EN 81 : PART 1

HANDBOOK

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① Pages 1 to 54

HANDBOOK and COMMENTS on the EN 81/Part I SAFETY CODE

by Andre Leenders

with some references to other leading Codes

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NOTE

The 16 APPENDICES to this HANDBOOK are in a separate volume.

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FOREWORD

FWD/01: Reasons for a Safety Code

In any community where lifts are used, the need for a safety code will arise sooner or later because accidents will occur and:

- ruling authorities will want to require a minimum set of safety devices to insure a reasonable safety level for the public,
- responsible lift suppliers will welcome those requirements which help to avoid the wild competition of irresponsible lift contractors,
- manufacturers and contractors in general will need a way to prove that they have made sufficient provision for safety, considering the current state of the art, in case of accidents.

Originally, the safety codes were largely empirical.
Rules have been written and gradually added to when the recurrence of accidents prompted the code makers to issue rules to avoid such accidents. These new rules were based on existing designs and current practice, including new technological developments after the facts.

At a later stage, a more scientifical approach has sometimes been attempted for some subjects but, quite naturally,:

- manufacturers are often not favourable to rules which would impose changes in existing designs and habits,
- Governement representatives and Inspectors are, in general, reluctant to change time-honoured national practice because of the responsibility involved or the difficulty of changing existing laws.

Moreover, the necessity to gain the acceptance of the code by various groups of people who sometimes have a limited knowledge of the lift techniques, has oriented the wording of certain rules.

It should be noted also that things which are recognized as common practice are not always spelled out again in the code specific for lifts, but this "common practice" is not always the same in all communities.

FWD/02: Reasons for a European Safety Code

The E.E.C., back in the 60's, had listed, amomgst the existing "technical barriers to trade", the various national Lift Safety Codes because of the differences between their requirements. It was planned to issue a Directive for removing this barrier to trade within the Common Market and the C.E.N. received a mandate to prepare a European lift safety code which could be the basis for this specific Directive.

Beside the political decision to remove technical barriers to trade within the E.E.C., one must admit that there are no rational technical reasons for not having the same safety requirements everywhere.

FWD/03: History of the EN 81-Part I

The CEN/TC 10 was established in 1971 and appointed a Work Group N° 1 for preparing a draft code for lifts, starting with the electric ropes lifts.

Immediately, the Technical Committee of FEM-Section VII prepared a working paper based on the CIRA code and this working paper was accepted as a basis for discussions by the WG N°1.

From 1971 to the final voting of the EN 81-I in 1977, the WG N°1 had a great number of meetings and the drafts were revised several times. The cost of such a document is tremendous but the result of the voting was rewarding: the EN 81-Part 1 had a very large acceptance.

In 1982, the CEN/TC 10 decided to initiate the first revision of EN 81-I. Proposals were collected from all CEN members and the WG N°1 started to review the proposals. However, in 1984, the TC 10 decided to limit the revision to a selected number of amendments for not conflicting with the EEC Directive which had finally been issued after many years of delay.

A more important revision will be reconsidered in a few years.

FWD/04: Value of EN 81-Part 1 by comparison with national codes

Each of the European countries which participated to the preparation of the new code had had a satisfactory experience with its own national code. Some codes were more complete than others but I think it can be said that the level of safety achieved by any of them was satisfactory.

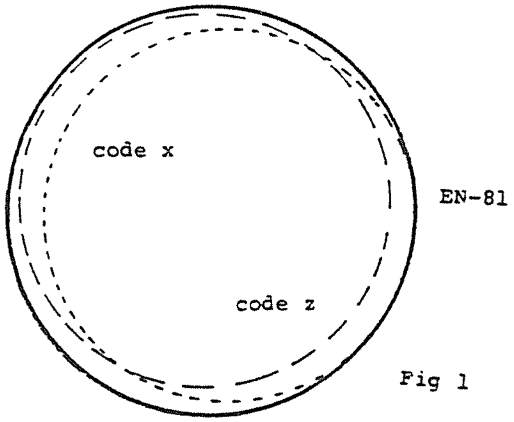


Fig 1

The WG N°1 tried to incorporate the best of each code in the EN 81 and, if for the sake of illustrating what I mean, we say that the levels of safety achieved by the national codes were ranging from 90 to 95%, the EN 81 achieves probably something like 97 or 98%.
The figure N° 1 illustrates this.

Moreover, I think it compares favourably with any other code in the world.

FWD/05: Justification of the "National options"

A lift does not emerge from the factory in a completed form.
The components have to be installed in the spaces as provided in the building for the machine room, the well and the pit.
To be complete, the requirements have to cover not only the mechanical and electrical equipment supplied by the lift manufacturer but also some part of the civil work and other peripheral equipment.

Those other things are sometimes, but not always, subject to other local regulations; so it was proper to give, in the EN 81:

- indications for the case that there was no local rule,
- the degree of freedom as regards the above indications for the case where there was a local rule.

One must keep in mind that, with one exception only, the possible variations do not concern the lift components themselves and are not creating barriers to trade. This exception is the landing doors fire test on which I will comment under the references 7.2 and F.2.

Recently, the CEN Central Secretariat expressed reservations about the National options. I think it would be a big mistake to suppress them because the only way to do that would be:

- either not to say anything on the subject which would lead to a lesser degree of safety,
- or be specific and run the risk of making it impossible for some countries to approve the code because of a few requirements not related to the lift material itself.

FWD/06: Justification of the Interpretation Committee

A text can be rephrased and reviewed ten times, there will always be people who will twist the words to suit their wishes or new designs for which the wording is not clearly applicable. The national code committees are in many cases able to answer the questions, but in case of doubt they can ask the CEN/TC 10/WG N°1 what the experts had in mind when the rule was written.

The WG N° 1, when acting as "Interpretation Committee", is following the rules laid down in Document 59 (See Appendix N°1)
Such interpretation committees have been operating satisfactorily in various countries for many years. The interpretations are the equivalent of the jurisprudence which is a necessary complement to any law.

For anyone who has to work with the code, I do recommend that the reference number of all published interpretations be written, in the code book, in the margin of the clause to which it applies.

FWD/07: Difficulties in preparing a European code

It took 7 years, but the Part 1 of EN 81 was finally put to the vote and passed with a large majority.
It seems that it will be much more difficult in the future because of the presence of people who find difficulty in accepting new rules.

Then,although they will never openly admit it, there are people who do not welcome a European code.

First, you have some manufacturers working only on a national level under the protection of trade barriers.

Second, you have national government officials and laboratories who fear to lose some of their prerogatives or to become redundant when codes are prepared and components tested on a european level.

When the economic situation is gloomy, those opponents become more active and put forward all kinds of alleged technical reasons for not accepting a common European solution.

Unfortunately, the internal rules of the CEN Organization make it possible for them to torpedo a project or at least to slow it down considerably, depriving their fellow citizens of the advantages:

- either of lower costs due to a wider and more competitive market
- or of a better degree of safety due to sounder technical requirements.

The suggestions presented in FWD/08 below are, I am afraid, wishful thinking. Let us hope however, that they will generate some fruitful reflexions on the subject.

FWD/08: Suggestions for a modification of the CEN procedures

The making of a European safety code is the result of a political decision based on the analysis of the economic benefits of removing trade barriers.

It should not be allowed to delay the acceptance of the code on alleged technical grounds. I use the term "alleged technical grounds" because, when it comes to the point of selecting between two different proposals which have already been discussed at high levels, one could flip the coin and be sure that no big mistake can result.
There can only be a marginal difference between the values of the two, none of which can possibly be perfect in any event. Moreover, the difference will depend on the circumstances.It is much more important to have consistant requirements throughout the code.

Consequently, it is my opinion that the following rules should apply:

a) When a country has decided in favour of preparing a European code, it should be a commitment to accept automatically the outcome of the work of the ad-hoc Work Group.

   This would be automatically the case for all the countries which are part of the Common Market when the CEN has a mandate of the E.E.C..

b) Only the countries having decided in favour of the code as per (a) should be allowed to send an expert for participating in the Work Group.

c) There should be only one appointed expert for each country. He might be, in case of need, assisted by a specialist in a specific field but the appointed expert only could voice an opinion.

d) The members of the Work Group should be working as independant experts when preparing the code. They would be expected to put at the disposal of the group the experience and expertise developed in their respective countries.There should be no need to refer back to their National Committees on purely technical discussions; their expertise should be sufficient for them to reach satisfactory compromises.

   Incidentally, experts should be changed as rarely as possible because newcomers will question many past settlements or compromises and slow down the progress of the work.

e) When no clear consensus comes out of the discussions, the Chairman should take the responsibility of selecting the alternative which is best in line with the rest of the code.

f) The Work Group might decide to circulate a draft amongst the National Committees for comments to make sure that none of the aspects of the question has been neglected.

   The Work Group might also decide to refer questions to the Technical Committee for examination at a later date.

   When the Chairman of the Work Group feels that the code is ready,it should be sent to the Central Secretariat for editorial review and publication as an European Norm without having to submit it to a vote as is the case today.

With such rules, the role of the Technnical Committe would be:

- to create the Work Group,
- to appoint the Chairman of the Work Group,
- to define the frame of the work,
- to fix the deadline,
- to answer the questions raised by the Work Group.

With rules along the above lines, it would not be possible for a country to participate in the Work Group and to fight at length for the acceptance of its own peculiar technical solutions, knowing all the while that they will cast a "NO" vote for other reasons.

FWD/09: Implementation of the code on national levels

When the code has been voted, all countries having voted positively must accept it, unchanged, as national code.

However, the meaning of this varies from country to country.
For example, France is enforcing the application of the code by decree.
In Great Britain, nothing is enforced by law but the insurance companies do not insure lifts which are not built according to the code.
In some other countries, governement officials insist on making a difference between code and regulations.

At best, these regulations are reproducing part of the code specifications (this is, by force, the case if the country is a member of the E.E.C.).
At worst, the regulations are diverging from the code requirements.
In my opinion, this is wrong in both cases because:

- the regulations can only be redundant and preparing them is a waste of time and thus of money,
- having two parallel sets of rules is confusing,
- trying to say the same thing with different words leads to different interpretations thus to conflicts.

I strongly believe that the French approach is the best and the only one which would really remove all barriers to trade if applied by all countries. Even on a national level this leads to fair competition because, forced by law or not, responsible lift contractors will apply the code whereas others will limit themselves to the minimum required to have the lift accepted. I also believe that, even if the code is not enforced by law, it would be to the advantage of insurance companies to insure only lifts built according to the code because this is the only way to know the risk involved and to be sure that this is a low risk.

FWD/10: Purpose of this HANDBOOK

The aims in writing this HANDBOOK were:
- to explain the reasons for certain rules,
- to make the application of the rules easier, sometimes by giving examples or proposing formulae,
- to analyze how the degree of safety achieved by certain rules varies throughout the range of application,
- to compare some of the EN specifications with the specifications of the USA (ANSI) and Canadian (CSA) codes.
- to call the attention to unusual applications of the code specifications.

The ideas developped in this HANDBOOK are personnal but I hope that, despite probable shortcomings and errors, it will help the industry and the code makers in Europe and elsewhhere.

I do not expect that all my ideas and considerations will be accepted without reservations by everyone but, hopefully, they will generate fruitful thinking.

FWD/11: Arrangment of the HANDBOOK

All the chapters of the code will be reviewed one after the other.
When I have a comment to make on a given clause, the text of the code will not be reproduced but I will use the same reference as in the code with a letter as suffix i.e.: .../a, .../b, .../c, depending on the length of the comments I want to make.

If the reference of a given clause does not appear in the HANDBOOK, it means that I had no comment to make on the subject.

To make reading easier, I will refer to APPENDICES for all lengthy mathematical demonstrations and other references.
APPENDICES will be labelled APP:01 to APP-nn and the paragraphs identified with a digit and a letter as suffix: ex APP:06/3/b.

An APPENDIX may itself have attachments: ex Att 1 to APP:01.

FWD/12: Typical lift selected for developing calculations

Unless otherwise specified, I selected the following characteristics when I had to use a mathematical approach to analyze the behaviour of a lift:

- rated load	Q = 1000 kg	
- actual load	Qa= from 0 to 1250 kg	
- mass of empty car	P1= 1300 kg	
- mass of counterweight	CT= 1800 kg	
- linear mass of travelling cable	= 1 kg/m	
- linear mass of hoisting ropes	= 3 kg/m	(1)(2)
- linear mass of compensating ropes	= 3 kg/m	(3)
- mass of compensator	P4= 700 kg	(4)

Notes
(1) 5 or 6 ropes of 13mm in diameter are needed
each of the ropes has a linear mass of 0.6 kg/m
(2) the travel should be selected such that it will take 20 to 30 sec. at the selected rated speed.
(3) if there are any compensating ropes, we will assume that their mass is equal to that of the hoisting ropes.
(4) if there is any compensator, its mass can vary:
from 500 kg for low speed and short travel
to 1100 kg for speeds > 5m/s and travels > 100m

Other values, of lesser importance were introduced when needed and selected to be consistant with the above values.

FWD/13: References to the USA and Canadian codes

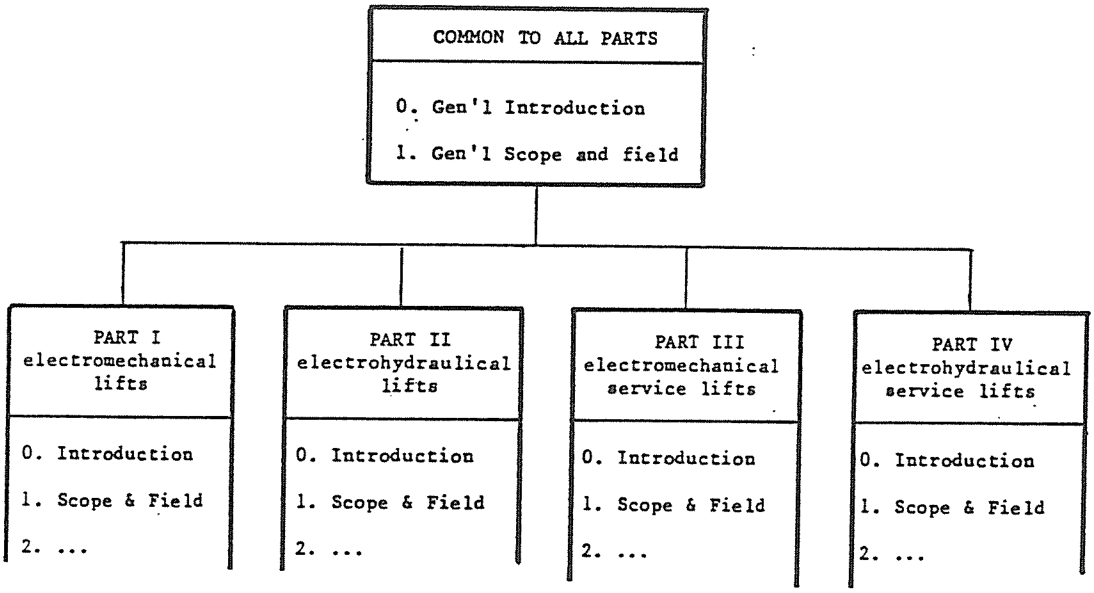
Occasionally, I will make refere ces to those other codes.
I call the attention of whom might be interested to the comparison work made by the ISO/TC 178/WG 4 and to the cross reference books they have prepared.

Their comparisons cover, not only the CEN, ANSI and CSA codes, but also the USSR and CMEA codes.

Gen'1-00/a: Presentation of the code

The presentation is a little confusing although the referencing has been improved in the new edition.

The general layout of the EN-81 is as follows:



Each Part is published separately but the common generalities are printed at the beginning of each book.

As regards the Note (1) mentioning the Interpretation Committee, see also FWD/06 in the Foreword.

Gen'1-00/b: Relativity of the concept of safety

It should be understood that the perfect safety does not exist.

A device is said to be "safe" when the probability of an accident is so remote that one can forget it.
A safety code is supposed to define when a device is safe enough, but how can the code makers know when they are requiring enough and when they are requiring too much?

Mr Vilkko Virkkala has remarked that a safety device was economically justified if its cost was inferior to the cost of the accidents it was avoiding. It may sound inhuman to talk of "cost" when dealing with human life and permanent injuries, but insurance companies are doing it all the time and they are the only source of information about these costs.

This is really the only objective approach but, unfortunately, there is a lack of statistics and, moreover, it is difficult to evaluate what the rate and gravity of accidents would be if a new device were added or if an existing device, made at least partially redundant by other technical developments, were removed.

Nevertheless, even if a strict mathematical calculation is impossible in most cases, the general principle should be kept in mind when preparing a code and trying to have consistant degrees of safety for all the devices.

This is using the same principle that this HANDBOOK proposes an approach based on probabilities for the evaluation of the relative values of electric safety devices (see paragraph ref 14.1.2.3)

Gen'1-0.1.2.1: Safety factors

On the ground of that statement, the EN-81 does normally not specify safety factors considering that the necessary information can be found in the standards relating to the structural steel or mechanical construction, the standards for the bolts industry etc..The EN 81 specify safety factors only when they are specific to the lift industry (eg: hoisting ropes).

On the contrary the USA and Canadian codes specify safety factors for all kinds of components. I suspect that the factors they mention have been taken from various other standards and that is why the presentation does vary whilst the values are very close to each other.

If there is no better information available, I think that the following values can be used:

For structures
from 4 if the material has an elongation >= 20%
to 10 if the material has an elongation < 10%

For machinery
from 8 if the material has an elongation >= 20%
to 12 if the material has an elongation < 10%

Gen'1-0.7/a: Basic assumptions

The statement in the code under this reference is exactly in the spirit of the paragraph 00/b above.

In fact, when analyzing the first edition of EN 81-Part I, the Work Group N°1 realized that additional basic assumptions were needed to insure consistency in the analyses of the various clauses during the revision which was planned. Although the revision has been postponed and although these additional basic assumptions were not intended to be part of the code, I think it is interesting to list them because most of them were already implicitly built in the first edition and because they are a necessary support for logical comments on many subjects.

The Document N° 66 of CEN/TC 10/WG 1 reads as follows:

> Spelling out these assumptions does not mean that, in case of a failure or an accident, the fact that one of these assumptions is not verified will nullify all the other assumptions.
>
> 1. A safe operation of the lift shall be assured for loads ranging from zero to 100% rated load.
>
> 2. The possibility of a failure of an electric safety device complying with all the requirements of the standard is not taken into consideration.
>
> 3. With the exception of the items listed below, a mechanical device built according to good practice and the requirements of the standard is assumed not to deteriorate to the point of creating a hazard before the failure is detected.

The possibility of the following mechanical failures shall be taken into consideration:

3.1 rupture of the suspension means,
3.2 uncontrolled slipping of the ropes on the traction sheave,
3.3 rupture and slackening of all linking by auxiliary ropes, chains and belts.

The possibility of the car (or counterweight) striking the buffer(s) in free fall before tripping the safety gear is not taken into consideration.

4. A user may, in certain cases, make one imprudent act. The possibility of two simultaneous acts of imprudence and/or the abuse of instructions for use will not be considered.

5. If, in the course of servicing work, a safety device normally not accessible to the users is deliberately neutralized, safe operation of the lift is no longer assured.

6. When the speed of the car is linked to the frequency of the mains up to the moment of the application of the mechanical brake, the speed is assumed not to exceed 115% of the rated speed or a corresponding fractional speed.

7. The following horizontal forces a person can exert shall be taken into consideration:

a) static force : 300 N
b) force resulting from impact : 1000 N.

Gen'l-0.7/b: Other guide lines

Although not retained as a basic assumption, the following limits have been used throughout the code:

- Horizontal dimensions:
0.15 m neither body nor head can pass
0.35 m a body can pass side-ways in case of emergency
0.60 m minimum width of a regular entrance.

- Vertical dimensions:
2.00 m everybody can enter without problem,
1.80 m one can enter when paying attention (emergency doors)
1.40 m a competent person can enter when paying attention
0.40 m maximum altitude of obstacle which can be stepped over by a competent person.

- miscellaneous dimensions
0.01 m gap through which a finger cannot be introduced,
0.50*0.60 m*m minimum dimensions for trap doors.

For dimensions related to the human morphology which are not specified in the EN 81, you could refer to the German standa. d DIN 31001.

Gen'l-0.7/c: Comments on some of the basic assumptions

Assumption N° 1
The USA and Canadian codes are specifying a safe operation with 25% overload for most of the important lift components.

The first remark to be made is that 25% of the rated load means only about 10% of the total suspended load and the CEN/TC 10/WG 1 considered that:

- those 10% were negligeable in comparison with the safety factors used in the design of the components,
- mentioning that the operation was safe with 125% of the rated load could induce people systematically to overload the car.

I am inclined to agree with the CEN reasoning.

The second remark to be made is that, for the traction calculation and the brake requirements, CEN is mentioning 25% overload. In that case it is not compounded by other safety factors. However, in the case of the traction calculations, there are hidden safety margins in the selection of the friction factor as will be explained in the paragraph HB/9.Notes.
In the case of the brake, the origin of the extra safety depends on the type of motor drive as explained in paragraph HB/12.4.2.1.

<u>Assumption N°3 exceptions 3.1 and 3.2</u>
The probability of suspension failure is controversial.
The CEN/TC 10 did ask the opinion of all member countries and the majority was in favour of keeping the safety gear as a protection against the possibility of free fall or uncontrolled rope slipping.

I am personally convinced that:

- the few suspension failures which could be reported did occur on lifts built according to old codes,
- as far as I know, all the rope slippings reported did occur with lifts where, for a given rated load, the car area was much larger than the one allowed by EN-81, as it was customarily the case with appartment lifts in the nordic countries, or with freight elevators.
- unwanted accidental safety settings create a risk which is not compensated by what would be saved by the safety gear in the case of hypothetical suspension failures or rope slippings.

I am convinced that there is practically no risk of total suspension failure if all the equipment meets the specifications of the code and if the lift is regularly maintained and inspected.

As regards the possibility of rope slipping, I am also convinced that the risk is infinitesimal for passenger lifts meeting all the code specifications.I explain why in the paragraph related to traction and if the more refined method of calculation I am proposing was adopted, therewould practically be no possibility of rope slipping.

I am not as positive for freight elevators.

This question should be reconsidered periodically because, if it werefinally recognized that the probability of accident is low enough to be disregarded, the design of the lifts could be dramatically modified (deletion of the safety gear and of the mechanical overspeed governor, change in the design and selection of the guide rails, of the compensator).

Incidentally, it should be remarked that code makers have a natural tendency to add new requirements without removing the ones which become eventually redundant.They fear being accused of advocating lesser safety but the principle exposed in paragraph HB/00/b "Relativity of the concept of safety" should apply for old-time devices as well as for new ones and their justification reconsidered from time to time with the evolution of materials and of other coexisting safety requirements.

Assumption N° 6

The value of 115% was selected to be in line with values mentioned in various places in the code and applicable to all lifts, including the variable speed drives.

In fact, for the lifts envisaged in the assumption N° 6, experience shows that the speed will not exceed 105% of the speed measured as described in clause 12.6 of the code. This, combined with the 5% tolerance allowed in the clause, would lead to a maximum of 110% of the rated speed.

2: Additional definitions

2/a: The code does not give definitions for electric devices.

For convenience, I would remind you that there are:
- electric safety devices divided into:
  - electric safety contacts
    (because of the mechanical design, the contact cannot be established if the safety condition is not fulfilled)
  - electric safety circuits
    (because of redundancy or cyclic checking, the probability of having a dangerous condition whilst the circuit is out of order is so low that it can be disregarded)

- electric "protective" devices
  (since there are no specifications for those devices, the probability of having a dangerous condition whilst the device is out of order depends solely on the quality and the life time of the device combined with the frequency of inspections).

It should be noted that the American codes are using the wording "protective device" for what we call "safety contact".
It has been suggested that CEN replaces "protective device" by "guarding device" for avoiding confusion.

The concept of the "electric safety circuits" has not been introduced in the North American codes.

2/b: In many places, the code refers to a "competent person".

A competent person is somebody who knows what he is doing.

