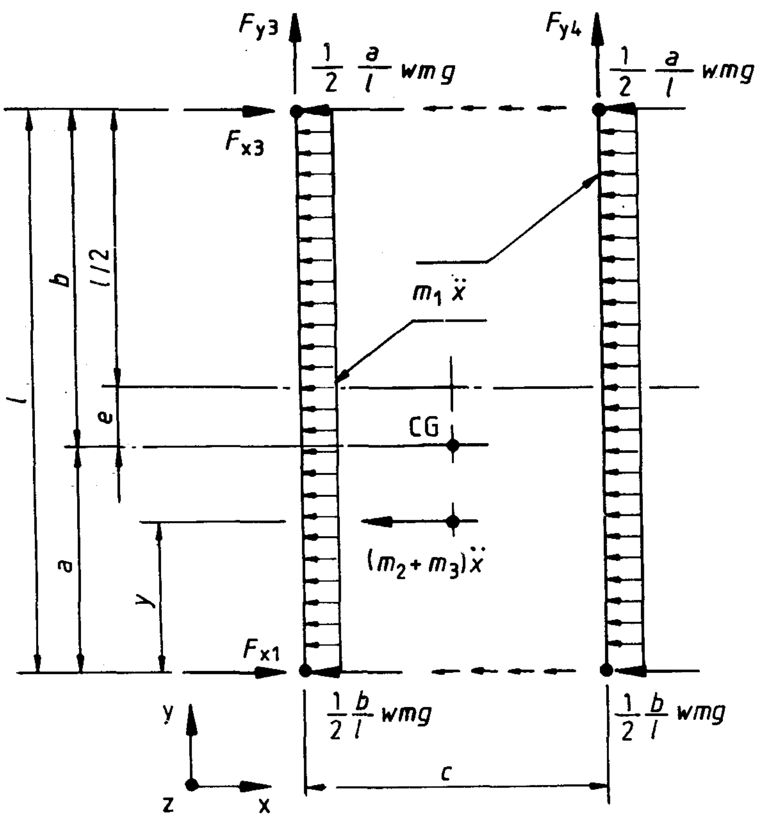
$$F_{(f)} = [-|\hat{M}_B| - \theta_1 r_1^{-1} r_2 r^{-1} \ddot{x}_{(f)}]r_1^{-1}$$

E.8 Design load effects in the structural components

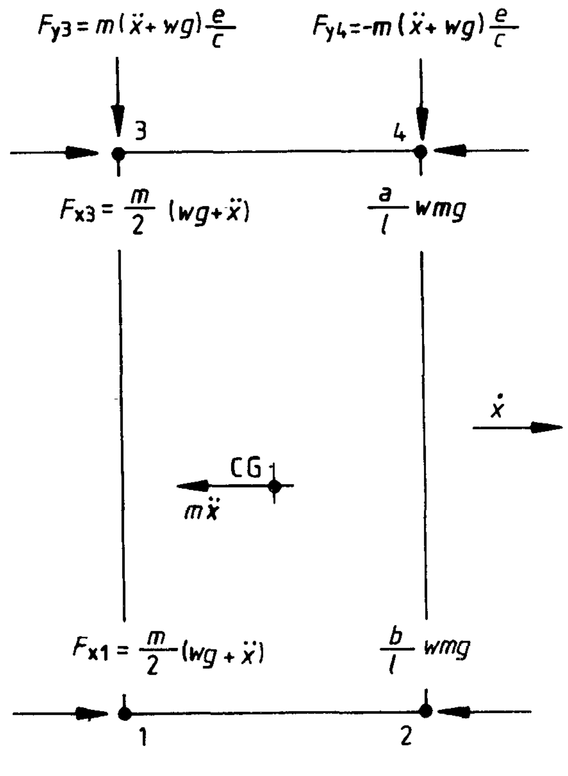
As an example, the horizontal loads and reactions to be transferred by the crane girder, and to be considered in design, are estimated as follows. (See figure E.7.)

The horizontal loads and forces are caused by drive forces accelerating the crane and do not include skewing forces.