In most cases, the term will refer to somebody having a professional expertise in the construction, operation and maintenance of the specific lift under consideration, knowing the dangers involved and knowing what to do in case of shutdown or malfunctioning.

In a limited number of cases. the term might refer to somebody having received a minimum of training for making emergency rescue of trapped passengers and having enough maturity for not putting his fingers where he is not supposed to. By "limited number of cases", I refer to the EN-81 clauses 6.1.1 , 6.3.2 (1st alinea), 6.4.2.1 , 7.7.3.2 and 12.5.
I say "might refer" because, in some countries, only people having the full training are allowed to perform even these simple emergency operations.

4.2: Symbols

In many places, the code proposes general formulae.
In order to develop formulae more readily applicable, I have used the following additional symbols:

- actual load in the car (mass)	Qa	kg
- travel	H	m
- diameter of the smallest pulley	D1	mm
- number of buffers under the car	n1	(1)
- number of buffers under the counterweight	n2	(1)
- number of car guide rails	n3	(1)
- number of counterweight guide rails	n4	(1)
- mass of a rope per unit meter	m	kg/m
- minimum breaking strength of a rope	cc	N
- roping ratio	rs	(1)
- mass of the empty car including sling,doors etc	P1	kg
- mass of the travelling cable(s)	P2	kg
- mass of the compensating ropes (or chains)	P3	kg
- mass of the compensator	P4	kg
- mass of the hoisting ropes (or chains)	P5	kg
- mass of the counterweight	CT	kg
- angle of the throat opening (circ. grooves)	Teta	rad
- Factor depending on the shape of the groove	C3	(1)

Note: P2 (or P3) is the mass of the section of ropes hanging from the car (or counterweight) when at the top end of its travel.

LIFT WELL

5.1: General provisions

The international standard ISO 4190/1 fixes, for rated speeds up to 2.5 m/s, the dimensions to permit the installation of passenger lifts of:

- class I : lifts designed for the transport of persons.
- class II : lifts designed mainly for the transport of persons but in which goods may be carried.
- class III : lifts designed for the transport of beds (hospitals)

The international standard ISO 4190/2 fixes, for rated speeds up to 1 m/s the dimensions to permit the installation of lifts of:

- class IV : lifts designed mainly for the transport of goods which are generally accompanied by persons.

These standards have been prepared to meet the requirements of EN 81 and their use is recommended. Several countries have published them under their own reference system but always mentionning the ISO origin. Occasionally, there are some identified national deviations.

5.2.3 Ventilation of the well

The architects and consulting engineers should pay attention to these requirements because this concerns not only the comfort of the pasengers in the car but also the system of protection of the building against fire as a whole. The requirements for the fire tests of lift doors are assuming that the well is correctly ventilated.

5.7.1.3: Calculation of 0.035*v2 in the case of short stroke buffers

The condition (a) is practically meaningless because of the specified minimum of 0.25 m.

Indeed, if, using the 50%, you calculate the speed corresponding to 0.25 m by the formula

$$1/2 * 0.035*v2 = 0.25$$

you find v = 3.78 m/s which means that you are already on the verge of being allowed to use the reduction to 33%.
This appears clearly on the graph of Note 3.

Practically, it means that for speeds higher than 2.5 m/s and under the conditions laid down in EN/5.7.1.3, the value may be reduced to as low as 33% of 0.035*v2 with a minimum of 0.25 m.

Clause 5.Note 1/a: Calculation of vertical forces on guide rails

The formulae proposed by the code are based on the following assumptions:

- there are always 2 guide rails
- the deceleration is, in case (a)(1), 4 gn (4*9.81 m/s2)
  in case (a)(2), 2 gn (2*9.81 m/s2)
  in case (b) , 1 gn (9.81 m/s2)
- a compensator is never used with instantaneous safeties.

We shall, using the additional symbols proposed in HB/4.2, develop detailed formulas for the application in each of the cases and, at the same time, analyze the validity of the assumptions and their consistency with some mathematical developments made in other clauses of the code.

Clause 5. Note 1/b: <u>Instantaneous safety gears other than roller types</u>

b.1) For application, the formula of the code can be made explicit as follows:

25*(P1+P2+P3+Q) Newtons applied on each car guide.

25*(CT+P3) Newtons applied to each counterweight guide.

b.2) I have no direct or indirect experience with that type of safety but, obviously, all the comments made on the roller type in the next paragraph would apply with amplification.

Clause 5. Note 1/c: <u>Instantaneous safeties gears, roller types</u>

c.1) For application, the formula of the code can be made explicit as follows:

15*(P1+P2+P3+Q) Newtons applied on each car guide.

15*(CT+P3) Newtons applied on each counterweight guide.

c.2) The above formula has been used for many years by some manufacturers and no bad experience has been reported.

Yet,- if free fall was really happening as retained in the basic assumptions (see HB/00.7/a:3.1),
- if the bases retained by the code for the evaluation of the capacity of the safety blocks were correct (see EN/F.3.2.4),

the forces applied to the guide rails would be much higher than the ones calculated by the above formula.

In all the test reports I have seen for roller type safety blocks, the diagrams "Force/Travel" were linear.
Let us try to figure out what the travel during the safety operation should be if the total reaction on the guides was not to exceed 3 times the weight of the suspended mass in the case of free fall with an overspeed governor adjusted for triggering at 1 m/s and with the response time and distance for taking up clearances envisaged in EN/F.3.2.4.

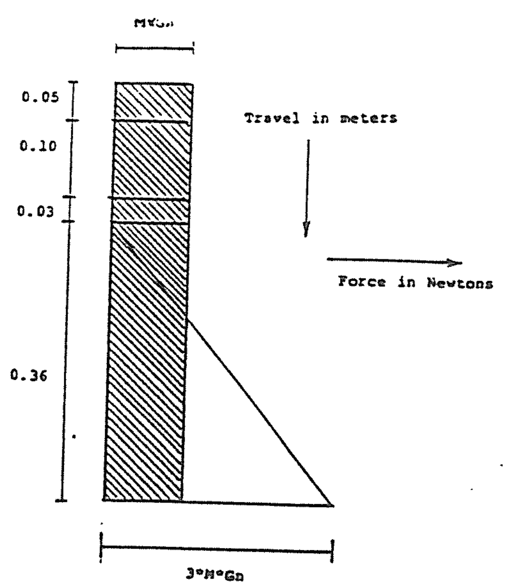


Fig. 3

To have the blocks absorbing not only the kinetic energy but also the potential energy lost during operation, the shaded area must be equal to the triangle. This is how the 0.36 m are calculated.

One can see that the distance which would be needed, 0.36 m, is an impossibility. If you ask professionals what distances they have noted in case of actual safety settings, you will get answers in the order of magnitude of 0.036 m i.e. ten times less.

If you have a look at the analysis made in HB/F.3.2.4, you will see that, with an actual safety declared good for 5000 kg suspended load after testing, the travel during safety operation would be 0.048 m and the maximum reaction force 48000*9.81 Newtons instead of the 15000*9.81 Newtons coming out of the above formula.

So, the least that can be said is that the formula in EN/Clause 5.Notes is not consistant with the mathematical approach developed in EN/F.3.2.4. Nevertheless the experience seems to indicate that both the guide rails and the safeties do behave satisfactorily. How does this come about?

First of all, as I explained in HB/00.7/c (comment on assumption N° 3.1), I strongly believe that suspension failures practically never happen.
If, with a lift built according to an older and less complete code, it did happen once, it was not necessarily with a full load and the safety gear might have set by inertia because all overspeed governors have a certain amount of inertia whether required by the code or not.
This means that, under normal conditions, the safety block will never experience a demand as severe as the one taken into account for testing.

However, manufacturers have made drop tests from time to time either to learn something or to satisfy some suspicious inspectors.

I think that, in fact, some of the energy is accumulated or dissipated in the guide rails, in the brackets and in the building itself which smooths the peaks. What part of the energy can be disposed of that way is not known; it is certainly bigger with long than with short guide rails.

In the case of big freight elevators with extremely short travels, I would recommend, to be on the safe side, that some resilient material be installed where the rail does transmit the force to the building.
An alternative would be to insert the resilient material between the car frame and the safety beam; in that case, the safety would work more or less as if it had a buffered effect.

Clause 5.Note 1/d: <u>Instantaneous safety gears with buffered effect</u>

d.1) Although not mentionned in EN/Clause 5, you may use the same formulae as the ones proposed in the preceeding paragraph for instantaneous safety gears (roller type) whatever the type of instantaneous blocks are used.

d.2) It should be noted that, in this case, the evaluation is much safer than if there were no buffer(s) because of the specifications of the buffer(s) (see EN/9.8.6.2, EN/10.4.2, EN/10.4.3).

Indeed, the total buffer(s) reaction on the safety beam cannot possibly exceed 35*(suspended mass) for more thai 0.04 sec and the average will be 20*(suspended mass), those values to be divided between the 2 rails.

Most of the energy will be dissipated in the buffer(s) but a small fraction will be accumulated in the safety blocks and even if only a very small part goes into the rails,the brackets and the building, it will be enough to assure that the peak forces remain below the expected limit.

Clause 5. Note 1/e: Progressive safety gears

e.1) In this case, it should be noted that there is no technical reason preventing the use of more than 2 guide rails. It may be an advantageous solution in the field of big freight elevators.

Several combinations are even possible, but we will assume that:
- all guides are identical,
- all safety blocks are identical.

It should also be noted that, in the case of progressive safety gears, the braking force will be the same whether the car is loaded or not, whether there is a compensator or not: only the time during which the force is applied will change.

For application to the car rails, the formula of the code can be made explicit as follows:

(20/n3)*(P1+P2+P3+Q) Newtons applied to each guide.

where (n3) represents the number of car guide rails.

For application to the counterweight rails, the code does not propose a formula but the following one may be used:

(20/n4)*(CT+P3) Newtons applied to each guide.

where (n4) represents the number of counterweight guide rails.

e.2) Here also, it should be noted that the peak forces will exceed the above value as soon as the tripping speed exceeds 2 m/s if the brake shoes are made of cast iron (see Fig 4) but only during a very short time.

The Fig 5 shows how the speed and the deceleration do vary in relation with time in the case of a setting speed of 5 m/s.
Whereas the phenomenum lasts 0.8 sec, the deceleration exceeds 1 gn only during 0.1 sec and this peak would be reached only if everything was absolutely rigid both on the car side and on the guide rails side.
In fact, as for the instantaneous safety gears, this peak is most certainly beheaded due to the elasticity of the guide rails, the brackets, the building and also the car frame with its rubber pads.

Clause 5. Note 2/a: Reaction of rails on the pit floor

As already explained above, those values are conventional.
Whether the rails are resting on the bottom or suspended from the ceiling, part of the reaction is supplied by the brackets which is dramatically illustrated by the fact that, in the case of high rise lifts, it is current practice to leave the rails "floating" using even devices to limit the reaction of individual brackets.

However, as already mentioned in HB/Clause 5. Note 1/c, in the case of short travels one cannot expect the brackets to take much load because their number is very low.

To summarize, I would recommend special consideration for the cases of abnormally short travels not only for instantaneous safeties but also for high speed lifts (example: test towers).

Clause 5. Note 2/b: Reaction of buffers on the pit floor

b.1) In the case of energy accumulation buffers, the maximum reaction is defined in EN/10.4.1.2 (4 times the weight of the car or counterweight) and ,here, (gn) is rounded to 10 m/s2.

In HB/10.4.1.2, I will explain that this is a maximum because the spring will never be fully compressed.

b.2) In the case of energy dissipation buffers, the clause EN/10.4.3.3 specifies that the deceleration shall not exceed 2.5*(gn) during more than 0.04 second with the rated load.

The 2.5*(gn) correspond to a reaction of 3.5*9.81*(falling mass) so the reaction to be taken into consideration can safely be estimated to 40 times the falling mass because any peak exceeding this value will necessarily be of very short duration and probably be dissipated in the car frame, car insulation and pit floor.

b.3) The proposed formulae give the total reaction. Assuming that, if more than one buffer is used, they are identical and designating their number by (n1) for the car and (n2) for the counterweight, the formulae of the code can be made explicit as follows for application:

a)beneath each car buffer(s) supports:

(1/n1)*40*(P1+Q) Newtons

b)beneath each CT buffer(s) supports:

(1/n2)*40*(CT) Newtons

b.4) Those who are familiar with the ANSI code have perhaps noticed that ANSI is using a calculation method involving the energy for finding the reactions. I do not see why one should complicate things since the codes do give directly the forces or the decelerations.

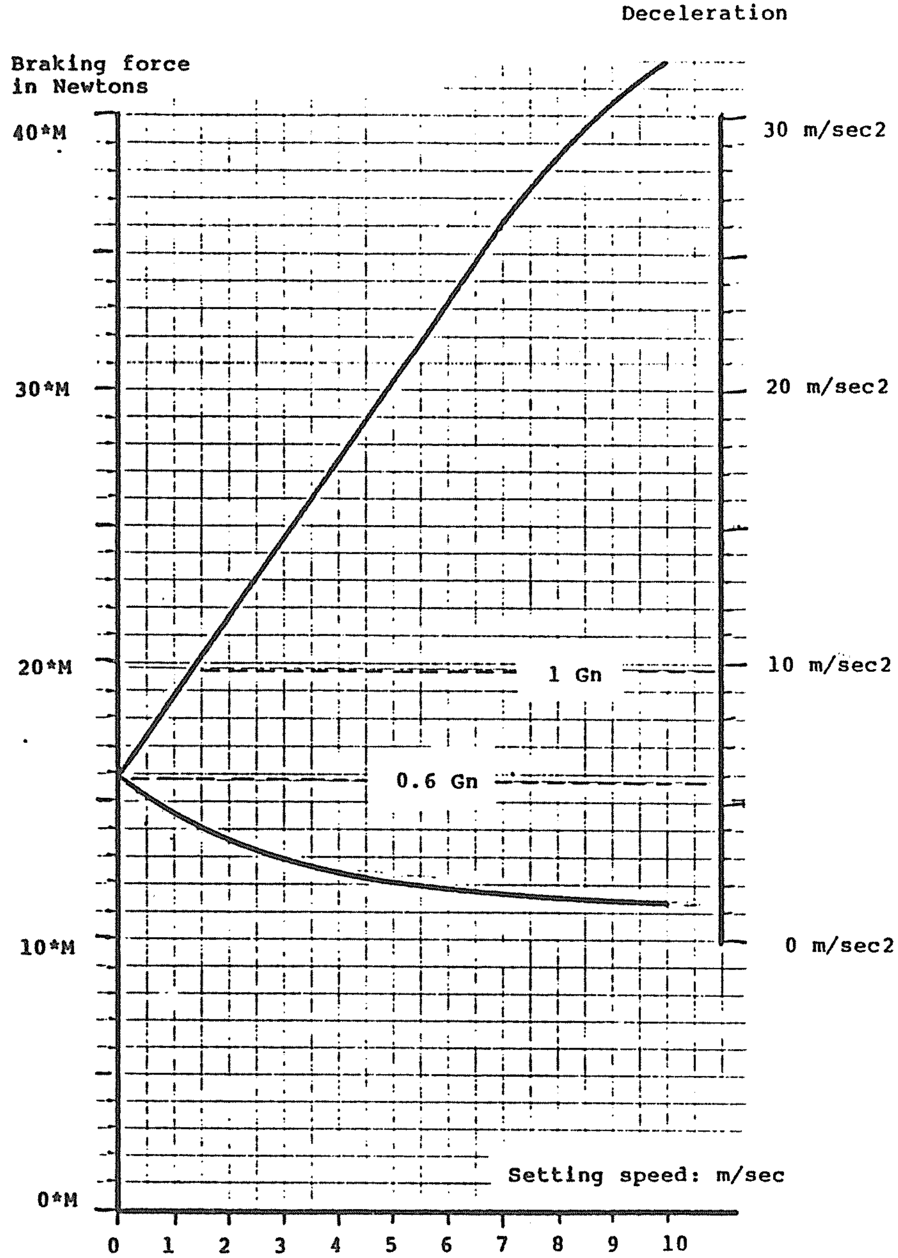


Fig 4 - Minimum and maximum instantaneous decelerations for safety setting with mean deceleration of 0.6 Gn for setting speeds up to 10 m/sec

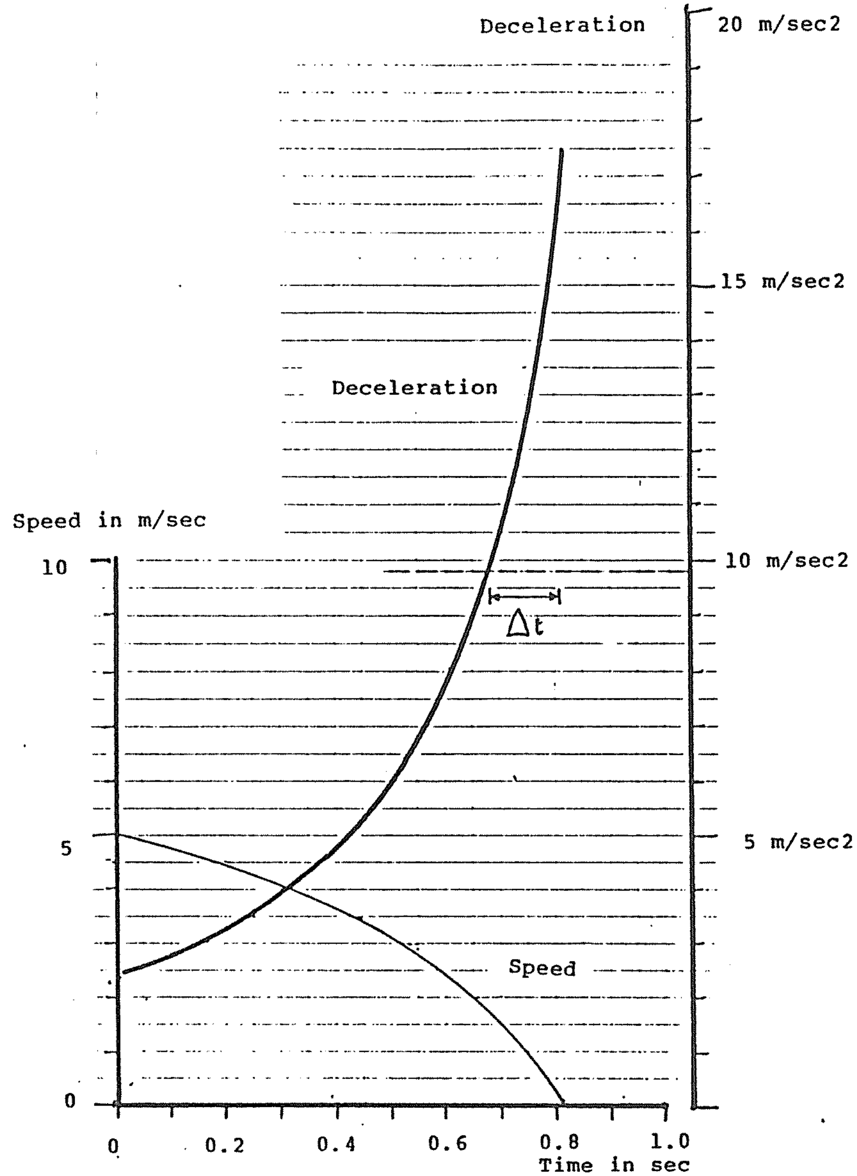
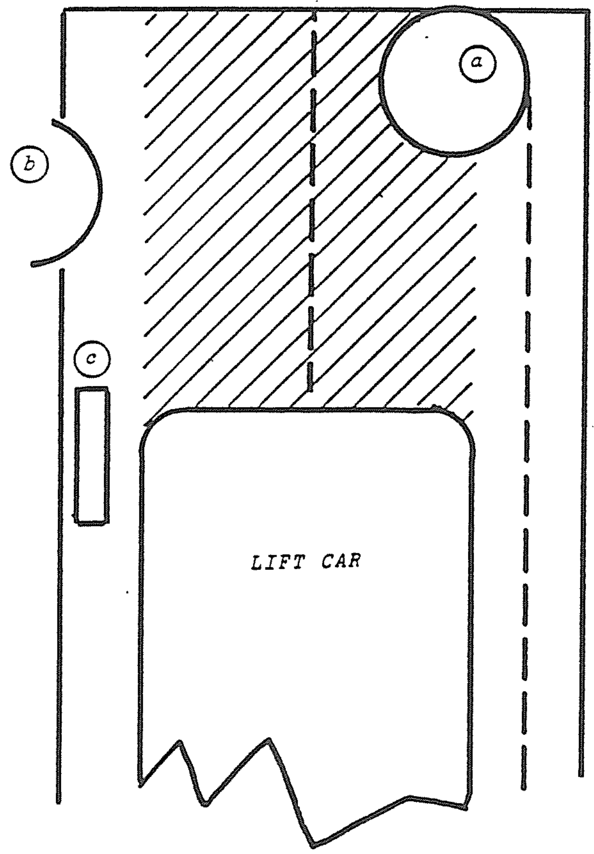


Fig 5 - Speed and deceleration related to time during safety setting with an initial speed of 5 m/sec.

6.1.2: Pulleys in the well

This innocent looking clause has been the subject of one of the longest and least interesting discussion I can remember. It would not be wise to reopen the discussion, so I am not going to propose any argument pro or contra.

The requirements as regards the location are illustrared Fig 6.



No pulleys are allowed in the shaded area

The only pulleys tolerated in the well are:

- (a) *a diverter pulley, single or double wrap, for diverting towards the counterweight*
- (b) *traction sheave, deflection and diverter pulleys providing they do not intrude in the shaded area*
- (c) *same as* (b)

All these pulleys shall have the required protective devices. Examination, tests and servicing shall be possible in complete safety from the car roof or from outside the well

Figure 6

I will, nevertheless, try to summarize the opinion of the majority of CEN/TC 10 members as well as I can.

1) It is definitively better to have the machine room on top of the well.

2) Although locating the diverter pulley (if any) also in the machine room would be better, it is considered as common practice to limit the height of the machine room by locating the diverter pulley in the hatch if it leads towards the counterweight (see (a) fig 6).

3) The architects should not be encouraged to locate the machine room on the side of the hatch (neither top nor bottom) but if this is unavoidable, the arrangments (b) and (c) have to be tolerated.
It is considered as an iuferior design.

6.3.1.3 Noise level requirements

It must be understood that making a noiseless lift is impossible.
An acceptable installation is made of a reasonably quiet lift and of a suitable building arrangment.
It has often be remarked that, even without special care, a lift generates less noise than a toilet flush or than the street traffic.

The reason for the (N.b) is that some countries have issued regulations or standards on the subject. Unfortunately, most of the time only the final result is required and the responsibilities are not divided between the lift manufacturer and the other parties (architects, consulting engineers and building contractor).

As far as I know, the countries enforcing regulations about noise levels are Germany, France and the Netherlands. Belgium has some "Norms" on the subject but it has not been an issue so far.

The noise problem can be a source of dispute between the various parties involved. In Appendix 2, you will find some hints for architects and lift contractors. They are not the answer to all the problems; they merely indicate what to avoid and give some guide lines on what to do.
All that can be said is that, for appartment buildings, following these simple advices has generally avoided conflicts.
For critical buildings such as hotels, hospitals, libraries, etc, more sophisticated noise barriers might be needed.

6.3.2 Dimensions of machine rooms

The international standard ISO 4190/1, already mentioned in HB/5.1, gives indications for the machine rooms (see HB/5.1 for more details).

6.3.5.1 Ventilation of machine rooms

Special attention must be paid to a correct ventilation of the machine room, especially if the well is ventilated through the machine room for the reason explained in HB/5.2.3.

6.3.5.2 Temperature of the machine rooms

The attention of architects must be drawn on the fact that the performances and even the functionning of the lifts are not guaranteed if the temperature falls below 5° C or exceeds 40° C.
Electronic components may be particularly sensitive to high temperature.

The same limitations apply to the so-called "Fireman's Lift".

6.4 Pulley rooms

Pulley rooms are a rarity nowadays. They would be almost solely needed if the machine room is not located above the well, which is not recommended and, consequently, not even envisaged by the ISO standards.

Even when the building limitations make it impossible to locate the machine room where it should be, manufacturers usually get by without pulley rooms using the exceptions tolerated in EN/6.1.2.1.

In the case of big goods lifts (freight elevators) with low rated speed, which almost necessarily use multiple roping, the reeving pulleys can usually be located in the machine room.

7.2.2: Behaviour under fire conditions

It must be stressed that lift doors cannot play the role of fire doors.

The role that lift doors have to play under fire conditions has not been clearly defined in many countries or had different definitions depending on the country under consideration.

This indicates that firemen and authorities involved in protection against fire could not agree on what the problem was and logically were proposing different solutions. Also, in each country, the fire test laboratories were geared to the local Credo and did not want to lose the business of testing the lift doors according to the local requirements.

All this explains why the fire test procedure proposed in EN/F.2 of the first edition of the EN code had so little success.

This diversity of requirements does create technical barriers to trade and is directly against the objective of the E.E.C. Directive.

In an attempt to solve the problem, the CEN/TC 10 established a special Work Group, the WG 3, with the participation of, beside lift experts, representatives of fire protection authorities and fire tests laboratories.

This group has not, to date, finalized its work but has already agreed on a number of basic assumptions and definitions.
Hopefully, they will have a project ready by the time the WG 1 has completed the in-depth revision of Part 1 (which is to be done after the vote of the amendment of Part I and the vote of Part II).

Manufacturers would be happy with any testing requirements providing all countries could agree with them. However they have indicated many times that, in their opinion, any plain door retaining enough integrity to protect people against falling in the hatch would be good enough.
There is no single report that a lift door, providing it was plain and stayed in place, did contribute to the spreading of fire from floor to floor in a building.

7.2.3 Mechanical strength

7.2.3/a) To meet the requirements of the code, the door must resist when a force of 300 Newtons is applied in certain conditions.

The North American codes require that the door resists a force of 1130 Newtons without specifying elastic or permanent deformation.

In fact, if a safety factor of 5 is used, a component resisting 300 Newtons with only elastic deformation will usually resist 1000 Newtons with, possibly, permanent deformation but other things (such as the fixing to the walls) will not necessarily resist.

7.2.3/b) Because they had registered a few fatal accidents due, I believe, to wild horse-play, the Canadians have developed a new testing procedure.

Recognizing, like the CEN/TC 10/WG 1 (see Basic assumtion N° 7 under the reference HB/0.7/a), that the danger would come from people hitting the door at a certain speed, they made tests using a mass consisting of an elastic envelope containing sand or water to reproduce as much as possible the mass and consistence of human bodies.

A mass of 200 kg was selected to represent the bodies of 2 men running in the corridor and hitting the door.

Hanging this mass like a pendulum, they let it bang against the door at a speed of 2.78 m/s corresponding to a moderate running speed of 10 km/h.

After much testing and analyses, they came to the conclusion that testing each type of door with this system would not be practical but that adequate requirements based on static forces could be the answer to the problem.

The following requirements have been introduced in the CSA/B44 code:

a) Requirements for the "primary fastening means"
(CSA ref 2.11.10.4.7)
"Panels, hangers, guides and guide shoes shall be capable of withstanding a force of 2500 N applied on the landing side at right angles to and approximately at the center of the panel.
This force shall be distributed over an area of 100*100 mm.
There shall be no appreciable permanent displacement or deformation of any part of the entrance assembly resulting from this test and the door shall remain operational after the force has been removed."

b) Requirements for the "safety retainers"
(CSA ref 2.11.10.5.1)
"The top and bottom of horizontal slide landing doors shall be provided with means for retaining the door panel in position should the replaceable primary guiding means...fail, and prevent displacement of the door panel top and bottom by more than 20mm when the door panel is subjected to a force of 5000 N applied at right angles over an area of 300*300 mm at the approximate center of the panel."

There are many more details in the Canadian code but I want just to mention the forces which are taken into consideration.

If we compare now the forces selected in the various codes, we shall see that they vary widely:

EN-81	300 N
ANSI	1130 N
CSA	2500 N (primary means)
	5000 N (safety retainers)

On the European market, lift contractors are bound only by the EN code and I have personally not heard of doors falling in the well.
However, I also believe that experienced door manufacturers make their doors stronger than strictly required by the code.

In Canada, existing doors, built according to the previous local code and tested until failure, were able to withstand 4000 to 5000 N before rollers failed (it seems to be always the weak point).
So their new requirements do not seem to create big problems.

Manufacturers would be wise to keep these figures in mind when designing new doors. Let us remember that Americans and Canadians have more experience than we Europeans with big crowds in large halls and corridors.

7.5.2.1.1.2 Limitation of kinetic energy for the panels

7.5.2.1.1.2/a) Measuring the average closing speed of the leading panel, according to the EN/Foot note (2), is relatively easy. However, one should not forget that all the parts rigidly connected to the leading panel are not moving at the same speed (low speed panels, linking arms etc).

So, when calculating the kinetic energy, each mass must be associated with the corresponding speed, and the rotating masses of the driving mechanism should not be forgotten. It is easy for the manufacturer to know these masses, but not so easy for the inspectors wishing to ensure that the door satisfies the requirements of the code.

7.5.2.1.1.2/b) This is why a measuring method has been proposed in the EN/Footnote (2) for measuring the kinetic energy of the whole assembly.

However, to be sure to measure the same thing as that defined as the base for the calculation, some precautions must be taken:

a) the measuring device shall be equipped with an extension such that the impact will take place when the instantaneous speed is exactly equal to the average speed defined in footnote (2),
b) the protective device required in EN/7.5.2.1.1.3 shall be disconnected,
c) another switch shall be installed just cutting the power from the door operator at the time of hitting the measuring device or just before (no reopening and no braking).

7.5.2.1.1.2/c) Validity of the limits selected in the EN-81.
These limits are the same as the ones used in the ANSI and CSA codes, however they have been recently questioned and the ISO/TC 178/WG 4 has started investigating the subject.

No conclusions have been reached so far, but I do recommend reading the very good presentation made by Heikki Nykanen on the problem at the International Conference "Lift 2000", in Budapest, and communicated to the WG 4 of ISO/TC 178. (see APP:09/HB)

7.5.2.1.1.3 Protective device for reopening the door

7.5.2.1.1.3/a) The code does not require any "comfort level" for the assembly of the door, its door operator and its protective device.

Some systems are, without any question, smoother than some others.

In my opinion, the measuring device described in footnote (1) could be used to evaluate the "comfort level" providing that:

a) the extension mentioned above be adjusted so that the impact happens at the maximum speed of the leading panel,
b) the protective device be maintained in service and the rigging be such that it is operated.

The best "door systems" would be the ones where the lowest values would be measured using this procedure.

7.5.2.1.1.3/b) Although not specified in the code, the inertia should be limited to a low value during the last 50 mm of travel of each door panel because this is practically the only way of limiting the harmful consequences of pinching fingers.

7.7.2.1 Condition for running the lift

The condition should preferably be that the panels be locked and not merely closed but, in the case of "indirect linkage" where only one panel is locked, this is possible only if the integrity of the linkage is checked instead of only the closed position of the non-locked panel(s).
(see HB/7.7.6.2 below)

7.7.3 General comment about locking devices

In my opinion, the whole door system contributes to the "locking" function and a lock should never be tested without the door to which it is to be associated. This is particularly true for multipanels doors where the panels are not individually locked.

7.7.6.2 Multipanels with "indirect" linkage

The word "indirect" is perhaps confusing.
In fact, it refers to a linkage using one of the elements for which the possibility of failure has to be taken into consideration (i.e. ropes, chains or belts according to the Basic assumptions; see HB/Gen'1.0.7/a).
Besides that, the linkage is as direct as any other one.

Instead of only proving the closed position of the unlocked panels, it would be better to prove the integrity of the linkage because this would prove that the panels are locked and not only closed.

The whole system, including the linkage, shall be tested as per EN/F.1 which will guarantee an acceptable lifetime.

The attention is also called to the Interpretation N°100.
If there are hooks or stops which make sure that all panels are necessarily in place when the door is locked, the linking is considered as "direct" even if the cinematic of the system is based on belts ropes or chains.

7.8 Closing of landing doors

It is important to make sure that landing doors, whether automatic or not, are closed when the lift is not loading or unloading because this is one of the assumptions made for analyzing the behaviour of the lift in case of fire and establishing the requirements for the lift doors.

In the case of automatically operated doors, the control shall comply with the requirements of EN/7.8.

In the case of other doors, there should be preferably a door closer or at least very visible instructions at each floor.

7.9 Controls and signals associated with landing doors

There is no EN/7.9 but I think it proper to call attention here to the requirements of EN/14.2.

I recommend also the use of the ISO standard 4190/5 "Control devices, signals and additional fittings" which gives very useful additional details for the various cases of application.

8.2.1: Car dimensions

The recommended dimensions are given by the following ISO standards:

- ISO 4190/1 : Class I lifts (passengers lifts)
  Class II lifts (passengers-goods lifts)
  Class III lifts (hospital lifts)
- ISO 4190/2 : Class IV lifts (goods lifts = freight elevators)
- ISO 4190/3 : Class V lifts (service lifts = dumbwaiters)

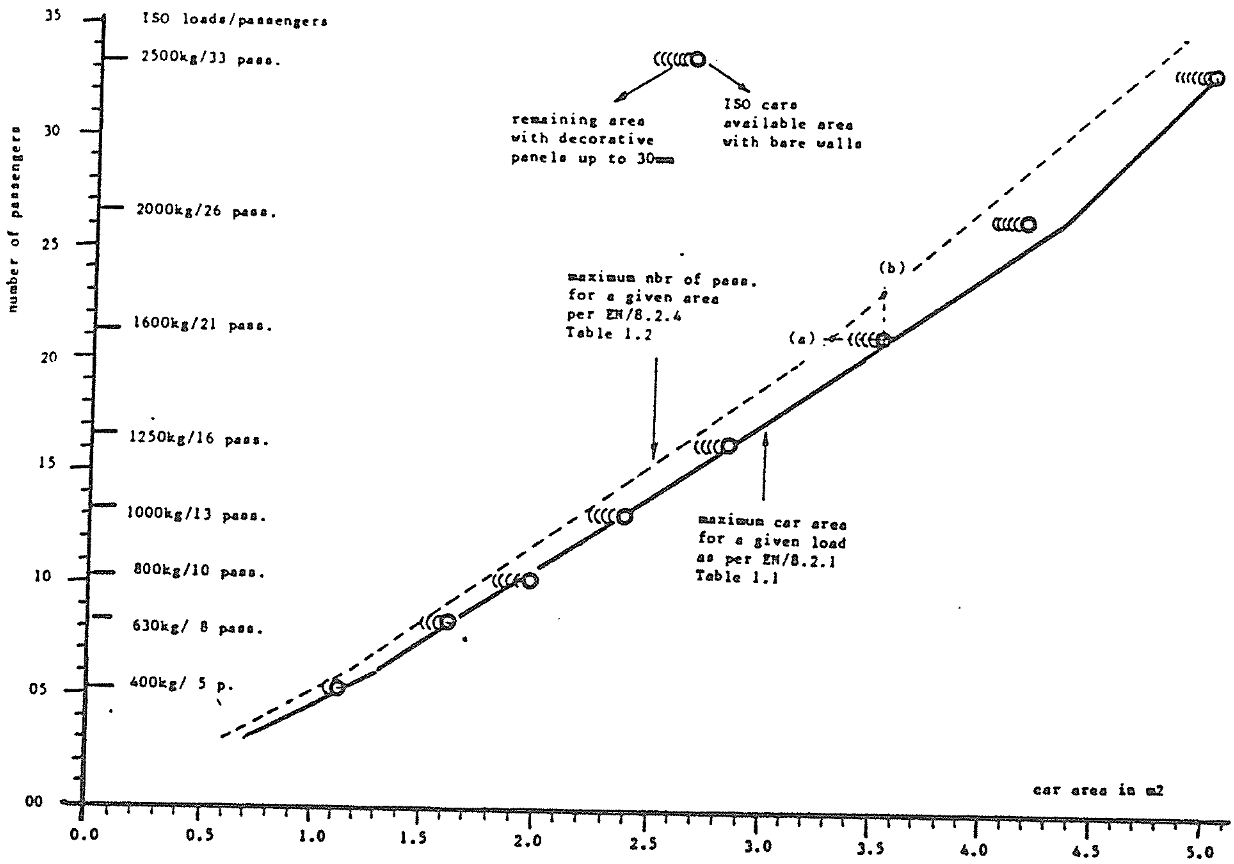
The ISO dimensions are based on the CEN specifications but the ANSI requirements are very close.

8.2.4 Maximum number of passengers

8.2.4/a)
The CEN/TC 10/WG 1 correctly decided to allow some margin to avoid that the carrying capacity of a lift expressed in number of passengers ought to be modified if the architect decided to change the decoration of the walls.

For example, the ISO lift 1600kg/21 passengers has exactly the available area corresponding to Table 1.1 of EN/8.2.1 when the car walls are bare. Thanks to the margin resulting from the table 1.2 of EN/8.2.4, the same lift may still be called 1600kg/21 passengers when 3 of the walls are decorated with panels 30 mm thick.

The Figure 7 illustrates that margin for car areas up to 5 m2 and rated loads up to 2500 kg/33 passengers.
All the ISO loads for Class I, II and III lifts are indicated.



The Figure 8 shows how the available area has to be calculated for CEN in relation with the Width and Depth as defined by ISO.

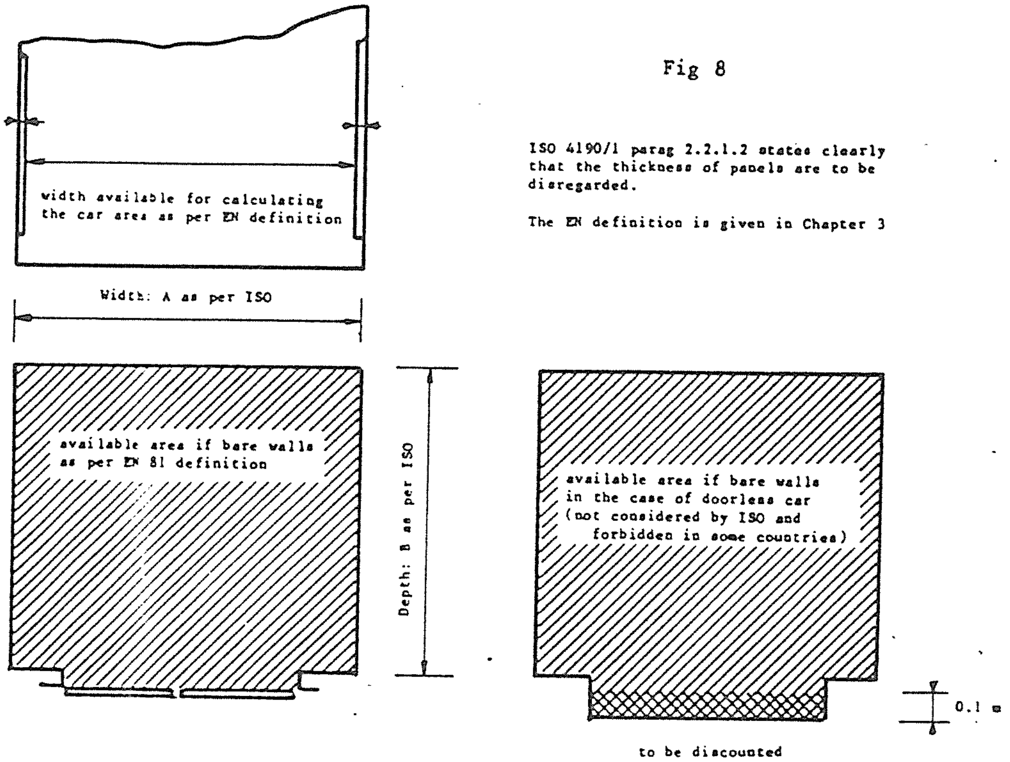


Fig 8

ISO 4190/1 parag 2.2.1.2 states clearly that the thickness of panels are to be disregarded.

The EN definition is given in Chapter 3

8.2.4/b)
It is to be feared that some smart fellow will take advantage of the way CEN has decided to express the above margin.

Indeed, still taking the example of the ISO 1600kg and supposing that a car with bare walls is acceptable, one can propose:

- either a 1600 kg/21 pass. using a smaller car area (3.3 m2 versus 3.54) (see point (a) of Fig 7) and a smaller well,
- or a 1750 kg/23 pass. using only the available 3.54 m2 (see point (b) of Fig 7) but giving a higher traffic capacity.

Reputable manufacturers will avoid using such tricks and consulting engineers and architects should discourage lift contractors of doing so. Competition on such bases leads to acrobatic designs and, ultimately, to lesser quality and lesser freedom for the architects.

The best way to avoid this problem is to use the ISO Standards.

If,exceptionnaly, the ISO Standards cannot be used, I suggest that the EN/8.2.4 rule be interpreted in the following way:

The number of passengers corresponding to a given area is given in the following tabulation (linked to EN/8.2.1 Table 1.1):

In line with the ISO standard, hand rails may be disregarded for the evaluation o' the available area. Decorative panels may also be disregarded if their thickness does not exceed 30 mm.

This number of passengers is to be used for the traffic calculations and may be indicated as guide line on the name plate remembering however that the rated load, expressed in kg, is the leading indication and the only contractual one.

car area in m2	number of pass.	car area in m2	number of pass
0.7	3	2.35	13
0.9	4	2.4	13
1.1	5	2.5	14
1.17	5	2.65	15
1.3	6	2.8	16
1.45	7	2.9	16
1.6	8	2.95	17
1.66	8	3.1	18
1.75	9	3.25	19
1.9	10	3.4	20
2.0	10	3.56	21
2.05	11	3.88	24
2.2	12	4.36	28
		5.0	33
Beyond 5 m2, add 1 passenger for each 0.12 m2			

Fig 9

8.2.4/c)
The EN code admits implicitly that a car can be overloaded by 4.3% with no noticeable discomfort for the passengers (0.12/0.115 = 1.043).

Laboratory experiences conducted in the States half a century ago indicated that people could squeeze themselves into a car up to 32% overload and that 42% could even be reached but I suspect that it needed some exterior help as in the Japanese subway.
It was decided that the probability of exceeding 25% was so low that it was not worth considering.

Reckoning that an overload in the car means only a minor addition to the total suspended mass, the CEN philosophy is that the usual safety factors used in designing the machine elements and all structures can largely take care of that. There are however 2 exceptions where these 25% had to be specifically introduced in the calculations: the brake and the traction.

The CEN/TC 10/WG 1 is of the opinion that calling the attention to the fact that the lift is designed for 25% overload would induce people to overload it systematically.

8.3.2.1 Mechanical strength of the car walls

All the considerations developed in HB/7.2.3 also apply here.

8.4.2 Height of the toe guard

The reason for the high apron is to limit the danger of having a passenger,trying to escape from a stalled car, falling in the well.

The Fig 10/a shows that, with a small apron, the passenger can practically not avoid curling himself around the sill if the car is stalled between 1 and 1.5m above the landing floor.

To the contrary, the 0.75m apron will either fill the gap or provide a point of support for the knees (see Fig 10/b)

Fig 10/a

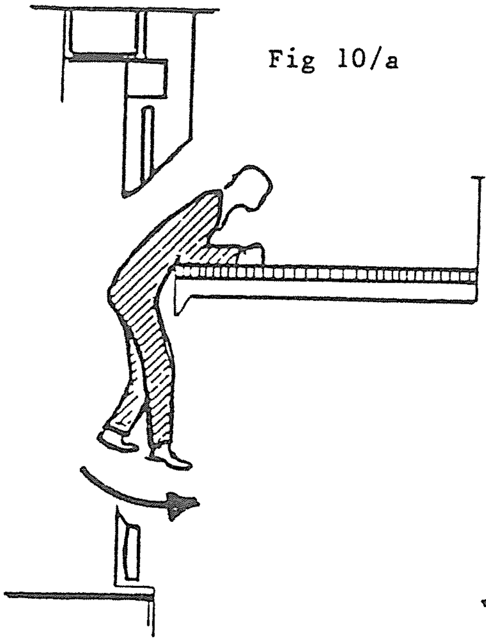
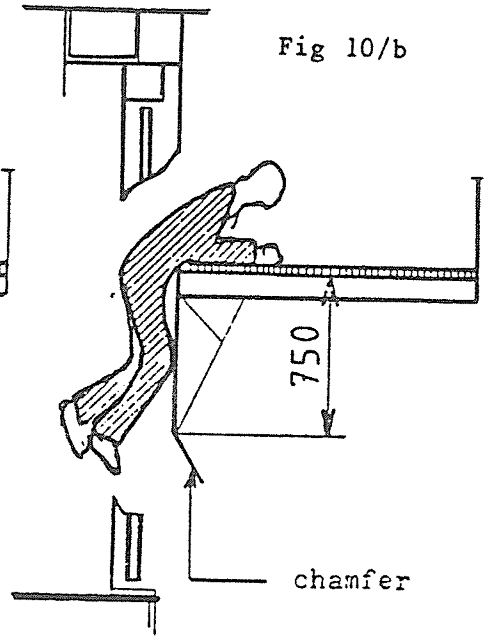


Fig 10/b



The apron shall of course be strong enough to resist the force applied in this case. In the absence of precise requirements, it is suggested to use the same as in HB/8.3.2.1.

8.5 Car entrances

It is almost universally recognized that car doors are desirable for passenger lifts.

Statistics made in Germany, France, Switzerland and the Netherlands when doorless cars were still allowed in these countries were all pointing to a rate of 0.3 reported accidents per 1000 lifts and per year.

The statistics did not indicate either if the lifts were provided with a sill protective device or which type of device.

Although the figure does not seem high, one should consider:
a) that the accidents may be serious and sometimes fatal,
b) that small children are often involved,
c) that the cost of an effective sill protective device is not that much lower than the cost of a car door.

In any event, these statistics did prompt Germany and France to require car doors even on existing lifts. In fact, within the CEN countries, only Finland and Belgium favoured the doorless cars for appartments.

8.5.2 Car entrance for industrial lifts

The Anglo Saxon countries do not want doorless cars even in industrial buildings. However, based on my experience, I believe it is a good solution for industrial applications because sturdy car doors are almost impossible to make and sturdy automatic landing doors are expensive whereas turdy swing doors are relatively easy to manufacture.

Of course all the conditions listed in EN/8.5.2 shall be fulfilled and the device required in EN/8.8 supplied (see HB/8.8 for details).

8.6.7 Mechanical strength of car doors

All the considerations developed in HB/7.2.3 apply here.

8.7 Protection during operation of doors

All the considerations developed in HB/7.5.2.1.1.2 and HB/7.5.2.1.1.3 apply of course here. When the car door is linked to the landing door during operation (which is usually the case), the total of the masses has to be taken for calculating the kinetic energy.

8.8 Entrance protection for doorless cars

The Fig 11/a and 11/b illustrate the possibilities of accident with doorless cars. The 11/a was the one most frequent when this type of entrance was tolerated in appartment lifts but, in industrial locations where the users are adults specially instructed, the 11/b would probably represent the possibility to be most feared.

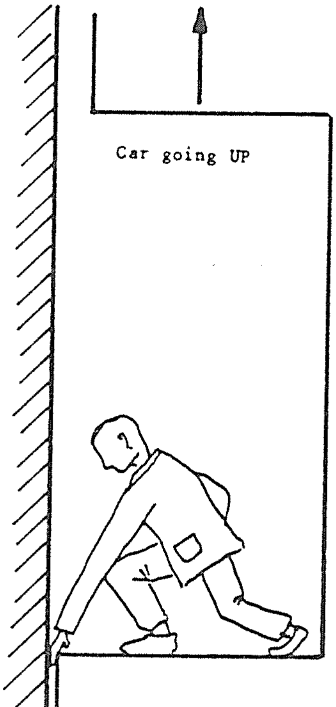


Fig 11/a

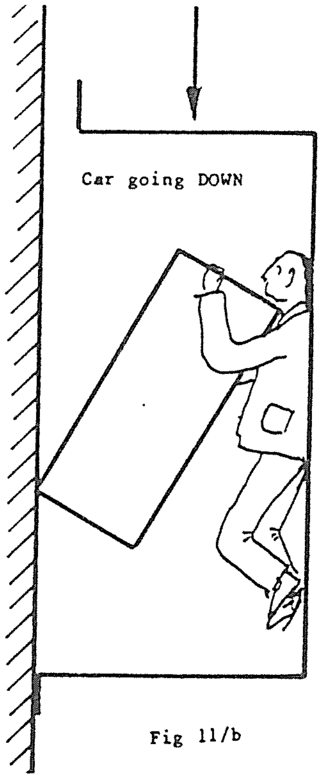


Fig 11/b

The code requires one photo cell without giving indications on the location. The fig 12/a,b and c suggest possible arrangments

The Fig 12/a would correspond to the minimum required by CEN.