During acceleration, the two acting drives balance the mass forces ($m\ddot{x} = m_1\ddot{x} + m_2\ddot{x} + m_3\ddot{x}$) and the forces due to the frictional losses at all wheels (wmg). It is assumed that the crane drive characteristics are identical as well as their control; therefore the drive forces are equally distributed to both of the drives ($F_{x3} = F_{x1} = \frac{1}{2} m\ddot{x} + \frac{1}{2} wmg$). The resultant drive force acts in the centreline of the span. Forces transverse to the



a) Distribution of horizontal loads during acceleration of the appliance



b) Resultant loads — Acting drive forces $F_{x3} = F_{x1}$ and reacting forces $F_{y4} = -F_{y3}$

Figure E.7 — Horizontal loads and reactions

runway, $F_{y4} = -F_{y3}$, usually occur due to the distance $e = \frac{l}{2} - a$ between the acting and reacting forces, and become

$$F_{y4} = F_{y3}$$

$$= m\,(\ddot{x} + wg)\,\frac{e}{c}$$

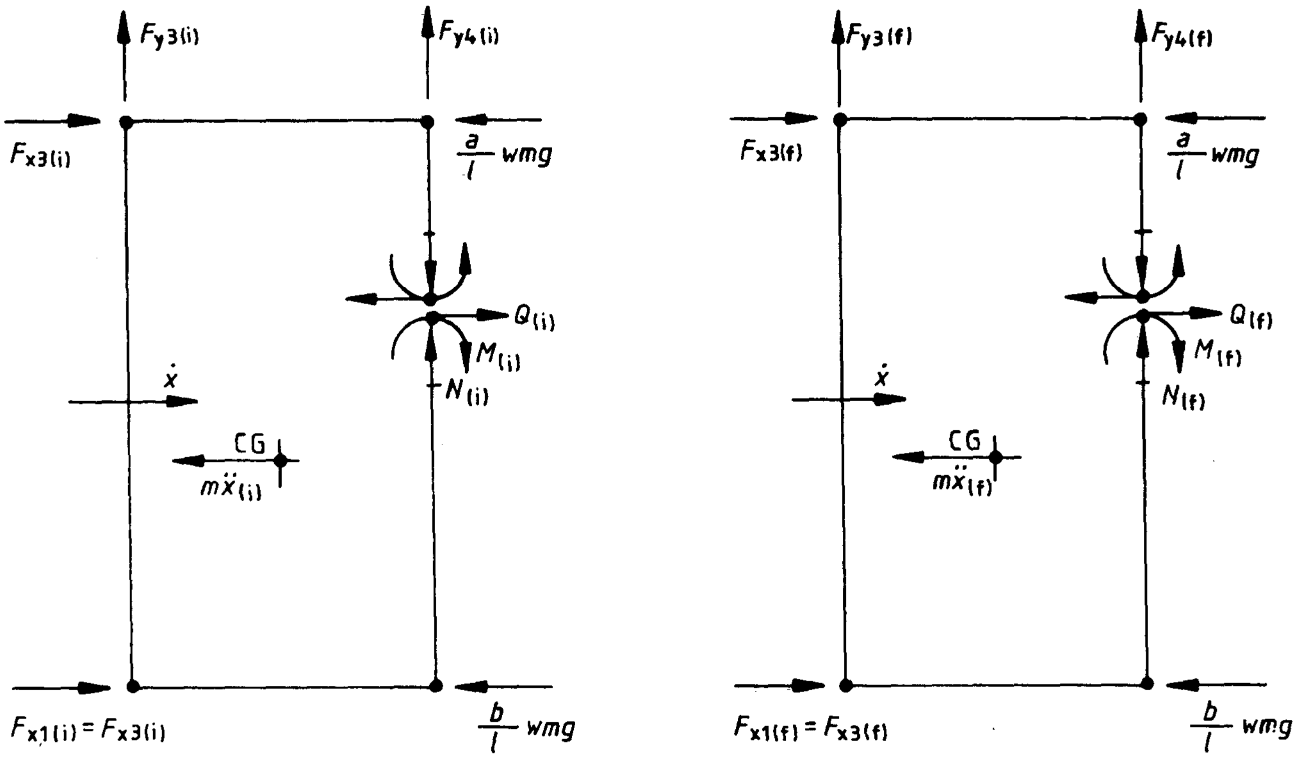
The design load effects $\hat{F}$ and the accelerations before $[\ddot{x}_{(i)}]$ and after $[\ddot{x}_{(f)}]$ changing the torques for any considered event should be evaluated.