In the Fig 12/b, the lower cell would give an improved protection for the danger illustrated in Fig 11/a whilst the upper one would cover more surely the case of the Fig 11/b.

In the Fig 12/c, the upper cell would suffice for meeting the code requirements and the lower cell is replaced by a retractible sill wich has proven effective for the case of Fig 11/a but is not adequate for rollers and small wheels and not so easy to manufacture with the long apron required in EN/8.4.2.

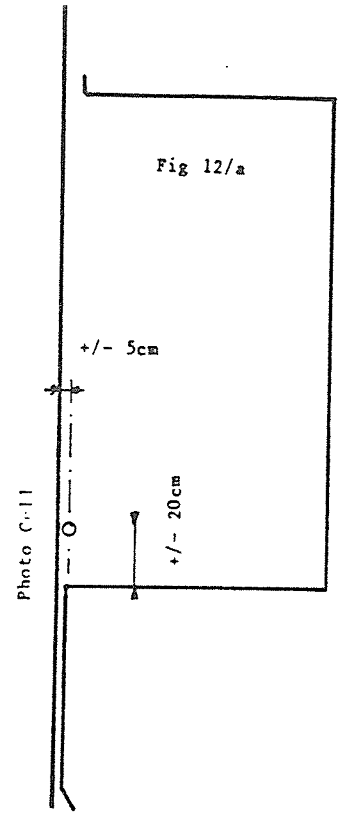


Fig 12/a

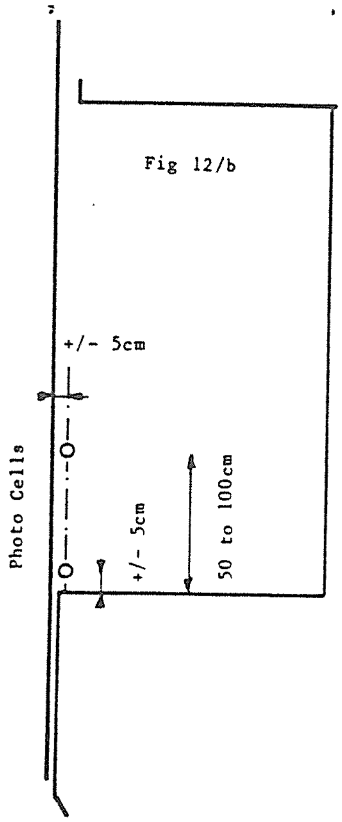


Fig 12/b

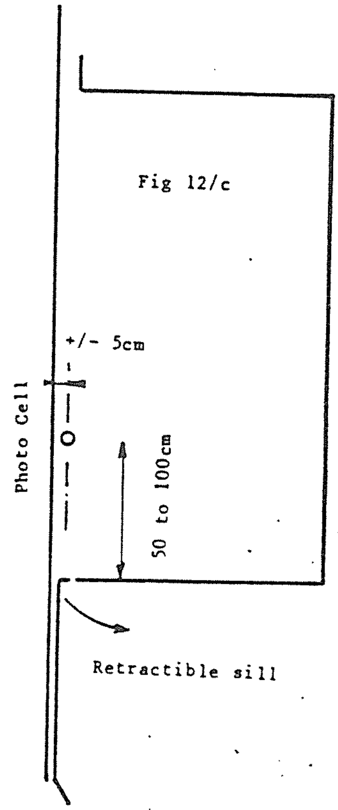


Fig 12/c

8.9 Locking of car doors

Although not mentionned in EN/8.9, the car door has to be locked in the case and conditions laid down in EN/5.4.3.2.2.

The code does not give any specifications for the car door lock but I see no reason why they should not be as reliable as the locks for landing doors which means that they should be tested under the same conditions. (see EN/F.1)

8.10 Multipanel doors

All the considerations developed in HB/7.7.6.2 are applicable here also.

8.17 Lighting, controls and signals

The code mentions only the requirements for the lighting but I think it proper to call attention to the controls and signals required in EN/14.2.

The ISO Standard 4190/5 "Control devices, signals and additional fittings" give useful additional details and its use is recommended.

8.13.3 Prohibition of the counterweight in case of drum drive

The reason for this is that the WG1 experts felt that, whenever possible, traction lifts should be preferred to drum drive lifts because of the extra safety resulting from:
- the possibility for the ropes to slip on the traction sheave if the car or the counterweight are stalled somewhere in the well,
- the greater number of suspension ropes.

The drum drive was not prohibited altoghether because it can be a solution in restricted quarters (ex: access for servicing industrial towers).
It was decided that, if there was room for a counterweight, a way of using the traction drive should be found.

9.1.1: Steel chains

As far as I know, chains are not used anymore for electromechanical lifts and one could wonder why they are still mentioned in EN 81 - Part 1.

Chains are apt to generate noise and vibrations. Moreover, although the counterweights are not prohibited here, all the reasons which led the CEN/TC 10/WG 1 to prefer traction drives to drum drives are equally valid for the chain drives (see HB/8.13.3).

9.1.2 Requirements for steel ropes

9.1.2/a
The relevant standard is the ISO/4344 "Steel Wire Ropes for Lifts". It refers to other applicable ISO standards.

It took the ISO more than 10 years to have this Standard agreed which indicates that reaching compromises was not easy.
I think it can be said that the Standard represents the greatest common denominator on which a sufficient number of national delegations could agree but, despite its limitations, it is a good set of basic specifications for lift ropes and an important step towards normalization of the market.

Any lift rope meeting the requirements of ISO/4344 is acceptable from the EN code point of view, however, as a consequence of the considerations developed in the following paragraphs, I suggest a minimum of 2 additional requirements to be imposed on the rope manufacturer:

a) The rope manufacturer shall guarantee the maximum loss of rope diameter after one year of normal operation and this maximum shall not exceed 5% of the nominal diameter value.
b) The diameter of the core shall be as constant as possible. In particular, there shall not be any hump because of splicing or any other reason.

The considerations in the following paragraphs will also provide guidance to make a choice amongst the many alternatives recognized by the ISO/4344.

9.1.2/b
The attention is called on the footnote N°3 (ISO/4344 page 1).
The dual tensile strength ropes might be removed from the ISO standard in a few years. It is because of the insistance of the lift industry representatives that, after long discussions, the wire manufacturers accepted to keep the 1370 N/mm2 wire as a standard and it will probably be more and more difficult to procure in the coming years.

All the tests seem to indicate that the high tensile strength wires behave better. All other industries have adopted them and, as regards the new lifts, most of the lift manufacturers have also adopted them now that better and harder cast iron qualities are easily available for the sheaves.

The reason for the lift manufacturers insisting on keeping the 1370 N/mm2 available for the outside layers was rope replacement on the several hundred thousands of existing lifts which had soft cast iron sheaves.
But, as explained later, installing new ropes in worn grooves is not recommended and existing sheaves can not be regrooved many times.

It is then advisable:

a) to use high tensile strength wire ropes in all new designs,
b) to use hard cast iron when replacing existing sheaves and switch to the new kind of ropes whenever the opportunity arises.

9.1.2/c
As far as I know, the most commonly used hoisting ropes for passenger lifts are of the Seale 8*19 construction.

The Seale 6*19 ropes are mostly reserved for the small diameters used with governors and service lifts.

Other choices are sometimes made, on grounds of the experience accumulated by some manufacturers with a given groove design or for given applications.

9.1.2/d
The bright wires are satisfactory for most applications.

It is only if the lift ropes are exposed to rain or chemically polluted atmospheres that a zinc coating might be desirable. Other coatings, if not unthinkable, are to be considered as very special.

9.1.2/e
The main cores made of natural fibers have been time tested and are still practically the only ones used today in North America.
Providing they are made carefully avoiding humps (refer to HB/9.1.2/a) they have proven to be satisfactory for many years as support for the strands and as a lubricant reservoir.

Humps are to be carefully avoided with any kind of core because they would be the cause of rapid local wear of the rope leading to its early replacement. Since humps are more likely to happen with natural fibers, extra care should be exercised during manufacturing.

Man-made fibers have been and are still used in Europe.
They have the advantages of more constant characteristics and almost indefinite length but experience has sometimes deceived: the core material not retaining its physical or chemical properties.
Unless you have personal experience or full confidence in the experience claimed by the rope manufacturer, it is safer to use natural fiber cores for the time being.

Although the ISO/4344 standard, in the introduction of parag 4.2, specifies that cores shall normally be made of fibers, steel cores cannot be forbidden because of parag 4.2.3.
Steel cores have the same advantages as man-made fibers but they are not good lubricant reservoirs and in some cases, the rope did deteriorate from the interior. As said for the man-made fibers, they should be used only if you have personal experience with a given application or if you can trust your rope supplier's recommendations.

By the same token, man made-fibers made of material other than the ones defined in 4.2.2.(a) are acceptable if agreed between manufacturer and purchaser because of 4.2.2.(b). They should be used with the same precautions as above.

9.1.2/f
The rope lubrication is important. The paragraphs 4.3 and 5.4 of ISO/4344 give what is required for new ropes but neither the ISO standard nor the EN code give any indications for the lubrication during operation.

It is always good to keep the ropes lubricated. It is highly recommended if the conditions of use are severe. The lubricant shall be selected according to the rope manufacturer recommendations and the best way of applying it is to use an automatic device acting permanently.
On the other hand, an excess of lubricant should be avoided because the traction conditions could be affected and lubricant would be spread all over the place.

9.1.2/g
The tolerances on the diamenter of the rope have been one of the hottest subject of discussion in the ISO Committee.

Some delegations wanted closer tolerances on the new ropes but, surprisingly, nobody ever raised the question of the shrinking of the rope during its life time.

In fact, for designing the grooves, the 2 important limits are:

a) the maximum diameter of the rope when new, to avoid squeezing,
b) the minimum diameter after the initial stretching, to be able to design the groove for proper seating of the rope during all its remaining lifetime.

Everybody knows that regular lift ropes stretch during the initial period of use and an old American rule of thumb indicates that a stretch in the order of magnitude of 0.75% of the length should be expected.

This stretch cannot go without a reduction of diameter which, according to some formulae, would be around 4% for the usual ropes.
Tests reported by Shitkow and Pospechow and made on a 25 mm rope loaded at 12.5% of its breaking strength indicate a reduction of 5% of the initial diameter after about 15% of the useful lifetime.

The 4 or 5% of diameter which will be lost during the initial period of use are more important that the 1 or 2% improvement of the tolerance which was the object of the discussions.

This loss of diameter can be reduced (for example by pre-stretching), but in the absence of commitment from the rope manufacturer you can reckon with a probable 5% after 1 year of operation.
Later, the rope will keep losing diameter but at a very low rate.

9.2 Ratio D/d - Safety factor of ropes

The values proposed by EN 81 are related to each other as is evidenced by Fig 13 which compares these values with values deducted from diagrams of the document VDI 2358.

The tests on which these diagrams are based were not conducted in "lift conditions" and the publication is a bit old but the general indications on the factors influencing the life time and the orders of magnitude are certainly still valid.

When the life time starts to be expressed in multiple of 10,000,000 bendings, it tends to be so long that the rope probably did not break during the laboratory test (see the VDI 2358 diagrams).

The useful lifetime in actual lift conditions is of course shorter because the rope is discarded long before there is any danger of breaking, but Fig 13 gives interesting indications.

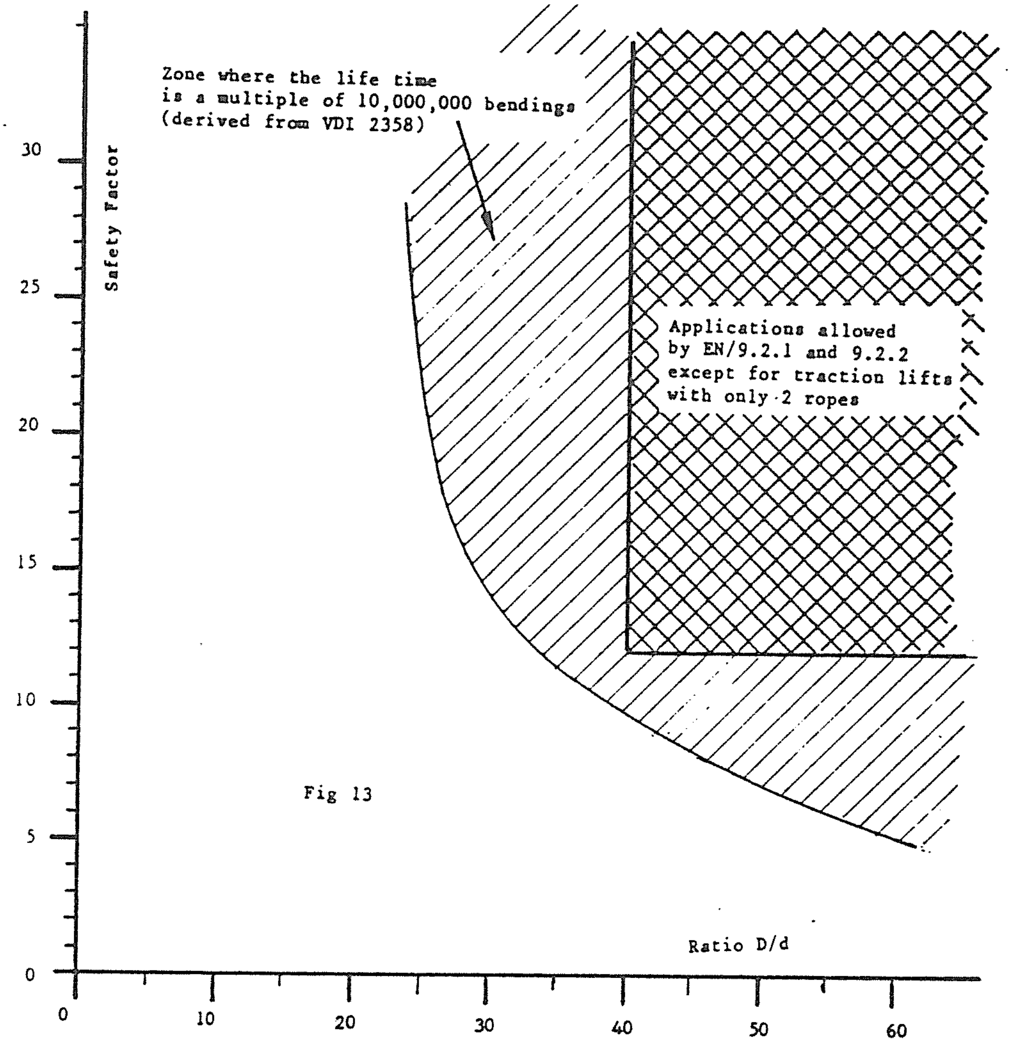


Fig 13

9.3 <u>Rope traction. Specific pressure</u>

9.3.1(a)
Although not specified, the code refers to the <u>empty</u> car in this case.

To play safe, it is recommended also to check the condition by calculation as proposed in HB/Clause 9.Note 1.
For the reason explained in HB/9.6.2, the mass corresponding to the rated load shall be, for the calculation, added to the mass of the compensator in the case of a (even only occasionally) tightly locked-down compensator.

9.3.1(b)
The code proposes simplified formulae.

You will find under HB/Clause 9.Note 1:

- a more accurate approach,
- how to make (nearly) constant traction grooves, whether V or round, by selecting the proper undercut.

9.3.2
The specific pressures indicated by the code are based on the experience with the usual combinations of groove hardness and tensile strength of the external wires of the rope.

If the groove is harder, lower pressures will be needed for avoiding early rope wear. If the groove is smoother, lower pressures will be needed to avoid early groove wear (see HB/Clause 9.Note 2)

9.5 Distribution of the load between the ropes

9.5.1
Every lift engineer has been dreaming of the perfect load balancing system but I believe that, besides the swing bar commonly used in the case of 2 ropes systems, the compression springs are almost generally used to comply with this requirement.

The balancing which can be expected from this system:
- is automatic only if properly installed,
- is only approximate once the ropes start stretching individually,
- is working only within a very narrow range.

So, the following precautions are imperatives:
a) make sure that, when installing new ropes the load is initially equally distributed between the ropes.
The equal compression of the springs should already give a very good indication but, in critical cases, you might want to check it by additional means such as: applying a given force perpendicularly on a given length of rope or measuring the time a wave takes to travel to the top of the hatch and back to the bottom.

b) make sure that the springs are correctly selected and properly installed. They should be long and strong enough so as to be only partially compressed when taking their share of the load.
The possibility of extension or further compression should ideally be proportional to the length of the ropes but, all other things being equal, can be inversely proportional to the diamenter of the driving sheave.

The code requires a device at least at one end of the ropes but it is better to have them at both ends if the ropes are very long.

c) inspect the device at regular intervals to check that it is still in good operating condition.

The reason for (a) is that, if one rope takes more than its share of the load, its lifetime will be shorter than the lifetime of the others and, if the safety factor for this rope is reduced enough, it might even fall in the zone of critical applications (see Fig 13)

This extreme situation can be encountered only if there is no equalization or if the design is so bad that the springs can detect only the very slack ropes. For example, I have seen a design where the tightening of the rope length adjustment came against a solid stop before full compression of the spring which could then give no indication on the rope load.

The reason for (b) is that the springs should be able to compensate, at least partially, small differences in the pitch diameter of the grooves. If one groove has a pitch diameter Delta mm smaller than normal, the rope is losing 3.14*(Delta) mm of travel at each turn of the sheave (irrespective of the sheave diameter). The longer the trave' the more turns for the sheave but a large sheave will need less turns.

This particular rope will tend to take more than its share of the load when lowering the car, unloading the counterweight side at the same time. Even if there were no spring, the limit would be given by the maximum traction which can be obtained from this particular groove arrangment and the rope would have excessive creep and eventually slip forward causing wear either in the groove or on the outside layer of wires.

In my opinion, the springs should be selected to be fully compressed with a load 30% over the normal rope tension and the rope should be considered as slack when loaded only to 70% of its normal share. The compression of the spring between these 2 extremes should of course be as long as possible (several centimeters).

The reason for (c) is that the device can only compensate automatically minor deviations and that adjustments will be necessary.
From that point of view, the use in all cases of a slack rope contact would make up for a possible lack of control.
In the case of very important lifts (long travels, heavy traffic), it seems to me that the contact should be operated not only when the load falls below a selected minimum but also if it exceeds a selected maximum. In a multiple ropes suspension, one rope could be overloaded without underloading the others to the extent of triggering the slack rope contact.

If you want to evaluate the behaviour of the springs, it can be programmed on a computer but, in this specific case, you must take into account the elasticity of the ropes because the relative impact is not negligible.
For the usual 8*19 rope with natural fiber core, the elasticity coefficient will be in the order of magnitude of E= +/-7800 Kg/mm2.

<u>Note</u>
The benefits which may be expected from the above precautions are dramatically illustrated by the experience of OTIS with the elevators of the CN Tower in Toronto. These 1800 Kg elevators, travelling at 6 m/s over 370 meters, were originally installed without equalization devices at all. The distribution of the load between the ropes was carefully adjusted every month by measuring the travelling time of a vibration wave along each rope, from the pit to the machine and back. An automatic rope lubricator was installed in the machine room.
Nevertheless, the ropes did not last more than 2 months in half round grooves!! Something had to be done.
The machining of the grooves was checked to insure equal pitch diameters.
The rope manufacturer was required to specially check the tolerances.
Equalization springs, 360 mm long, were fitted at both ends of the ropes and the initial tension was of course carefully adjusted.
Quite naturally, the ropes were frequently inspected when the elevators were restored to normal service and, pretty soon, early signs of local wear did show on a given spot of one of the ropes.
The OTIS technicians correctly diagnosed a bulge on the core and the rope manufacturer was able to correct the defect on the site by opening the strands and removing the excess material. Four years later, the ropes were still in good operating condition.

9.6 <u>Compensating ropes and compensators</u>

The general mathematical approach for analyzing the behaviour of the lift system during operation of either the safety gear or the buffer(s) with or without compensating ropes and compensator (locked or not) is explained in Appendix App:03/HB. It is necessary to master the use of these formulae to anticipate the behaviour of the lift and calculate the forces developed.

<u>9.6.1</u>
The code allows the use of free-hanging ropes or even free-hanging chains up to and including 2.5 m/sec rated speed.

Calculations made according to App:03/HB show that, at that rated speed, counterweight jumps up to 0.3 meter can be expected if the empty car hits the buffer(s) at 115% rated speed or is stopped by a progressive safety gear using cast iron brake shoes.

This is why I recommend the use of compensating ropes with tensioning pulleys as soon as the rated speed exceeds 2 m/sec which means that, in current practice, the use of free-hanging compensating ropes or chains would be limited to the geared lifts. Even so, their use could be questioned for the very high rises.
It should be noted that the device is justified, not only by the "compensation" which is needed or not for the traction, but also by the improvement in the operation of the safety gear and buffer(s) and the limitation of the counterweight jump.

9.6.1.(c)
Since the compensating ropes will generally have the same weight as the hoisting ropes (they are often exactly the same ones), their braking load will be comparable whereas, in normal operation, they support only the compensator. This means that, in normal operation, the safety factor is very high and, as shown by Fig 13, a ratio of only 30 is justified.

9.6.2/a
The code does not require an anti-rebound device for rated speeds up to and including 3.5 m/sec but, for the reasons already explained in HB/9.6.1, I suggest that the device be always associated with the tensioning pulley.

9.6.2/b
The anti-rebound device can be such that the compensator is permanently or occasionaly "tightly" locked-down. By that, I mean that it cannot rise at all after having been pulled down.

For example, the ratcheted devices are occasionaly tightly locking down the pulley when, the car being fully loaded at the lowest floor, the hoisting ropes are stretching just to the point of reaching the next tooth of the ratchet. The pulley will not go up if the car is unloaded.

The stretching has then to be divided between the hoisting ropes and the compensating ropes and this is as if a mass equal to the car load had been transferred to the compensator.
This must be taken into account in the traction calculations for the case of the empty car at the top floor and, logically, the phenomenon should be reproduced when testing the lift on the site.

9.6.2/c
-Calculation of the forces acting on the compensator-

9.6.2/c/1
I call first attention to the impact of the counterweight on the behaviour of the lift system as long as it is rigidly linked to the car.

Let us call "acceleration", the increase in the downwards speed and "deceleration", the reduction in the downwards speed of the car.

Due to the effect of gravity, the counterweight, if left alone, has a "natural" deceleration of 1 Gn.
It tends to bring closer to this 1 Gn, the deceleration which the car would have in free fall whenever it is linked rigidly to the car.

The Fig 14 illustrates what can be expected in the conditions of the typical lift selected in HB/FWD/12.

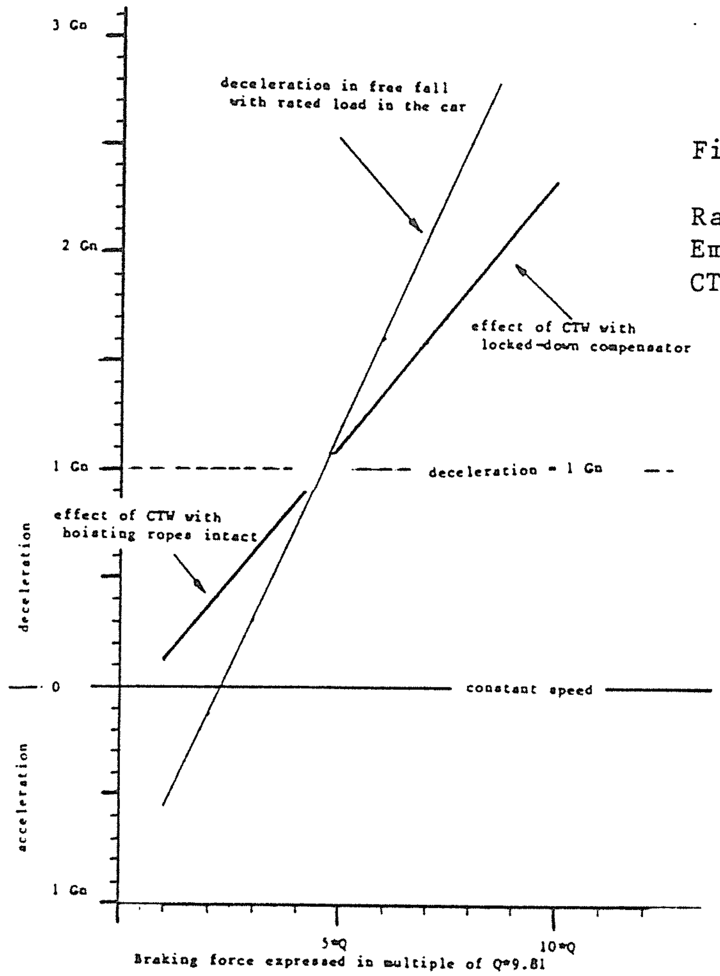


Fig 14

Rated load	Q
Empty car	Pl=1.3*Q
CT-weight	CT=1.8*Q

It should be noted that, besides avoiding the jump of the counterweight, the locking of the compensator has another beneficial effect: it reduces the peak deceleration of the car once it exceeds 1 Gn.

The Fig 14 shows also that, when the hoisting ropes are intact, the counterweight increases the decelerations when they are lower than 1 Gn.

At the limit the car would stop if the hoisting ropes are intact whereas it would keep accelerating in free fall: see Fig 14 for braking forces lower than 2*Q*9.81 Newtons. This illustrates what will happen with a safety gear adjusted for the longest sliding distances allowed by the American code.

9.6.2/c/2

To calculate the pull to be expected on the compensating ropes, the formulae developed in App:03/HB should be used.

First the maximum deceleration to be expected for the whole system should be calculated. It will of course correspond to the peak of the braking force developed either by the buffer(s) or by the safety gear when the car is empty (see below justification as regards the buffers).

Let us call this peak force : Fmax.
Using the simplified formula given in App:03/HB, this maximum deceleration (let us call it Gamma) will be:

$$\text{Gamma} = (\text{Fmax} - P*9.81 + CT*9.81)/(P+CT)$$

Then the maximum pull on the compensator ropes (let us call it Pull) can be calculated because the counterweight is of course experiencing the same deceleration and the only forces acting on it are the gravity forces and the pull of the compensator ropes.

$$Pull + CT*9.81 = Gamma*CT$$

which means:

$$Pull = CT*(Gamma - 9.81)$$

As regards this Fmax, 2 cases are to be considered: the action of the buffer(s) and the action of the safety gear.

Action of the buffer(s)
Because of the code requirements, the deceleration in the case of free fall may not exceed 2.5 Gn (disregarding the peaks lasting less than 0.04 sec which will be absorbed by the elasticity of ropes etc..).
With the hoisting ropes intact, this value will not even be attained with the load in the car because the counterweight will contribute to the deceleration of the system in the early part of the buffer action (see left part of the Fig 14). This means that the case to be considered is the empty car. The force Fmax will then be equal to P*3.5*9.81 Newtons.

Let calculate what happens if the ratios between the masses of the car and the rated load, between the counterweight and the rated load are the same as for the selected typical lift i.e:

$$P = 1.3*Q$$
$$CT = 1.8*Q$$

Calculations show that, for these values, the maximum rope pull will be about equal to 1.4*Q*9.81 Newtons

Action of the safety gear.
Of course, the calculations made above do apply as well for the instantaneous safety gears with buffered effect.

Locked-down compensators are, in practice, never used with instantaneous safeties. The code does not prohibit the combination but it should not be used because in that case the value of Fmax, and consequently the value of Gamma, would be extremely high which would mean a detrimental sudden pull on the compensating ropes.

If the material used for the braking shoes of a progressive safety gear had a friction factor remaining constant at any speed, the deceleration could be adjusted to 0.6 Gn, and would always stay equal to 0.6 Gn, in the case of free fall with rated load in the car (see EN/F.3.3.3.1).

The corresponding braking force would be equal to:

$$Fmax = 1.6*(P+Q)*9.81 \qquad \text{Newtons}$$

Using the same ratios for P and CT in relation to Q, this gives:

$$Fmax = 3.68*Q*9.81 \qquad \text{Newtons}$$

The deceleration of the lift system, in case of safety setting with empty car, would then be:

$$Gamma = (3.68*Q-1.3*Q*9.81+1.8*Q*9.81)/(1.3*Q+1.8*Q)$$

$$= 1.35*9.81 \text{ m/sec2}$$

Using the sames formulae as above, we will find, whatever the speed:

$$\text{Pull} = 1.8 * Q * (1.35-1) * 9.81$$
$$= 0.63 * Q * 9.81 \quad \text{Newtons}$$

However most of the progressive safety gears are made with cast iron brake shoes and, as exposed in HB/F.3.3, the friction factor depends on the rubbing speed. With increasing setting speeds, the force applying the shoe must increase to maintain the same mean retardation of 0.6 Gn and the retarding force when nearing the zero speed is increasing proportionally.

The Fig 15 shows the result of the calculations for speeds up to 10 m/sec.

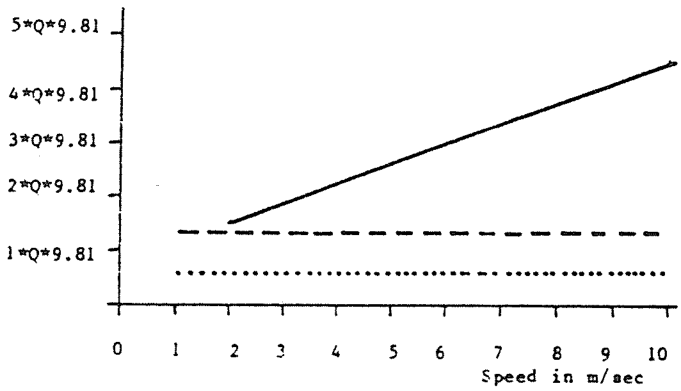


Fig 15 Pull on the compensating ropes expressed in multiples of the weight of the rated load (Q*9.81)

All the curves are for the case where:
P = 1.3*Q
CT = 1.8*Q
roping of compensation = 1/1

— — — — case of the hydraulic buffer(s)

.... case of the progressive safety with constant friction factor. Adjustment as per EN/F.3.3.3.1

———— case of the progressive safety using cast iron shoes. Adjustment as per EN/F.3.3.3.1

9.6.2/c/3

The ropes used for compensation are often identical to the ones used for hoisting or, at least they have the same weight which means that their breaking load is of the same order of magnitude.

Using again the same ratios, the Breaking Load of the hoisting ropes will at least be equal to:

$$MBL >= 12*(1.3*Q+Q) \text{ or } 27.6*Q$$

So the factor of safety of the compensating ropes is not a problem.

To the contrary, the forces calculated above must be taken into consideration for the design of the ropes terminations and the fixing to the bottom of the car and to the counterweight.

The force applied to the compensator itself is, of course, twice the pull in the ropes and this should govern the design of the compensator and of its attachment to the pit floor.

9.6.2/c/4

All the above reasoning assumes that the roping of the compensation is 1 to 1, which is generally the case.

If it were not the case, the forces would have to be divided accordingly.

9.6.3
-Compensation of the weight of the travelling cable(s)-

There is no 9.6.3 in the EN code but the problem related to the travelling cable is, at least in the principle, the same whatever the type of compensation: free hanging or with tensioning pulley.

It should be noted however that this problem is worth considering only for the long travels which are generally requiring high speeds and, as a consequence, compensating ropes with tensioning pulleys.

To compensate the transfer of the weight of the travelling cable from the wall of the hoistway to the car when the latter is going up:

- the weight of the counterweight must be increased by half the weight of the travelling cable(s),
- the weight of the compensating ropes (or chains) must be decreased by half the weight of the travelling cable(s).

You can easily check that the compensation is exact by analyzing the balancing of the weights with the car at the bottom, mid-travel and top positions.

Doing this will improve the traction conditions, might require less torque from the motor and might reduce the cost because a kilo of cast iron is less expensive than a kilo of steel rope.

It is significant only for long travels but it becomes important for extremely long travels because the travelling cables are becomming very heavy (even per meter of length).

It is amusing to note that, by increasing artificially the weight of the travelling cables (for example by adding steel ropes) to the extend of reaching twice the weight of the compensating ropes going from car to counterweight, the latter can be deleted altogether.
Reportedly, this trick has been used in small lifts but I fail to see the advantage of it.

9.8.2.1 Conditions of use for different types of safety gear

In EN/9.8.2.1(a), the use of the instantaneous type with buffered effect is limited to rated speeds not exceeding 1 m/s.

It is unfortunate because this type of safety is the only one which is able to decelerate smoothly a lightly loaded car whilst being able to stop a fully loaded car in the conditions required by the code.

This characteristic should be of particular interest for the bed lifts. Indeed these lifts have rated loads ranging from 1600 to 2500 kg but, when carrying a hospital bed with a patient under the supervision of an attendant, the actual load will be in the order of magnitude of 300 kg.
If the poor patient is in bad shape after an accident or surgery, a smooth operation of the safety gear is of paramount importance.
You can find Appendix APP:04 the results which may be expected and some guide lines for the design.

It is interesting to note that, in North America, this type of safety gear is allowed for rated speeds up to 2.5 m/s.

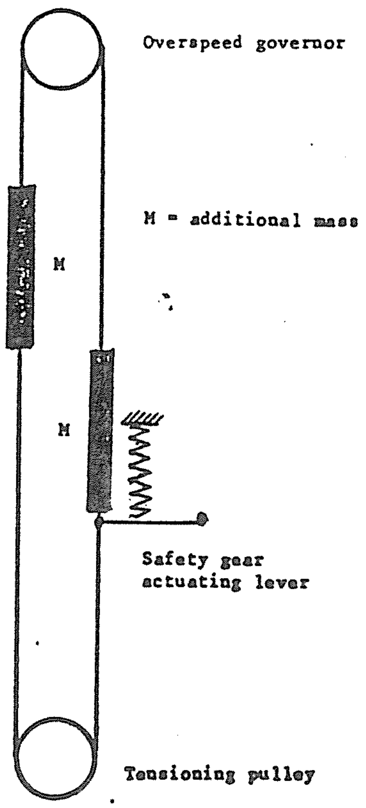
9.8.3 Methods of control

9.8.3.1

Tripping by an overspeed governor is all right for medium and high speeds. For low and very low speeds, it becomes questionable and EN/9.9.2.2 calls for a special design without giving indications.

My suggestion is to adjust the mass of the linkage between the car and the governor so as to trip the gear by inertia in case of free fall.
Tripping by inertia is required by ANSI for all instantaneous safety gears.

Fig 16



To achieve this you might have to add 2 equal masses on the leading and on the return ropes of the overspeed governor.

Let us call "Mtot" the total of the 2 additional masses, plus the mass of the governor rope, plus 50% of the mass of the governor and tensioning pulleys.

In normal operation, the system must be accelerated at 1 m/s (probable acceleration of the car) without tripping the safety gear. So, the actuating lever must be kept down by a spring exerting a force of Mtot*1 Newtons.

In case of free fall of the car, the system will pull the lever with a force of

$$Mtot*(9.81-1) \text{ Newtons}$$

By adjusting the additional masses, you can make sure that this value is large enough to operate the safety gear.
Make also sure that the design is such that the downward movement of the lever cannot disengage the rollers because, after the setting of the safety, the same masses will violently pull the lever down.

9.8.4 Retardation

9.8.4/a

First, let us remark that it seems reasonable to assume that "average retardation" means "mean deceleration based on time" which can be quite different from the "mean deceleration based on distance".

The "mean deceleration based on time" is the constant deceleration which would stop the car in the same time as the actual phenemenon. Its value is calculated by the formula: (Initial speed)/Time.

The "mean deceleration based on distance" is the constant deceleration which would stop the car on the same distance as the actual phenomenon. Its value is calculated by the formula: 0.5*(Square of speed)/Distance.

It is only when the actual deceleration is nearly constant, that the 2 mean decelerations are close to each other.

9.8.4/b

You will note that all that the clause requires is that the average retardation lies between 0.2*Gn and 1.0*Gn in the case of free fall with rated load in the car.

The type tests, required later (EN/16.1.2.2), are more demanding.

First, the allowed applications would lead to average decelerations ranging from 0.49*Gn to 0.73*Gn, see EN/F.3.4(a)(2). This is justified because, in the spirit of EN/General Introduction, the requirements of the clause shall be met on the installation notwithstanding the usual tolerances on the thickness of the guides, the surface conditions, etc.

In the second place, the testing procedure requires that~~, not only~~ the mean decelerations~~, but even the instantaneous values of the deceleration~~ do not differ by more than 25% in relation to an average based on 4 tests.

The Clause 9.8.4 itself sets no limits for the instantaneous values of the deceleration. This discrepancy, which can create problems for rated speeds exceeding 2.5 m/s when cast iron shoes are used, will be further discussed in this Handbook under the reference HB/F.3.3

Of course, with an empty car, the mean deceleration will be in the order of magnitude of 2.2*Gn and peak instantaneous values will be very high. This is why the instantaneous safety gears with buffered effect have an advantage: even at light loads, the mean decelerations can be kept below a fraction of 1*Gn with peak instantaneous decelerations exceeding 2.5*Gn limited to a duration of 0.04 s maximum.

9.8.6 Constructional conditions

9.8.6.2

The code does not give specifications for the instantaneous safety gear to be used in conjunctio- with the buffer(s).

The buffer being selected as per EN/10.4.2 and EN/10.4.3, is capable, by itself to absorb all the energy to be dissipated. All that is expected from the safety blocks is to provide a base for the buffer. This base must be able to resist a force of 40*(falling mass) (in Newtons) as explained in HB/Clause 5.Note 2.sub b.2.

An analysis of actual blocks characteristics shows that, in the present

application, they can safely be used for twice the suspended load which is allowed by the testing laboratory for use without buffers.
However, it is advisable to get an approval from an authorized organization before applying the above recommendation.

9.8.6.4
There is no 9.8.6.4 in the EN Code but I want to call the attention to the necessity of designing the safety gear operating mechanism so that it can resist, with a suitable safety factor, the maximum force which can be exerted by the overspeed governor.

Here appears the necessity to relate the choice of the overspeed governor to the choice of the safety gear.

9.9 Overspeed governor

9.9.2.2

See the comment on HB/9.8.3.1

9.9.4/a

The code gives the minimum required but there is a maximum not to be forgotten: the force exerted by the governor should not exceed the force which the safety gear operating mechanism can withstand divided by a suitable safety factor (see also HB/9.8.6.4).

9.9.4/b

The same minimum and maximum should apply to the pulling force developed by the friction device of a safety rope (if any).

9.9.6 Overspeed governor ropes

All the requirements should apply equally to the safety rope (if any) with the exception of EN/9.9.6.7.

9.9.6.2

The force to be taken into consideration is the actual force exerted by the overspeed governor or by the friction device of the safety rope.

9.9.7 Response time

9.9.7/a

What is called the "response time" should be related to the assumption made in the type testing procedure for instantaneous safety gears.
In EN/F.3.2.4.1, it is said that 0.10 m corresponds to the distance of travel during the response time.

The fig 17 (see next page) shows the general arrangment of a type of overspeed governor which has been, and probably still is, on the market. You can see that, if the rocker just barely missed the tripping speed when passing one of the summits of the cam, the rope will have to travel at least 157mm and possibly (157 + 52) before having another chance of tripping. With such a governor, there is 1 chance out of 2 or 3 to exceed the 0.10 m used for the calculation of the allowable suspended mass for the instantaneous safety gears.

To the contrary, the governor sketched Fig 18, using 2 symetrical flying masses and 5 possible catching teeth, will not travel more than 67mm after reaching the tripping speed. If the mass (1) just missed the bottom tooth, the mass (2) will catch the tooth on radius B.

The geometry of any mechanism should always be such that, after reaching the tripping speed,the rope does not have to travel more than 0.10 m before the actual tripping. When there is an intermediate mechanism between the speed detecting device and the brake applied on the rope, the travel needed to actuate this mechanism shall of course be taken into consideration.

Although the same assumption is not made in the procedure for testing the progressive safety gears, I think it is good practice to limit also the response time to a travel of 0.10 m.

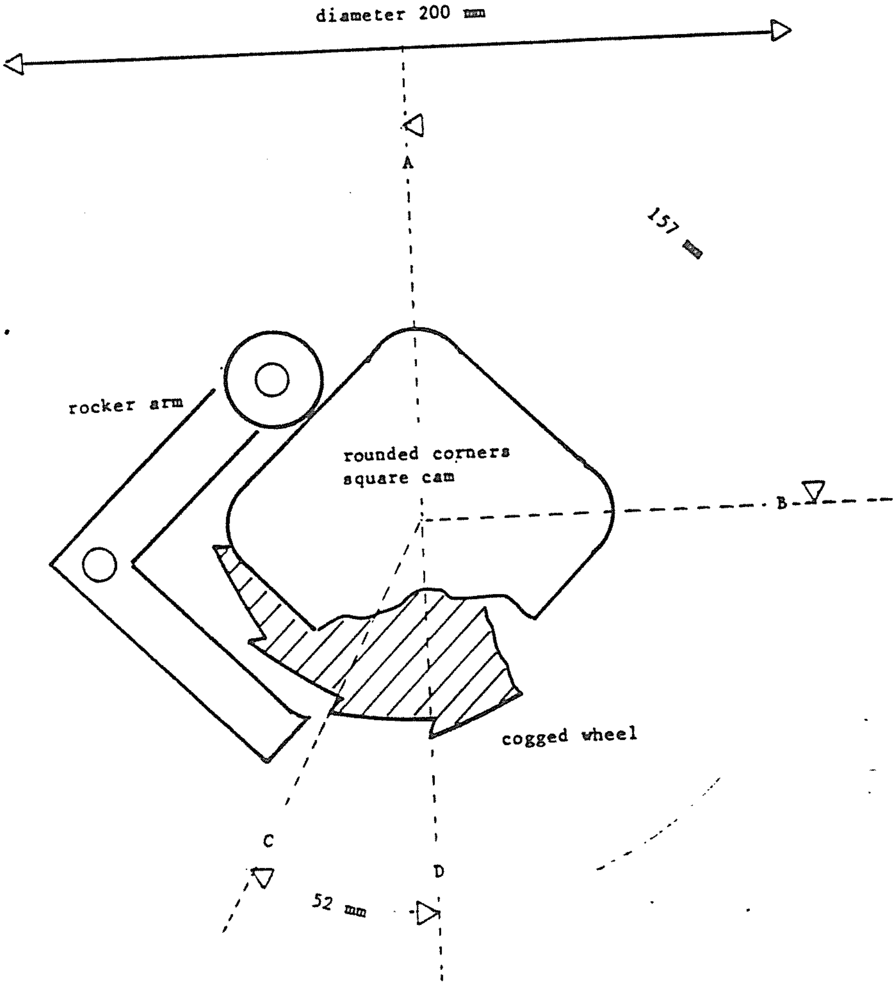


Fig 17 OVERSPEED GOVERNOR using the inertia rocker

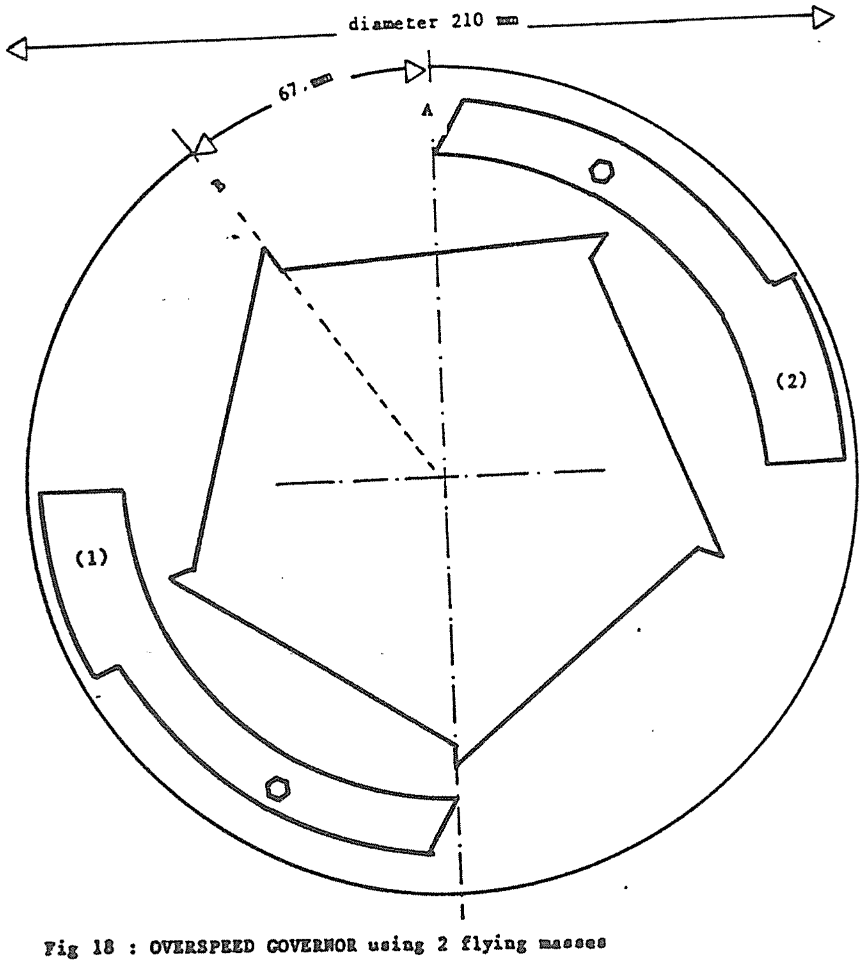


Fig 18 : OVERSPEED GOVERNOR using 2 flying masses

9.9.7/b

The same considerations should govern the design of the safety rope devices which have to allow some stretch of the hoisting ropes.
Not only the permanent stretch, but also the elastic stretch under load must be taken into account.

You can see Fig 19 the illustration of a possible arrangment.

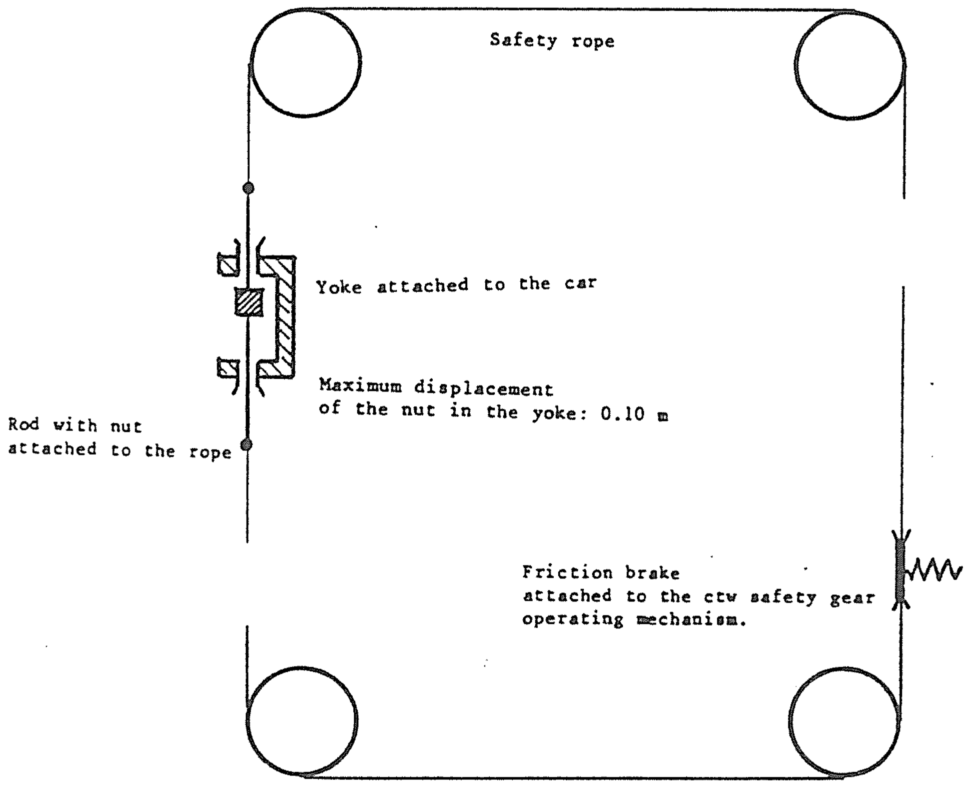


Fig 19 : Possible arrangment for a safety rope

9.9.11 Electrical checking

9.9.11.1/a

Fortunately, the EN Code (contrary to ANSI) states that the overspeed checking device need not be part of the overspeed governor.