From the acting loads, the mass forces $m\ddot{x}_{(i)}$ and $m\ddot{x}_{(f)}$ as well as the resultant friction forces, all relevant load effects $F_{(i)}$ $[N_{(i)}, Q_{(i)}, M_{(i)}]$ and $F_{(f)}$ $[N_{(f)}, Q_{(f)}, M_{(f)}]$ respectively should be estimated by an elastostatic calculation considering the crane girder as a plane (or space) frame (see figure E.8).

The design load effects may be evaluated, having reference to clauses E.4 and E.7, from

$$\hat{F} = F_{(i)} + \phi_5\,\Delta F$$

where $\Delta F = F_{(f)} - F_{(i)}$



In the special cases of events I and II:

$$\ddot{x}_{(i)} = 0$$

a) Before changing the torques (i)

b) After changing the torques (f)

Figure E.8 — Loading state

Annex F
(informative)

Example of a method for analysing loads due to skewing

(see 6.2.2)

F.1 Model of appliance

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

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

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

The different combinations of transversally in-line wheel pairs that are possible are shown in figure F.1.

In figure F.2, the positions of the wheel pairs relative to the position of the guide means in front of the travelling crane are defined by the distances d_i.

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

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

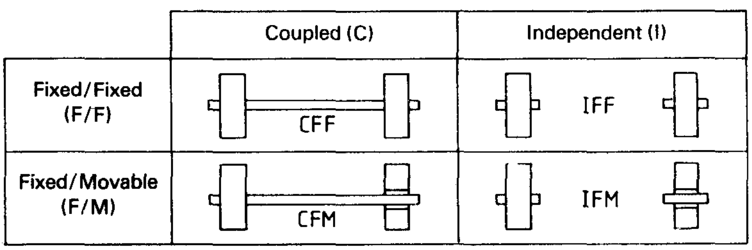


Figure F.1 — Different combinations of wheel pairs

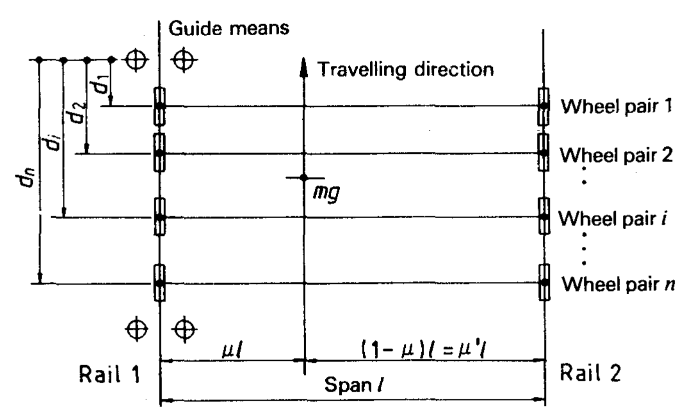


Figure F.2 — Positions of wheel pairs

F.2 Relationship between tangential forces and displacements

It is first necessary to assume a relationship between the tangential forces and the corresponding displacements occurring between wheel and rail. Since the wheel has to transfer drive moments (M_y) to the rail and its movement is restricted by the system (crane and runway) it slides in longitudinal and lateral directions [$u(u_x, u_y)$]; corresponding tangential forces (F_x, F_y) react on the crane (see figure F.3).

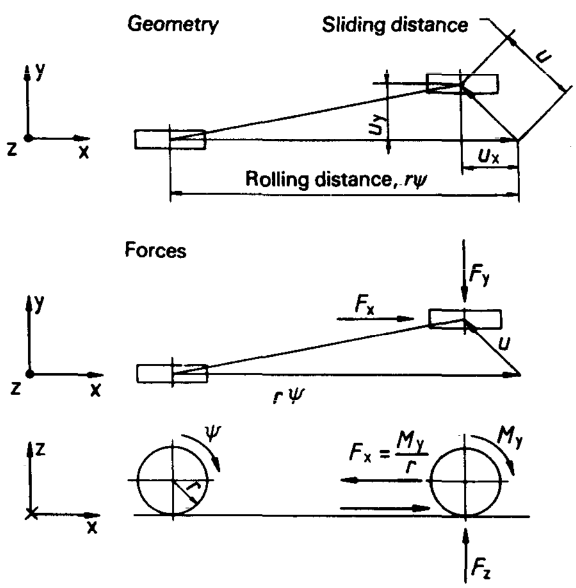


Figure F.3 — Tangential forces and displacements

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

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

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

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

$$f_x = 0{,}3\ [1 - \exp(-250\ s_x)],\ \text{for}\ s_x < 0{,}015$$

$$f_y = 0{,}3\ [1 - \exp(-250\ s_y)],\ \text{for}\ s_y < 0{,}015$$

F.3 Loads due to skewing

The crane model is assumed to be travelling in steady motion and to have skewed to an angle α, as shown in figure F.4. The appliance may be guided horizontally by external means or by wheel flanges.