In fact, for the time being, most of the overspeed switches are mounted on the governors, but, with the flying masses type of governor,:

- this can work only if the spring has a characteristic stiff enough to have a stable position of the masses for any rotational speed,

  Note:(see Fig 20 next page)
  Omega being the rotational speed in rad/s, we have:

  centrifugal force = (mass)*(square of omega)*(radius)
  spring force = (constant) + (spring characteristic)*(radius)

- the adjustment is delicate because the actuation of the switch must take place:
  - when the masses are above R1 which corresponds to the resting stops of the flying masses,
  - before the masses are close to the mechanical setting of the governor (practically before R2),

c) the qualification of the overspeed switch as "safety switch" is often a subject of controversy because the actuation should require only a very light touch and at the same time be positive and snappy,
d) only one overspeed can be controlled: the one corresponding to the rated speed.

The problems are the same (or worse) with the rocker type of governor.

Fig 20 : Flying masses governor using a stiff spring

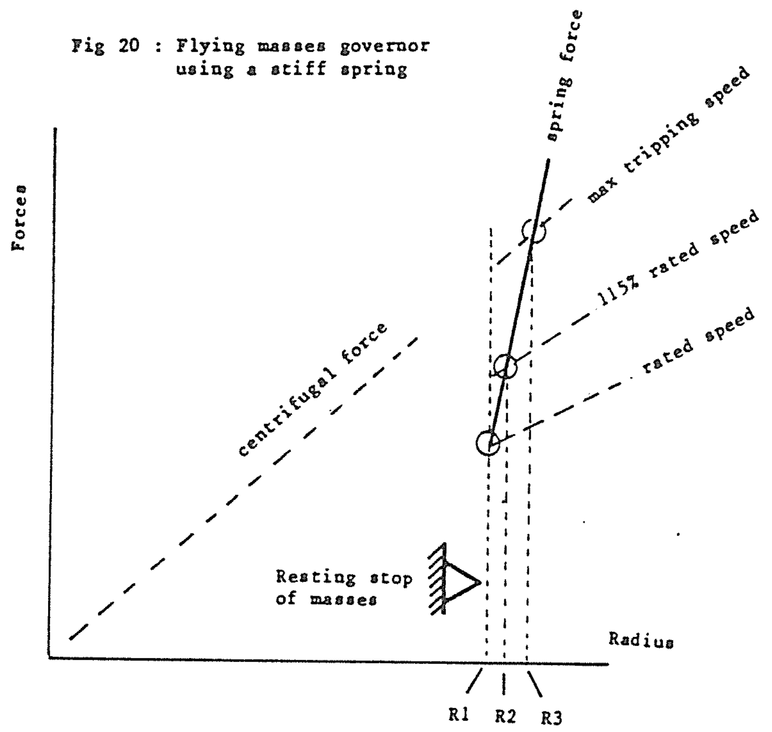
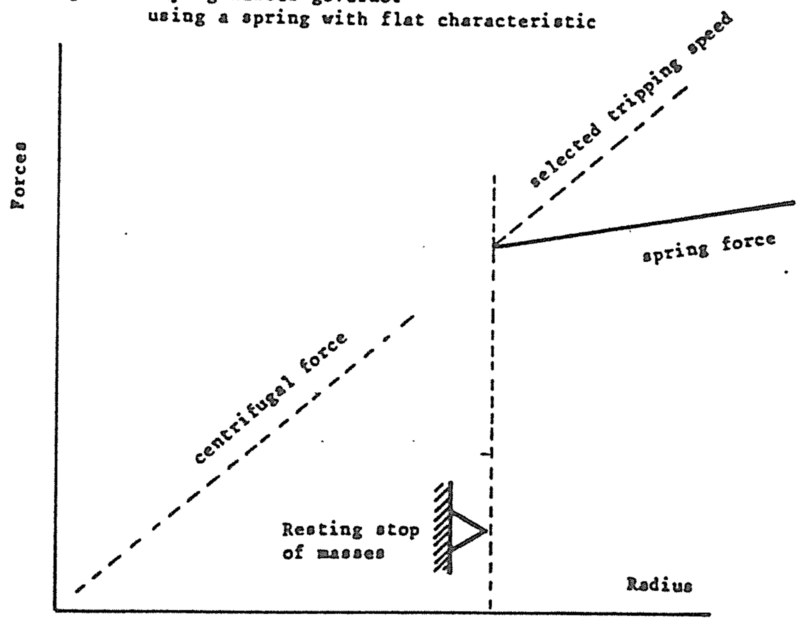


Fig 21 : Flying masses governor using a spring with flat characteristic



In my opinion, it would be better to control electrically the speed on the motor or on the hoisting machine.
Such an electrical control would have the following advantages:

- realizing a safety circuit is not a problem,
- the same device can be used to control, not only the overspeed related to the rated speed, but also the limits required for:
  - the levelling speed,
  - the inspection speed,

  which are both as important as the rated speed,
- the overspeed governor itself could be built differently to better achieve its original purpose.
  A flying masses type of governor using a spring with flat characteristic would snap open as soon as the triggering speed is reached because of its instability (see Fig 21). Better speed adjustments would be possible.

  A flat characteristic means a long spring or a special spring arrangment: the "constant" in the formula above should be positive.

<u>9.9.11.1/b</u>
In its sub parag (a) , the EN code says that, in some lifts, the overspeed switch need not operate before the safety switch installed on the safety gear, but it is still needed because it has to operate in both directions.

In fact, in all cases where the car speed is linked to the frequency of the mains, the overspeed switch is completely useless because:

- in hypersynchronism, the speed will never exceed 115% of the rated speed (normally not even 105%!)
- if the ropes are slipping or if the machine is broken, the brake cannot help,
- due to other requirements, there is no possibility of having the brake open when the machine is not energized.

Of course, you cannot take advantage of this remark for the rated speed until the code is eventually revised accordingly.
As regards the levelling and the control speeds, it is sufficient to ensure that the proper windings are energized to make sure that the motor runs at the selected fractional speed and that the prescribed limit is not exceeded.

<u>9.9.11.3</u>
These requirements should apply as well to the safety ropes.
Moreover, in this case, to avoid unwanted safety settings, it mmight be advisable to install a switch to control that, due to the stretching of the hoisting ropes, the nut of Fig 19 has not come to close to the upper arm of the yoke (same principle for other designs).

Clause 9. Note 1 <u>Traction calculation</u>

Note 1/a

We will first, using the additional symbols listed in HB/4.2, make explicit the formula proposed by the code, but, for taking the compensator in account (if any), we have to include the effect of acceleration in T1 and T2 and the formula reads now:

$$(T1/T2)*(C2) =< e\ldots\ldots(\text{see EN formula})$$

a.1: With the fully loaded car at the lowest landing,

$$n*T1 = (P1 + 1.25*Q + P5)*(g + a) + 0.5*P4*g$$

$$n*T2 = (CT + P3)*(g - a) + 0.5*P4*g$$

a.2: With the empty car at the highest landing,

$$n*T1 = (CT + P5)*(g + a) + 0.5*P4*g$$

$$n*T2 = (P1 + P2 + P3)*(g - a) + 0.5*P4*g$$

NOTES:

- In the case of reeving, part of the ropes go faster than the car whilst the others go slower. But, for passing from one side of the sheave to the other, the ropes take exactly the same time as the car takes for completing its travel.
So it make sense to apply to the mass of the ropes the same acceleration as to the car.

- The forces needed for the translation of the reeving pulleys attached to the car and to the counterweight are already taken into account because their masses are included in P1 and CT.
The forces needed for rotating the pulleys are generally disregarded. To be more accurate you would have to add one term to the expression of T1 and T2 given in HB/Note 1 to Clause 9-a.1.
Calling Mp1 the total mass of the pulleys (whether travelling or stationery) on the side 1 of the machine, Mp2 being the same for the side 2, you would have for the case (a.1)

$$n*T1= (P1+ 1.25*Q + P5)*(g + a) + 0.5*P4*g + 0.5*Mp1*a$$

$$n*T2= (CT + P3)*(g - a) + 0.5*P4*g - 0.5*Mp2*a$$

The 0.5*Mp1 and 0.5*Mp2 are an approximation of the equivalent masses of the pulleys if concentrated on the rim.

Of course, the same reasoning applys to the formulae of (a.2).

Note 1/b

Let us however remark that the formulae in the code are only approximations for the following reasons:

- the formulae do not take into account the throat opening (teta) always existing in the circular grooves,
- the formulae do not take into account the clearance always provided on the radius of circular grooves,
- the formulae do not take into account the thinning of the ropes due to the stretching under load (both permanent and elastic),

- the coefficient C2 is introduced at the wrong place in the formulae which leads to wrong conclusions when wide V's are used,
- the code does not set a limit related to C2 for the wear in tight V grooves, this can result in insufficient traction in old sheaves,
- the formulae do not cover the case of undercut V grooves (sometimes called Y grooves) which can provide a constant traction if the undercut is properly selected in relation to the opening of the V.

Because there is a hidden safety factor in the selection of 0.09 for the value of the friction factor μ, the approximation does not lead to dangerous situations. However, the real margin of safety is not known, very different unit pressure can develop, with subsequent different rates of wear, and, in extreme cases, rope slidings have been experienced during braking in normal operation.

<u>Note 1/c</u>
To have a better understanding of the traction problem and avoid, as much as possible, the problems of occasionnal rope slipping or unexpected rapid wear of the ropes or of the sheaves, please refer to:
- the appendix APP:05/HB which develops a more accurate approach to the traction problem and gives indications for the design of "constant-traction" grooves,
- the appendix APP:06/HB which, step by step, explains how to select the appropriate groove section and how to calculate its performances.

<u>Note 1/d</u>
Despite these reservations, the EN code formulas are better than the mere acceptance test required by the ANSI code because such a test does not give any indication of the rate of creep and of how close you are to failure. By luck, only circular grooves are used in North America and these grooves develop their minimum traction when they are new. It would be the contrary with tight V grooves (see APP:05/HB)

<u>Note 1/e</u>
Some additional specifications are recommended:

e.1
The groove should accomodate 5% oversized ropes without pinching.
This is justified by the tolerances allowed by the standard ISO/4344.

e.2
Even after regrooving, the depth of V grooves and of undercut grooves should be such that the rope can sink the equivalent of 0.75*d before touching the bottom of the groove.

If the rope came to rest on the bottom, the traction conditions would be completely different. The value of 0.75*d is consistant with the other recommendation which follows.

e.3
The sheave should be regrooved or replaced and new ropes installed if:
a) one of the ropes has sunk in the groove the equivalent of 0.5*d
b) the difference of sinking between 2 ropes exceeds 1mm.

(a) is in accordance with the recommendation (e.2), but this is even too much for plain V grooves. To be sure that enough traction is retained, the Appendices APP:05 and APP:06 should be applied.

As for the (b), this limit of 1mm is really a maximum which should be revised down in the case of very high rises as in the case described in the last sub parag of HB/9.5.1 (CN Tower in Toronto).

Note 1/f
The value of the friction factor, $\mu = 0.09$, is meant for cast iron grooves. If another material is used, you must get a derogation.

Even for cast iron, the validity of this 0.09 will be discussed in APP:5/HB before proposing new formulae (see APP:05/4).

Clause 9. Note 2 Specific pressure of the ropes in the grooves

Note 2/a
The ropes being made of wire, this wire can have ,at best, linear contacts with the flanks of the grooves and Hertz pressures develop.
They depend not only on the external forces applied and on the shape of the groove, but also on the wiring method of the rope and on the relative hardness of the wires and of the groove material.

Using these Hertz pressures is totally impractical and ,for comparison purposes, conventional ways are proposed for calculating the pressure.

Note 2/b
The formulae proposed by EN/81 are based on the following conventions:

b/1: For full-round grooves

- The rope is supposed to fill 180° in the groove.
- The radial force (radial sheave wise) applied to a unit area of the groove is supposed to be proportional to the volume of rope taken above this area along the sheave radius.

The value given by the EN/81 formula is the maximum value which can be found with the above assumptions (i.e. at the bottom of the groove)

b/2: For undercut semi-circular grooves

- "Beta" being the angle of undercut, the rope is supposed to fill (180°-beta) in the groove.
- The radial force (radial sheave wise) applied to a unit area of the groove is supposed to be proportional to the volume of rope taken above this area along the sheave radius.

The value given by the EN/81 formula is the maximum value which can be found with the above assumptions (i.e. at the edges of the undercut).

b/3: For V grooves

- the rope is supposed to rest on a flat corresponding to 25° of the rope circular section on each flank of the V.
- the pressure is supposed to be uniform on these flats.

Note 2/c

All these assumptions are wrong but probably as good as any other for comparison purposes and providing:
- the selected limits are in line with the same assumptions,
- the second formula is used only for V grooves between 30° and 40°.

Note 2/d

It should always be remembered that the conventions selected for V grooves are not fully consistant with the ones selected for semi-circular grooves and that the formulae do not give totally comparable results.

For practical purposes, they are commensurate enough to allow the use of the same limit value within the limits given above.

For V grooves with "gamma" angles ranging from 55° to 65°, the calculated pressure would be a little more favorable that the pressure calculated for the semi-circular grooves having the same traction capacity (undercuts between 55° and 65°) which is not logical.
This is not overimportant but shows the limit of application of the conventions and formulae.

Note 2/e Limit for the specific pressure

EN/81 relates the limit to the rope speed.
There seems to be no experiments showing that the rope speed has a direct influence on the behaviour of the rope and the sheave but, what is sure is that high rope speeds are always associated to long ropes and often high activity of the lift.
It means that rope replacements will be very costly and should be as infrequent as possible despite high activity.

We know, from various reports, that the wear of the ropes and of the grooves depends on the pressure of the ropes in the grooves; some say that it is proportional.
However, it seems that the formula proposed by EN/81 appears to give satisfactory results.

Another factor governing the wear is the amount of slip of the ropes in the grooves. By using the value 0.09 for μ as in EN/81 or by applying the factor 0.8 to the traction limit, as proposed in appendix APP:05, the slip will be limited to an acceptable value as demonstrated by Dr Molkow.

Finally, the wear also depends upon the relative hardness of the grooves material and of the external wires of the ropes.
With the 1770 N/mm2 wires, a hardness of about 230 HB seems to give good results. With the lower tensile strength a hardness below 200 seems better.

EN 81 : PART 1

HANDBOOK

A. LEENDERS

1986

(2) Pages 55 to 101

HANDBOOK and COMMENTS on the EN 81/Part I SAFETY CODE

by Andre Leenders

with some references to other leading Codes

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NOTE

The 16 APPENDICES to this HANDBOOK are in a separate volume.

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1986

10.1: General provisions

10.1.1: The forces to be taken in consideration are not all defined in the code. In the note at the end of the clause, only the vertical forces due to the operation of the safety gears are given by formulae.
I have already demonstrated, in HB/Clause 5.Note 1/a,b,c,d & e, that these values were purely conventional.

The code mentions that the forces due to uneven loading of the car shall be taken into consideration but does not propose assumptions as regards the distribution of the load.
On the other hand, the following forces are completely ignored:

- compression due to settling of the building and transmitted by the brackets. Even with sliding clips, this force can reach several tons in the middle of the travel (typically, a clip slides if the vertical force exceeds about 300kg. These forces being cumulative towards the middle of the travel, it takes only 20 brackets to reach 3 tons).
- bending forces due to the fact that the force applied by the safety shoes is not applied exactly to the neutral fibre of the rail.

This means that guides calculated strictly according to the EN code will, in all probability, be able to hold the car in the case of safety setting but that the quality of the ride is by no means guaranteed.

Various assumptions can be made for developing a more sophisticated mathematical approach but, whatever the set of assumptions, they will be valid only when associated with the limits resulting from the experience gained when applying the given method of calculation.

The problem is so complex that the method will be conventional in any event. Different time proven approaches have been developed in various countries, each country having specialized experience in different ranges of applications. I am not in a position to propose a better one.

The "ANSI/ASME A.17/1 rule 200", has been used successfully for high speed elevators but the rail sizes used in the USA being much bigger than the ISO ones, I would recommend using the British "BS 5655:Part 9" which has been revised recently and is, at least partly, based on the ISO sizes. (see HB/10.2 below)

10.1.2: The only justification for neglecting the forces due to the settling of the building, would be the use of sliding guide clamps of such a design that the guides could practically slide freely in the clamp.
In that case, provisions should be taken to absorb the reactions due to the setting of safety gears. This could consist of a kind of buffer at one end of the guide stack (either top or bottom). It would enable the exact calculation of the vertical reaction on the building.

10.2 Guiding of the car and counterweight

10.2.1: The number of guide rails is not limited, contrary to the ANSI Code. This might be of interest for heavy loads when very big guide rail sections are not available along with a suitable safety gear.

However, the number of guide rails should always be even, and the rails located on the same side of the car should not be too far apart to avoid developing high torques in case of uneven setting of the braking shoes.
The safety gears must necessarily be of the progressive type.

Guides rails made of materials other than steel are forbidden, contrary to some previous European codes which allowed wooden rails at very low speed for use in explosive atmospheres in view of avoiding the danger of sparks during safety settings.

10.2.2 and 10.2.3: These rules mean that:

Rolled steel guides may be used only under 2 simultaneous conditions:
- if the speed does not exceed 0.4 m/s
- if an instantaneous safety gear is used (if any).

Drawn steel rails may, according to the code, be used whatever the speed.

I believe, however, that nobody uses them for very high speeds.
The limit for the use of drawn steel is not obvious.
From my personal experience, it is questionable whenever roller guide shoes are used. The front roller seems to wear fast, probably because the nose of the rail is not perfectly flat.

This means that, in general, the use of large size drawn steel rails is restricted to big freight elevators (goods lifts).
This leads me to comment on the ISO 7465 Standard "Passenger lifts and service lifts - Guide rails for lifts and counterweights - T type".

When, in 1977, the FEM Section VII made a survey of the rail cross-sections used throughout Europe, it was found that more than 60 different rail sections were in use!
It was felt that a standard was necessary for the following reasons:
- the increasing internationalization of the lift business made it necessary to avoid shipping across Europe, guides to suit a given lift design. Suitable rails should be found locally.
- the increasing importance of components'manufacturing made it desirable to define for which guide rails safety gears and guide shoes should be designed in order to suit most lift makers.

The ISO TC/178 accepted the job of making a standard and after a few years produced the ISO 7465 Standard.
It proposes 13 sizes, which was already quite an improvement from the existing number of 60. However, one could wonder why there are still so many sizes with only minor technical differences.

In fact, the original draft proposal from the ad-hoc Working Group was proposing only half a dozen sizes rationaly spaced between the selected extremes and with only 3 different blade ticknesses 5, 9 and 16 mm.

In order to have the support of enough countries, additional sections had to be introduced because some manufacturers were fighting hard to keep their existing designs whatever the logic behind the proposal of the Work Group. This explains why blade thicknesses (noses) of 8 and 10mm sneaked into the standard along with an odd 125 mm size when there was already a 127mm*89mm with world wide acceptance. In my opinion this is a pity because the implementation of the ISO Standard is not mandatory and the standard itself states that ..."Guide rails with other dimensions can be delivered on specific agreement between the manufacturer and the customer".

I strongly believe that, for the design of future lifts and for the design of safety gears and guide shoes by components'manufacturers, a further selection should be made amongst the 13 sizes.

First of all (this is the most important item), the following blade thicknesses, and only these, should be taken into consideration:

- 5 mm for designing sliding guide shoes
- 9 mm for designing sliding guide shoes
  for designing roller guide shoes
  for designing safety gears
- 16 mm for designing sliding guide shoes
  for designing roller guide shoes
  for designing safety gears

(note: 15.88 mm is assimilated to 16 mm)

If, in the design, a manufacturer uses a different thickness he should be aware that finding suitable rails in the country of erection might be a problem. Inversely, customers and consulting engineers should be aware that, when accepting equipment using blade thicknesses other than 5, 9 or 16 mm, they restrict themselves to using components from the original manufacturer for replacement or modernization.

Then, for the design of lifts or rail brackets, my recommendation is that only the following ISO sizes should be retained:

50*50*5	(Wxx=3.15)	drawn steel	T 50/A	(ISO/BS)
75*55*9	(Wxx=6.58)	drawn steel	T 75-1/A	(ISO)
		machined	T 75-1/B	(ISO)
89*62*15.88	(Wxx=14.5)	machined	T 89/B	(ISO/BS)
127*88.9*15.88	(Wxx=31)	machined	T 127-2/B	(ISO/BS/ANSI)

Any other size would be considered as special.
However, if exceptionaly a bigger size is needed, I recommend using the next bigger size selected in the BS-5655:Part 9 which is the 139.7*107.9*19 also used in the USA. It has a Wxx of 52.9 cm3.

The Figure 22 (next page) illustrates the reason for these choices.
As usual, a good selection of sizes should be based on an exponential progression of the main characteristic of the guide rails.
In the case of the guide rails, there are 2 values of importance: the moment of inertia and the cross-sectional area modulus. I selected the cross-sectional area modulus Wxx as the leading characteristic.

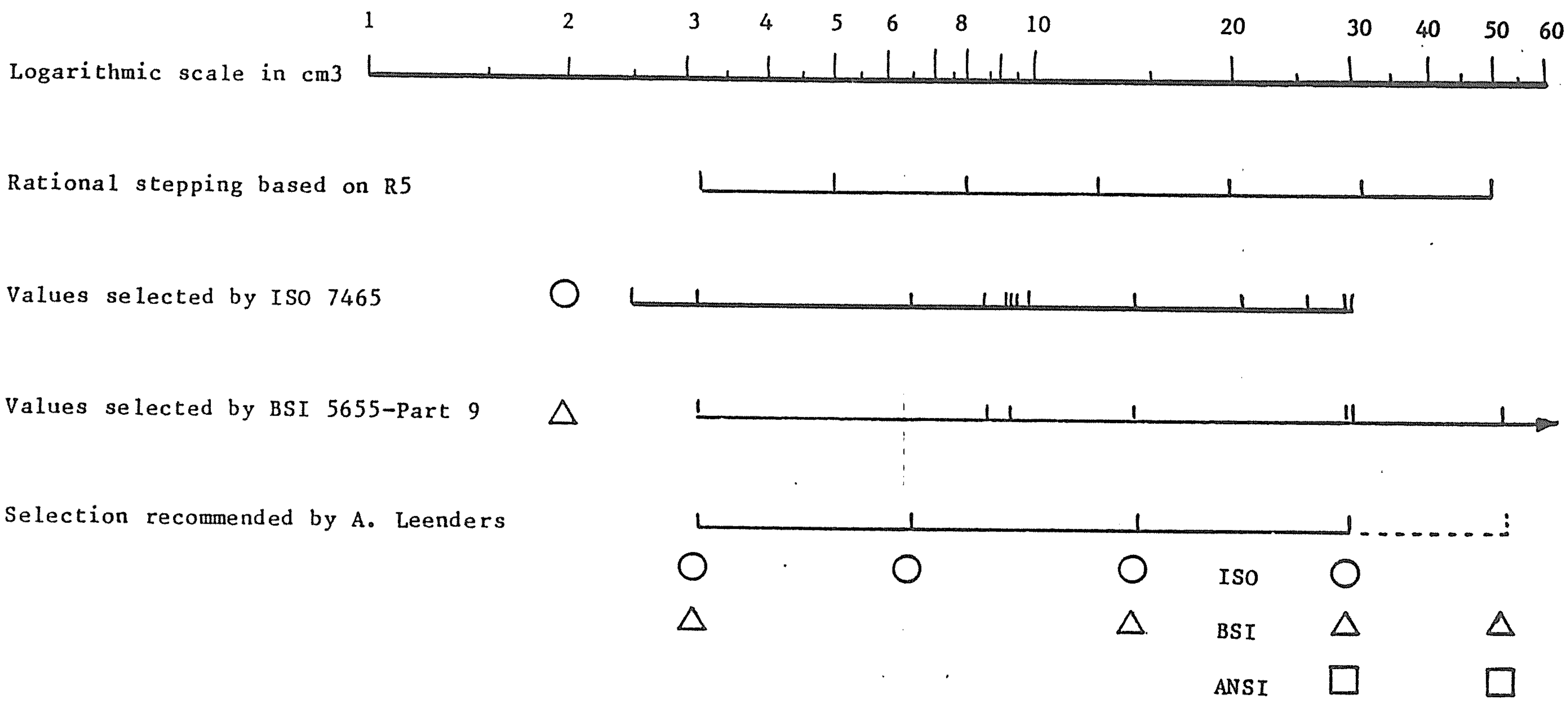
The experience has shown that the smallest practical value for Wxx is around 3 cm3 and I took the 3.15 cm3 of the T 50/A as starting value.
For other lift characteristics (loads and speeds), the progression based on the Renard Serie R5 has proven satisfactory and there is no reason to use another one for a rational stepping of the Wxx values.

When using a logarithmic scale, the points representing this rational selection would be evenly spaced as illustrated in Line 1 of Fig 22.

The Line 2 of Fig 22 shows clearly that the selection made by ISO is not rational. The numerous sizes having about 9 cm3 for Wxx cannot be justified by a rational approach.

The Line 3 shows that the BSI has already tried to cut out some of the unnecessary sizes and the Line 4 illustrates the reasonably even spacing of the sizes I recommend here above. The steps are a little bigger than the one selected as ideal in Line 1 but I am convinced that this is acceptable.

Fig N°22 : Distribution of Wxx values



10.3: General comments

The first thing to keep in mind is that the possibility of the car (or counterweight) striking the buffer(s) in free fall before tripping the safety gear is not taken into consideration.
(see Basic assumption N°3 in HB/Gen'1-0.7/a)

This is why the speed taken for the calculation is 115% of the rated speed. In practice, it is the setting of the overspeed contact (EN/9.9.11.1)

Nevertheless, the specifications for the spring buffers and the type testing procedure for the hydraulic buffers are based on free fall. This will practically be the case whenever the hoisting ropes are slack, but a special analysis is needed in the case of decelerations lower than 1 Gn or if there is a locked-down compensator. Whenever the ropes are taut, the counterweight has the effect of bringing the deceleration closer to 1 Gn (either decreasing or increasing the value in free fall).

See, in appendix APP:03/HB, how to calculate speeds and decelerations.

Now, in normal operation, the chances of hitting the buffer at full speed are very remote (see comments on EN/10.5 below); the only tangible probability is that the buffer will function only at low speed in case of a bad brake or severe overload.

In the course of erection or servicing work, the car (or counterweight) might hit the buffer(s) at full speed because of a wrong connection, but the code is not expected to protect people or equipment if the "competent person" makes a mistake.
This is in line with the assumption N°5 (HB/Gen'1 1-0.7/a).
(In the present case, it happens that the buffer would avoid damage although it has not been designed with such a mistake in view).

10.3.1: Case of the buffer(s) attached to the car (or counterweight)

10.3.1/a First let us remark that, when an instantaneous safety gear with buffered effect is used (EN/9.8.6.2), its buffer may be used as an end-of-ravel buffer and stops placed judiciously on the guide rails may replace the 0.5 m pedestal (see Interpretation N°111).

I assume that, using the same idea, the "solid pier" required in EN/5.5.2.a could be replaced by a beam anchored in 2 walls extending down to solid ground, but this would require another interpretation.

10.3.4: Energy accumulation type buffers with buffered return

I have never seen any but I assume, based on the requirements, that such a buffer is supposed to act only as a spring in the down direction and have something to brake down only the return speed.

In fact, most of the hydraulic buffers have a return spring and there is no limit set for the force they can exert when compressed and the part they will take in the deceleration of the car (or counterweight).

I think that, providing some of the energy is dissipated in the hydraulic system in the down direction, any combination "spring+hydraulic" buffer is to be considered as an "energy dissipation type buffer" and tested as such.

10.4.1: Energy accumulation type buffers

10.4.1.2/a: It should be noted that the values given by the code relate to springs with a linear characteristic which are the most commonly used.

When something else is used (ex: polyurethan), you may use the approach proposed by Dr Feyrer in the issue N°6 of the "Lift Report" dated November-December 1981 (page 16 and following) as indicated in the Interpretation N°112.

Coming back to the use of springs with linear characteristics, it is better to use the lowest allowable spring forces with strokes as long as possible to avoid, as much as possible, jumping of the counterweight.

10.4.1.2/b: If the spring has a stroke longer than neecessary, the force needed to fully compress it may exceed 4 times the weight of the mass defined in the code providing that, with this value of 4 times, the partial depression of the spring is at least equal to the required minimum stroke.

10.4.2: Energy accumulation type buffers with buffered return

See comment under the reference HB/10.3.4 above.

10.4.3: Energy dissipation type buffers

10.4.3/a: These buffers are, as far as I know, always hydraulic and generally combined with a spring. This spring can be there only for returning the buffer in the extended position but there is no reason why it could not participate in the slowing down of the mass.

In fact, some buffers advertised in the U.S.A. use the compression of a gas for building an increasing retarding force and returning the buffer in the extended position after operation.

10.4.3/b: Considerations on the average retardation (mean retardation)

Let us remark that the considerations developed previously under the reference HB/9.8.4/a for the safety gears do apply equally here.

10.4.3.1: If the stroke is exactly equal to the gravity distance, the mean deceleration based on distance is automatically equal to 1 Gn.

There is no minimum required for the stroke of a hydraulic buffer, but in good practice, strokes shorter than 0.07 m should not be used.

10.4.3.2: There is a strange indentation in the heavy line of EN/Fig 3. For all practical purposes, this indentation is meaningless because nominal speeds higher than 3.5 and lower than 4 m/s are never used. It would apply for 4 m/s, so to use the 0.42 m buffer stroke, you must have at least a nominal speed of 4.01 m/s!

You should note that monitoring the retardation of the lift to make sure that the speed will not exceed 50% or 33% of (rated speed)*1.15 when hitting the buffer, is delicate and requires, not only a precise measurement of the instantaneous lift speed but also a delicate balancing between the available traction and the control of the emergency braking.

Indeed, all that you can do is measuring the speed and, if higher that what it should be, initiate an emergency braking.
But this should happen at a distance of the buffer such that, given the emergency deceleration, the lift can slow down to the required reduced

speed when reaching the buffer.

Let us see what would happen in the following conditions:

- tripping of the emergency braking if the speed reaches 115% of what it should be at the point where the normal deceleration is initiated,
- deceleration in normal operation: 1.1 m/s2,
- time delay for initiating deceleration in normal operation: 0.1 s,
- deceleration in emergency braking: 1.5 m/s2,
- time delay for application of the brake in emergency: 0.25 s,
- last floor and buffer head practically level.

The Fig 23 shows, for rated speeds ranging from 2 m/s to 10 m/s:

- the distance needed for decelerating from rated speed to zero speed in normal operating conditions (X1 in m),
- the distance needed for decelerating from 115% of the rated speed to one half of this same value (X2 in m),
- the distance needed for decelerating from 115% of the rated speed to one third of this same value (X3 in m).

Figure Nr 23

For V= 2	X1= 2.02	X2= 1.90	X3= 2.14
For V= 3	X1= 4.39	X2= 3.84	X3= 4.39
For V= 4	X1= 7.67	X2= 6.44	X3= 7.42
For V= 5	X1=11.86	X2= 9.70	X3=11.24
For V= 6	X1=16.96	X2=13.63	X3=15.83
For V= 7	X1=22.97	X2=18.21	X3=21.22
For V= 8	X1=29.89	X2=23.46	X3=27.38
For V= 9	X1=37.72	X2=29.37	X3=34.33
For V=10	X1=46.45	X2=35.94	X3=42.07

You will note that, if something goes wrong, the triggering of the emergency braking should nearly be at the same level as the initiation of the normal braking, specially in the case of the reduction to 33%.

Let us take, for example a 6 m/s lift and let us imagine that the car does not start decelerating when it was supposed to.
The speed might be just under the adjustment of the overspeed switch, so the possibility of 115% of rated speed must be taken in consideration.

If the buffer stroke has been calculated on 33% of the speed, then the triggering of the emergency braking must take place between 17 m and 15.8 m ahead of the last floor level which, time wise, represents about 0.2 s.

If the deceleration did start at all, the speed will almost certainly be reduced at least to the required value by the end of the travel, so there is little advantage in having multiple check points.
However, for high speeds, there should be at least one such point for each possible jump (except perhaps for the one-floor jump).

To make things a little easier, you might:

- reduce the acceleration in normal operation, but you can not go lower than 1m/s (which makes a very comfortable lift) if you want to achieve a decent floor-to-floor time,
- increase the deceleration in emergency braking but it is limited by what has been taken in calculating the traction and we have seen that it was wise to limit the traction to the minimum needed to facilitate the relative slipping of the ropes in case of speed differential,
- lower the buffer as much as the local conditions permit.

10.4.3.3: The average retardation envisaged here is the mean retardation based on time because the limit would be meaningless if it were the one based on distance (see HB/10.4.3.1 above).

This limit of 1 Gn comes, I think, from an early American Code and has been successively adopted, whithout further thinking, by many other countries.

In fact, if this limit were rised to 1.2 or 1.3 Gn, the manufacturers could make better buffers as explained in Appendix APP:07.

With the present specifications, it is relatively easy to make a hydraulic buffer providing the area of the hole(s) through which the liquid flows,tends to zero at the end of the stroke.
Many designs are such that the deceleration is as high as allowed in the beginning of the stroke and it takes ages to complete the stroke, giving of course a very low average deceleration but a poor buffer.

For the time being, one has to live with this maximum of 1 Gn, but to make a reasonably good buffer, one should take into account the following additional requirements:

- the conditions defined in EN/10.4.3.3 should be met, not only with the rated load, but also with a load of 75 kg (1 passenger).
- the time during which the instantaneous acceleration exceeds 1.5*Gn should be as low as possible to avoid buffers acting only in the last part of the stroke. From that point of view, a buffer is generally satistactory if the mean deceleration exceeds 0.4*Gn.

In appendix APP:07, you will also find some guide lines for calculating a hydraulic buffer where all the energy is assumed to be dissipated in the liquid passing through a hole having a continuously varying area.
Many other designs are eligible.

10.5.1: Location of final limit switches

The Fig 23 illustrates the location of the various switches you will find at the bottom end of the travel of a lift with a rated speed of 1.6 m/s.

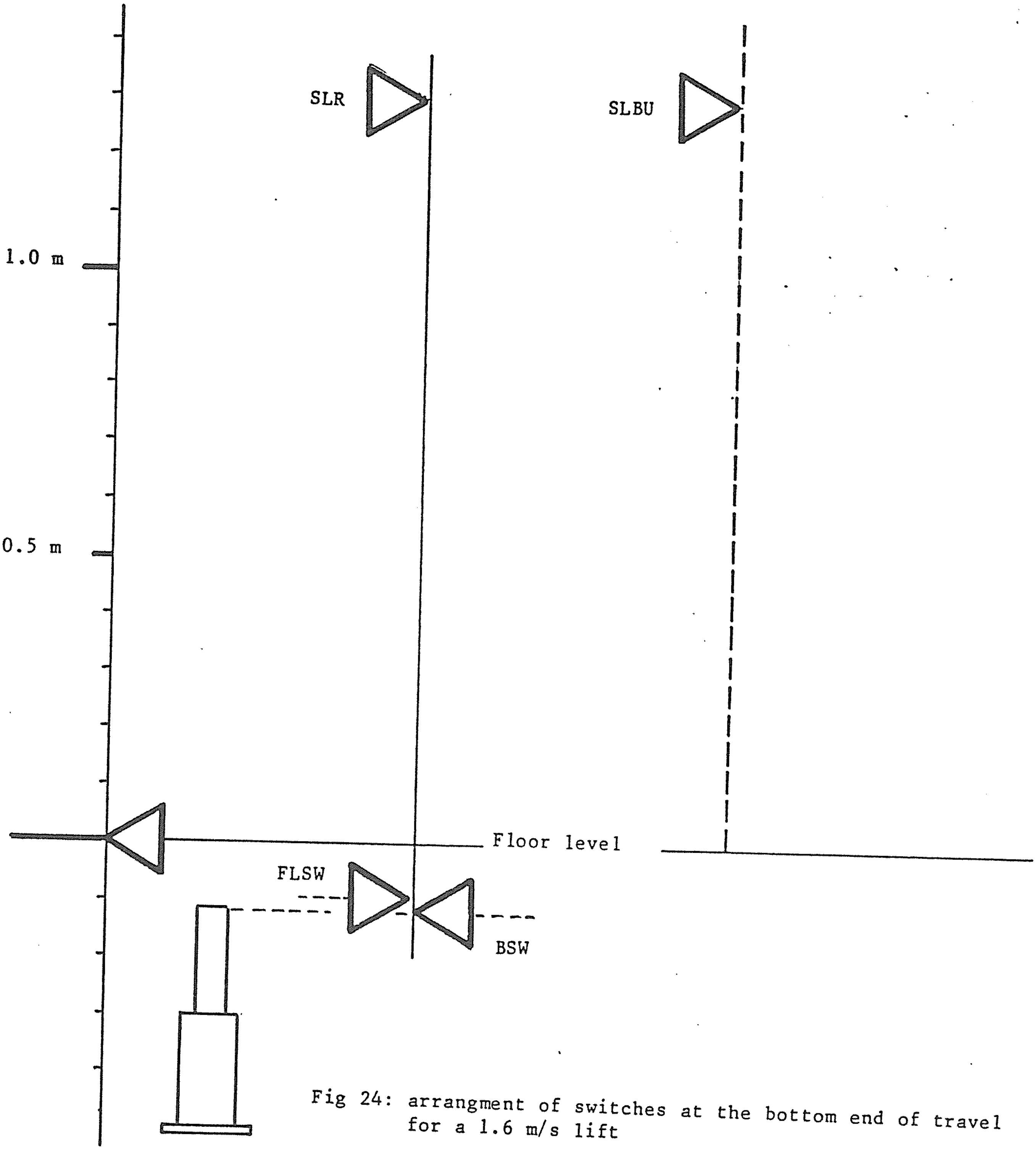


Fig 24: arrangment of switches at the bottom end of travel for a 1.6 m/s lift

The selector (SLR) will give the signal initiating the deceleration at about 1.3 m from the floor level (FL).

Whenever the selector is not "positive" (for example, if based on impulse counting) it is good practice, although not required by the EN-81 code, to either duplicate or replace the selector signal for stopping at the last floor by a "positive" selector back-up (SLBU).
This practice avoids going to the buffer when there is a simple miss of the selector. The system is generally such that, if the selector is out of pace, it will automatically reset each time the selector back-up operates.

It is to be noted that the North-American codes do require such a back-up whether the selector is "positive" or not.
In my opinion, this is unnecessary if the selector is "positive" but the Americans used to have high speed elevators with attendants who played the role of selectors and those attendants were probably often absent-minded. The American code makers are, I assume, still impressed by that.

Coming back to our EN code, the next switch will be the final limit (FLSW). The distance FL to FSW should just be over the expected stopping accuracy which means that it will be well within the levelling zone.

The head of the buffer and the associated switch (BSW) may come right after the FSW but it must come after because "...They shall operate before the car comes into contact with the buffers."

I do not know if there is any reason for this requirement.
In fact, the buffer switch BSW can do all that is required from the final limit switch FSW in addition to controlling the position of the piston and, if it were not for this unexplained requirement, the final limit switch would be redundant. It should be remarked that the American code allows even a partial compression of the buffer at each normal stop (it does not require a buffer switch).

For high rated speeds, several SLBU will be needed, depending on the jump, and, if short-stroke buffers are used, the devices required in EN/10.4.3.2 come in addition to the above.

NOTES at the end of CLAUSE 10

Clause 10.Note 1
See the comments the comments developed in this Hand-Book under the reference HB/Clause 5.Note 1/a,b,c,d,e.

CLEARANCES BETWEEN CAR, FRONT WALL AND COUNTERWEIGHT

11.1: General provisions

The experts of CEN/TC 10/WG 1, after discussion, decided not to mention any requirements for other clearances because the definitions are difficult. For example, it is obvious that some parts of the guide shoes and of the safety gears must come very close to the brakets of the guide rails.

The only objective requirement is that no moving part shall come in contact with stationery objects or other moving parts unless it is needed functionally. The clearances needed to achieve this will depend on:
- the clearances in other components of the system,
- the rigidity of the various components,
- the rate of wear of the various components subject to wear.

Considering that the clearances "...shall be maintained throughout the life of the lift", this rate of wear should be consistent with the frequency of inspections and the criteria for replacement or adjustment of components.

There are so many interrelated possible factors that the lift contractor must be left with the responsibility of achieving the objective.

The clearance needed between the car and the counterweight depends also on the same many factors. However, because of the possible consequences of a collision between car and counterweight, the C.C.E. wanted the EN code to mention, as a minimum, the 50mm clearance coming from ancient codes.

11.2: Lifts with car doors

For a better understanding, the present requirements should be analyzed along with the requirements of EN/5.4 which partially duplicate the present ones but also give additional indications.

The Interpretation N°38 is also helpful.

MACHINES

12.1: General provision

This means that the same machine may not be used for 2 different lifts but that 2 (or more) machines may be used for hoisting one single car.

This latter arrangment might be useful if the machines on hand do not have enough capacity. Some provisions are needed to insure a proper repartition of the power between the 2 machines but some advantages may be gained in the repartition of the load between the hoisting ropes by splitting the counterweight for example.

The case of one machine driving 2 cars balancing each other is not envisaged in the code but it is not clearly out of the field of application. Applications are rare and, to be on the safe side, a special authorization should be requested. The only case I know of is the modernization of the lifts serving the top floors of the Eiffel tower.

12.2: Drive of the car and the counterweight

The traction sheave is considered as a better drive and the only safe one at medium and high speeds.

The positive drives were maintained because they are the only elegant solution in specific applications (ex: very restricted areas with no possibility to secure more room in existing buildings or certain industrial structures). The limitations were set not only on safety criteria but also to discourage the use of such lifts in lieu of the regular traction lifts in new buildings and existing ones where room is available.

12.4: Braking system

12.4.1.1 : Although not stated, the code refers primarily to braking in case of emergency, even to a very specific case of emergency.

In fact, the braking "system" operates:

A - at each stop in normal operation; in this case, the order is given by the selector or from the selector back-up,

B - whenever almost any of the electrical safety devices listed in EN/Appendix A detects a faulty condition,

C - in the 2 cases mentioned in EN/12.4.1.1, i.e. loss of power supply either to the machine or to the control circuits.

Obviously, in the case of (A), the deceleration is under the best available control whereas, in the case (C), the code requires that the stopping of the lift be made by the mechanical brake.

In the case (B), the manufacturer has the choice, providing the deceleration circuitry can be considered as meeting the requirements of electrical safety devices (EN/14.1.2).

12.4.1.2: arrangment of the braking systems.

The double mechanical brake required in EN/12.4.2 must be provided whatever the type of motor drive; as regards the additional breaking means, I think that 3 different cases must be identified.

The 3 cases are:

- 1 speed AC drive:
The mechanical double brake is the only one used whether for normal operation (case A above) or for emergency (cases B and C).

- 2 speeds AC drive:
In normal operation (case A), the speed is first reduced to a fraction of the rated speed by switching from the high speed winding to the low speed winding; most of the kinetic energy is transformed in regenerative power; the mechanical brake is used only for the final stop.

In emergency (cases B and C), the mechanical brake alone must be able to stop the lift from 115% of the rated speed.

- Continuous speed control drives:
In normal operation (case A), the deceleration is controlled until zero speed in the most sophisticated speed controls; the mechanical brakes are applied only after full stop. In some less sophisticated systems, the mechanical brakes are applied when a residual speed remains.

In emergencies of the type (B), the deceleration and the final stop are sometimes controlled as in normal operation (case A): this is the best solution but the deceleration circuitry must meet the requirements of EN/14.1.2. If this is not the case, the emergency stop must be made by the mechanical brakes.

In emergencies of the type (C), the stop is made by the mechanical brakes eventually assisted by self-generated electrical circuits (ex: short circuiting of armatures in the Ward-Leonard systems).

12.4.2: Electro-mechanical brake

12.4.2.1/a
The formulae for calculating the brake and the behaviour of the lift under various braking conditions are given in APP:08/HB.

However, for the understanding of the tabulations and diagrams which follow, please note already that the rotating masses and the brake forces must be converted to translation values before attempting any mathematical approach. This to be consistent with the masses of the car and counterweight and all the forces directly applied to them.

The ratio of the rotating mass to the equivalent translating mass is equal to the square of translation speed, in m/s, divided by the square of (omega*RG) omega being the rotation speed in radians/sec and RG the radius of giration in meters.

The ratio of the brake force applied tangentially on the brake drum to the equivalent brake force applied to the car and counterweight opposite to the direction of travel is equal to the translation speed of the car divided by the tangential speed of the brake drum.

All the values refered to in the present chapter are exclusively values converted to translation.

They are all calculated for the typical lift defined at the beginning of this Handbook (see HB/FWD/12). The speeds envisaged range from 0.63 m/s to 6 m/s (rated speeds). For geared lifts, the motor speed is taken as 1500 RPM. Rotating masses and Radius of giration are estimated.

The calculated values give a good order of magnitude but, if you want to know exactly the values applying to a specific case, you have eventually to apply the formulae developed in HB/APP:08.

RM being the rotating mass converted to translation, the ratio of RM to Q (rated load) will be in the order of:

gearless 1/1 (4 m/s and up)	RM/Q < 1
gearless 2/1	RM/Q around 1
geared 1.6 m/s	RM/Q around 8
2 speeds AC 1 m/s	RM/Q around 30
1 speed AC 0.63 m/s	RM/Q around 50

12.4.2.1/b
In general, the "two sets" mentioned in EN/12.4.2.1 are 2 brake shoes and one of them alone must be able to slow down the lift going down with the rated load (the concomitance of an overload and of the failure of one brake shoe is not envisaged).

This means that with one shoe and rated load there must be a deceleration (let us call this deceleration G1 m/s2). Normally, the two shoes will be operating together and the deceleration achieved with 25% overload in the car can be calculated. It will depend on the characteristics of the lift. For the typical lift, the tabulation of Fig 25 gives the value of G2 in m/s2 (deceleration with 2 brake shoes and 25% overload) for various combinations of RM/Q and G1 (deceleration with only 1 brake shoe and rated load).

Fig 25 Tabulation giving G2 for various combinations of RM/Q and G1

	G1=0.0	G1=0.25	G1=0.50	G1=0.75	G1=1.00
RM/Q= 0.25	0.49	0.97	1.44	1.92	2.39
RM/Q= 0.50	0.47	0.94	1.42	1.90	2.37
RM/Q= 1.00	0.43	0.90	1.38	1.86	2.34
RM/Q= 2.00	0.36	0.84	1.33	1.81	2.29
RM/Q= 4.00	0.28	0.77	1.25	1.74	2.22
RM/Q= 8.00	0.19	0.68	1.17	1.66	2.15
RM/Q=16.00	0.12	0.61	1.11	1.60	2.09
RM/Q=32.00	0.07	0.56	1.06	1.56	2.05
RM/Q=64.00	0.04	0.53	1.03	1.53	2.03

To avoid ropes slipping, the deceleration taken into account for the traction calculation (see APP:06/HB) must be greater than the value of G2.