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

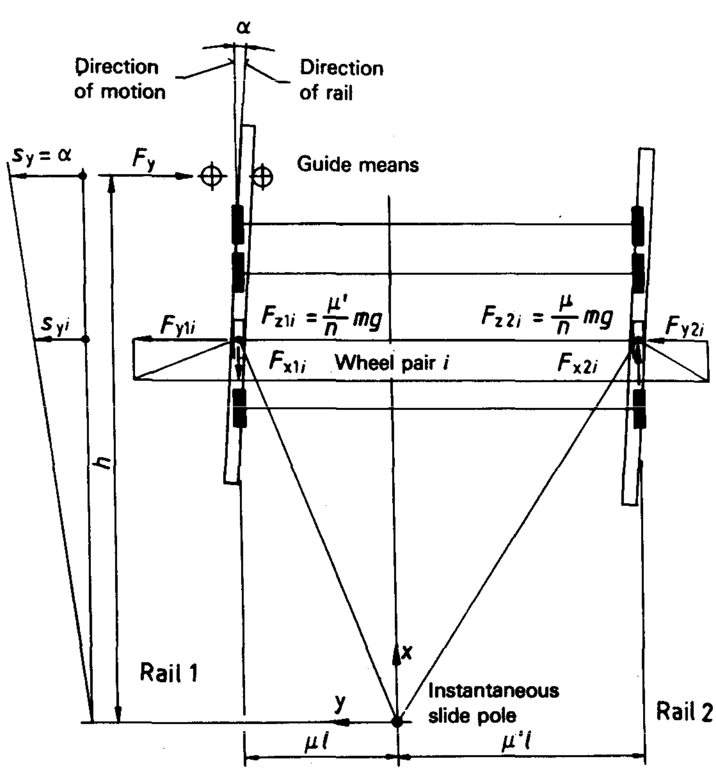


Figure F.4 — Loads acting on crane in skewed position

slide pole, the corresponding skewing forces can be calculated as follows:

a) Distance between instantaneous slide pole and guide means, h

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

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

where

p is the number of pairs of coupled wheels;

μ is the distance of the instantaneous slide pole from rail 1;

μ' is the distance of the instantaneous slide pole from rail 2;

l is the span of the appliance;

d_i is the distance of wheel pair i from the guide means.

b) Guide force, F_y

$F_y = \nu f m g$

where

$\nu = 1 - \Sigma d_i / nh$, for systems F/F,

$= \mu'(1 - \Sigma d_i / nh)$, for systems F/M;

$f = 0{,}3[1 - \exp(-250\,\alpha)]$, where $\alpha < 0{,}015$ rad;

mg is the gravitational force due to the mass of the loaded appliance.

NOTE — The skewing angle α, which should be equal to or less than 0,015, should be chosen taking into account the space between the guide means and the rail as well as reasonable dimensional variation and wear of the appliance wheels and the rails.

F.4 Tangential forces, F_x and F_y

$F_{x1i} = \xi_{1i} f m g$

$F_{x2i} = \xi_{2i} f m g$

$F_{y1i} = \nu_{1i} f m g$

$F_{y2i} = \nu_{2i} f m g$

where

f and mg are as given in F.3 b);

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

Table F.1 — Values of ξ_{1i}, ξ_{2i}, ν_{1i} and ν_{2i}

Combinations	$\xi_{1i} = \xi_{2i}$	ν_{1i}	ν_{2i}
CFF	$\mu\mu' l / nh$	$\frac{\mu'}{n}\left(1 - \frac{d_i}{h}\right)$	$\frac{\mu}{n}\left(1 - \frac{d_i}{h}\right)$
IFF	0		
CFM	$\mu\mu' l / nh$		0
IFM	0		

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