For 1 speed AC lifts, where the brake with its 2 shoes governs the deceleration in normal operation (i.e. around 1 m/s2) and where the ratio RM/Q will be 30 to 60, you can see that the value of G1 will necessarily be about 0.5 m/s2.

If for gearless lifts you were taking the same value for G2, the value of G1 would have to be in the order of magnitude of 0.25 m/s2 but this would not be very safe for 2 reasons:

- the calculated braking force is based on the friction factor of the lining in perfect conditions and on a perfect geometrical transport of the spring force. With any incident, the braking force could drop excessively.
- with low decelerations, the energy to be dissipated in the mechanical brake might increase to very high values and heat the drum rim so rapidly and to such an extend that the sheave arms could break.

The energy to be dissipated in the brake is the addition of the kinetic energy of the car, counterweight and rotating masses at the beginning of the braking and of the loss of potential energy due to the change of altitude of the travelling masses during the process.

The Fig 26 gives, for the typical 1000 kg lift, the energy to be dissipated in the brake for various initial speeds and various decelerations if there were no limit to the travel.

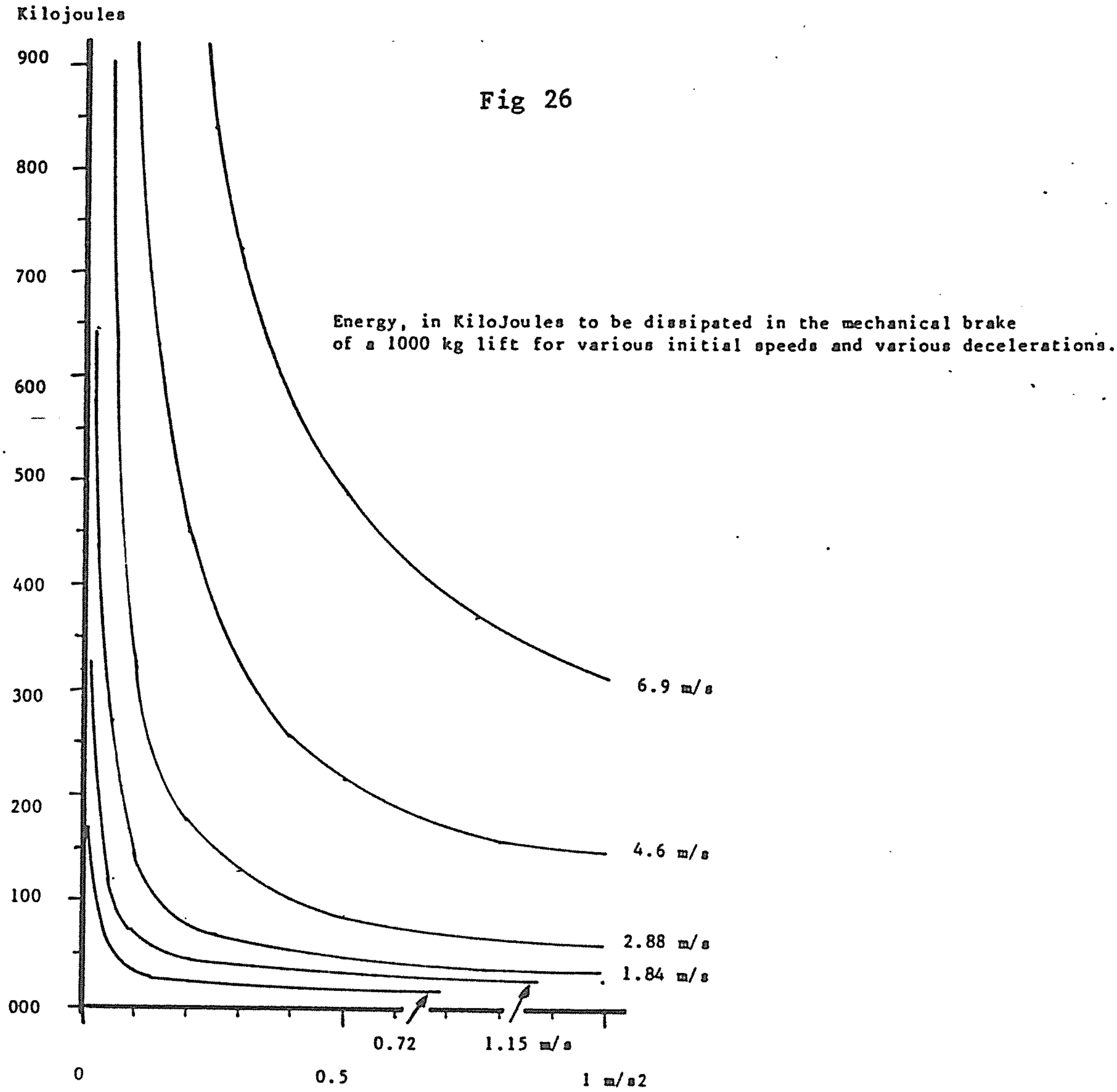


Fig 26

In Fig 26, the energy tends to infinite when the deceleration tends to zero but this would be possible only in the case of an endless travel.

The energy to be dissipated by the brake is maximum when the braking starts at the top of the well and the zero speed is reached just before hitting the buffer(s). If the braking distance is smaller there will be less energy coming from the lowering of the combined masses; if the zero speed is not reached before hitting the buffer, part of the kinetic energy will go in the buffer(s).

It is therefore easy to evaluate the deceleration corresponding to this maximum energy using the classical formula giving the deceleration, the initial speed and the stopping distance being known.

In any usual lift, the higher the speed the longer the travel.
The travel is normally completed in 20 to 30 sec in gearless lifts and in 45 sec for low speed lifts. On these bases, the deceleration corresponding to the maximum energy will be:

for 6.9 m/s	travel 150 m	deceleration 0.16 m/s2
4.6 m/s	100 m	0.11 m/s2
2.88 m/s	75 m	0.06 m/s2
1.84 m/s	50 m	0.03 m/s2
1.15 m/s	30 m	0.02 m/s2
0.72 m/s	25 m	0.01 m/s2

Those decelerations are much lower than any which can be envisaged.

To be reasonably safe when braking with one shoe, you should probably aim for a G1>= 0.5 m/s2, possibly 0.75 m/s2 for very high speed lifts.
But then the deceleration when braking with 2 shoes would reach 1.5 or even 2 m/s2 and it will not always be possible to avoid slipping of the ropes in the case of emergency braking.

Rope slipping in emergency braking is not forbidden by the code but should be avoided as much as possible because it will cause:

- uneven wear on the circumference of the sheave,
- some heating of the external wires of the ropes which could be detrimental for the ropes.

Moreover, the friction factor of wire ropes in cast iron grooves drops when the ropes are sliding (see APP:08/HB).
The Fig 27 illustrates the variation with time of the tangential speed of the traction sheave and of the speed of the car for a 2.5 m/s lift where the brakes have not been adjusted softly enough.

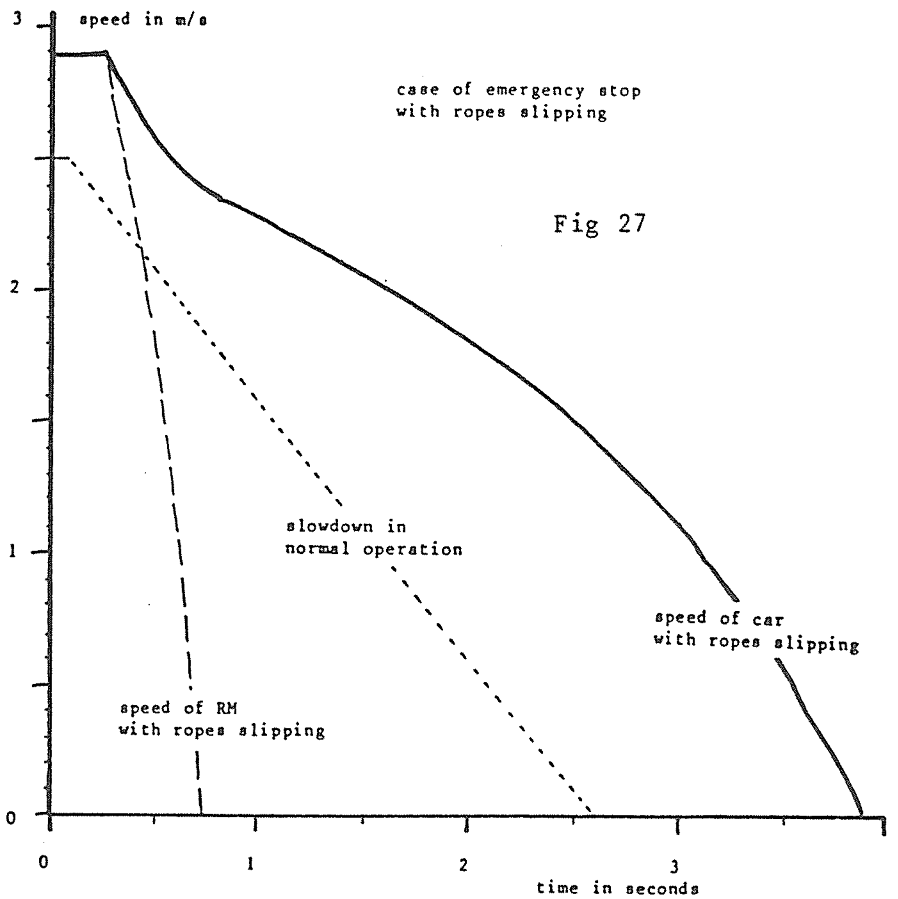


Fig 27

My own recommendation is that, knowing from the traction calculations the maximum acceptable deceleration, you select G2 safely lower than this limit in order to avoid the slipping of the ropes in emergency brakings.

This could lead to a very low corresponding G1 but the justification for this double brake contraption is, to say the least, questionable and, in my opinion, the risk of having an emergency braking with only one shoe is nil.

The only possible trouble would be that an Inspector requires to test thisbrake with one shoe. First of all, that test is not in the lists of EN/Appendices. Then, if it cannot be waived, you should insist on making the test in the lower part of the well (the amount of energy dissipated in the brake would be minimal).

Butthe adjustment will be delicate. It will require a well constructed brake enabling you to know exactly the pressure applied to the shoes for a given spring depression and fitted with good linings having a constant friction factor in a wide range of temperatures.

I recommend using articulated shoes with 2 separate linings, one at each end of the shoe. The point of articulation should be as close as possible to the intersection of the tangents to the drum at the point of application of the separate linings.

12.4.2.1/c
Very different from the gearless machines are the 1 speed AC geared machines. These also require a precise and stable adjustment of the mechanical brakes because the comfort of the deceleration depends solely on them. Moreover, the stopping accuracy depends on a specific combination of the braking force and of the RM value.

In the appendix APP:08/HB, you will find how to calculate the braking force and the flywheel once you have selected the deceleration and the stopping accuracy to be achieved.

12.4.2.1/d
The brakes for the 2 speed AC machines can be calculated on the same principles as the brakes for the 1 speed AC machines (starting of course from the fractional speed).

12.4.2.1/e
For geared machines using a continuous speed control, the RM is usually kept as low as possible and no flywheel is used. If a hand wheel is needed for emergency operation it is often made of aluminium.
Nevertheless, because of the high RPM of the rotor and associated rotating parts, the value of RM is around 10 and the adjustment of the brake is not a problem as it is for the gearless machines (see Fig 25 and 26).

12.4.2.1/f
The need for adjusting the braking force to a precise and stable value requires some precautions in the design of mechanical brakes.

The brake linings must be selected for having a friction factor not affected by the heat nor by the rubbing speed and, as much as possible, they should be independent of the cleanliness of the drum.

The geometry of the mechanical parts transmitting the spring (or weight) force to the brake shoe should be such that the pressure on the drum cannot be affected by assembly tolerances or by wear.
Some examples are given in APP:08/HB.

12.4.2.8: Brake linings
See also APP:08/5

12.6 Speed of the lift

All the calculations made in this Handbook are based on the rated speed. Exceptionally, the slip of the 1 Speed AC motor under load was considered.

When making tests on lifts, the actual speed must be recorded and results shall be interpreted accordingly.

12.7 Stopping the machine

The aim of these requirements is to make sure that the failure of one single component cannot prevent the machine from stopping and cannot allow the machine to restart accidentally after stopping.

This is of utmost importance when the power is supplied by static converters because the system has no internal inertia and might provide an extremely high surge of power which would wildly accelerate the machine.

The reading of the comments prepared by the WG1 of CEN when the first edition of the EN 81/Part I was issued (there are 5 of them) and of the Interpretations I.42 and I.108 (to date) is recommended.

12.8 Checking the slowdown for reduced stroke buffers

Relying on the mechanical brake to reduce the speed to the required value if something goes wrong with the normal deceleration control is, in my opinion, questionable.

First of course, an overspeed must be detected by a device accepted as meeting the requirements for the safety circuits. Some tolerance must be allowed on the speed before triggering an emergency stop.

Second, there is always a delay, inherent in the self of the solenoid and in the inertia of the mechanical parts, before the shoes start braking. This delay is often in the order of magnitude of 0.25 second and, at high speed, this means a considerable amount of travel.
Consequently, it is almost right after the point of the normal start of deceleration that the detection of the fault must take place and the emergency stop triggered (see HB/10.4.3.2).

Third, we have seen, in HB/12.4.2.1, that the adjustment of the brake is very critical if rope-slipping is to be avoided in the case of gearless machines. One can never be sure that an absent minded mechanic has not adjusted the brake too hard because this has no effect in normal operation. Remembering that the friction factor drops considerably at high rubbing speeds, it is difficult to say that the deceleration is under control in that case.

In my opinion, it is better to arrange the whole deceleration circuitry so that it can be considered as a safety device by itself.

13.1.1: Limits of application

It is very important to recognize that the whole lift, including the main switch and the car light switch, the control cabinet, its wiring to the safety devices, the control devices and the signal devices in the machine room, the well, the car and the pit is to be considered "as a machine with its built-in electrical equipment" exactly as it is universally accepted for a computer or a washing machine.

The heavy lines in the Fig 28 illustrate which parts of the circuits are to fulfil the requirements of the local authorities or of the power distribution company.

The Fig 28 represents the case of a single feeder line coming from the general distribution panel of the building to busbars from which the 3 switches required by EN-81 are taking the power.
Other arrangments, with 2 or even 3 separate feeders, are suitable.

In each machine room, there must be:

- 2 switches,(a) and (b), per lift (power and car lighting).
- 1 switch (c) for the lights and socket outlets of the machine room, the well and the pit.

Each switch must be associated with a suitable protection.

> NOTE:
> The responsibility for the supply and installation of the various parts of the electrical equipment should be defined clearly in the contract for installing the lift.
> See the comments in this Handbook under the reference HB/C relating to the "Technical Dossier".
> But, whoever instals any part of the electrical devices located in the machine room, in the well or in the pit, the responsibility for operating and maintaining it shall be left to the lift maintenance crew once the lift is in operating condition.

13.2.2: Components of safety circuits

All the components other than the ones specifically mentioned in the EN code, such as transistors, capacitors, Reed relays, etc, should be of good quality and have as long as possible a lifetime for the reasons explained in HB/14.1.2.1.1.

13.6 Lighting and socket outlets

The lighting and socket outlets illustrated in Fig 27 are required in the EN paragraphs 5.7.3.4 - 5.9 - 6.3.6 - 6.4.7 - 8.15 and 8.17.

Although the lighting of the landings is never associated with the lift construction, you should not forget to call the attention of the lift owner to the requirement of EN/7.6 (50 Lux at floor level).

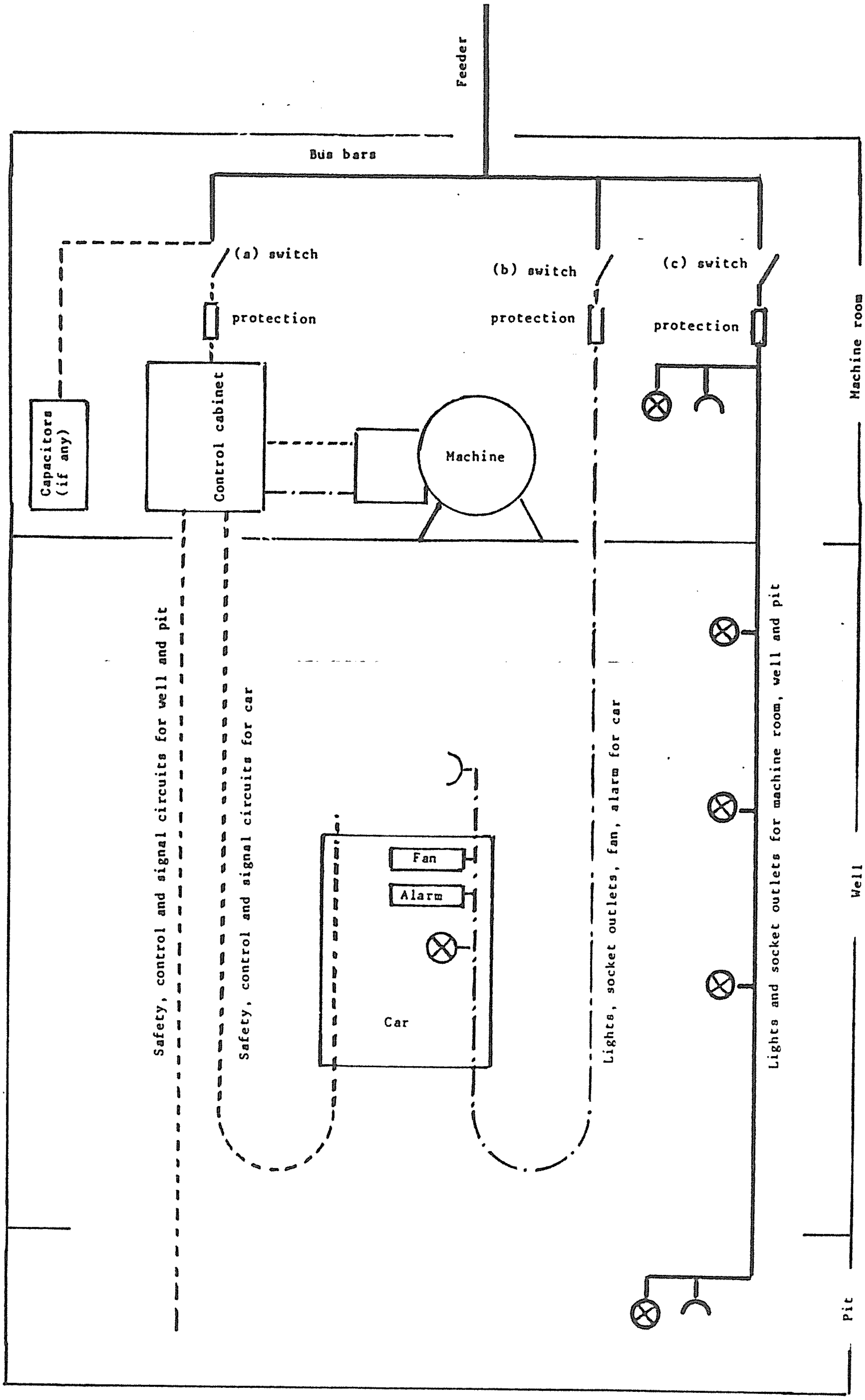


Fig 28: The 3 types of circuit in a lift installation

14.1.2.2.1: Positive mechanical separation

When the contacts are opened by a spring being an integral part of the switch, it is not considered as a "positive mechanical separation".

However, there are cases where the contacts are opened by the action of a spring which is part of the mechanism that the contact has to monitor or by the action of a weight (ex: tension pulley).
If the spring is strong or the weight heavy, no inspector will ever question the "positive mechanical separation".

This demonstrates that, in-between, there must be cases were discussion is possible. I suggest that, in case of doubt, you ask, beforehand, for a certificate from one of the authorized inspectors. It will help you later, surely with him, perhaps with others.

14.1.2.3: Safety circuits

14.1.2.3/a
A safety circuit may be made of components scattered throughout the lift. For this reason, I am not sure that type tests would be suitable.
In my opinion, an analysis of the proposed circuitry and of the quality of the components to be used should be of a more general application and probably give better indications.

14.1.2.3/b
In the evaluation of a safety circuit, the concept of the relativity of the "safety" is to be applied.

A safety circuit is considered as "safe", if the probability of having a dangerous occurence with the circuit out of order is low enough.

14.1.2.3/c
As far as I know, the safety circuits may be divided into 2 categories:

a) the ones which open at each of one of the cycles of normal operation of the lift.
b) the ones which, in normal operation, open only when something goes wrong.

An example of (a) is the check on the position during levelling and relevelling which is necessarily open between floors.

An example of (b) is the check on the overspeed which opens only if the speed regulation is out of order and the car accelerating.

There are various cycles in the operation of the lift:

- each start and stop
- each direction reversal
- each stop at a given floor
- each start of the MG set (Ward-Leonard)
- each operation of the main switch
- etc..

To a certain extent, the monthly visit by the service mechanic or the yearly inspection could be considered as cycles in the life of the lift.

14.1.2.3/d:
For type (a), one of the solutions is to duplicate the detectors, check the parallelism of their signals and stop the lift not only if both messages are "NO" when it should be "YES", but also if one of the messages is different from the other. In the latter case, the lift should be out of service until repair of the defect.

Another solution is to use a single detector but check, in an adequate part of the cycle, that the message is returning to "NO" when it should be. The lift should be stopped until repair if the condition is not satisfied.

A third, and better, solution is proposed below in HB/14.1.2.3/g.

14.1.2.3/e:
For type (b), you will have to select a suitable cycle and, at a convenient time in this cycle, to simulate somehow the phenomenon which the circuit is to monitor.

If the circuit does not detect the simulated phenomenon, then the lift should be stopped until repair.

Another, and better, solution is proposed below in HB/14.1.2.3/g.

14.1.2.3/f:
So, one way or another, the effectiveness of the safety circuit is checked at each of the selected cycles.

In the case of the duplicated detectors, it is assumed that the probability of having the 2 detectors freezing together in the "YES" position during the same cycle can be neglected.

In the case of the single detector with cyclic simulation, it is assumed that the probability of having the dangerous phenomenum occuring in the cycle preceeding the report of the detector's failure can be neglected.

14.1.2.3/g
For both the (a) and (b) types of circuits, an excellent solution is to always combine:

- the duplication of detectors,
- the checking of the parallelism of the messages,
- the check of the ability of switching from "YES" to "NO" position, either in the course of the normal operations (type (a) circuits) or by the cyclic simulation (type (b) circuits).

Indeed, the probability of having during the same cycle:

- the freezing of the 2 detectors in the "YES" position,
- the occurence of the dangerous situation,

is infinitesimal and can certainly be disregarded.

14.1.2.3/h:
The probabilities envisaged above depend upon the following factors:

- the length of the cycle
  (the shorter the cycle, the lower the probability),
- the lifetime of the detector(s)
  (the longer the lifetime, the lower the probability of simultaneous failures or of failure in the same cycle as the occurence of the dangerous situation)
- the frequency of occurence of the danger under consideration.

Lifetimes and cycle lengths are, depending on the components, expressed either in time or, more often, in number of operations.

Obviously, to make the best possible safety circuit you should select the shortest possible cycle and the longest possible lifetimes. The frequency of occurence of the dangerous situation does not depend on you.

A mathematical approach for the evaluation of the "quality" of a safety circuit has been attempted but there are 2 difficulties for the application of this evaluation method:
- manufacturers of electronic components are reluctant to commit themselves on the lifetime of their products,
- statistics about the occurence of the various "dangerous situations" are completely lacking.

Statistics related to lifts are scarce and the existing ones are based on reported accidents, but all occurences of a dangerous situation do not lead to accidents and accidents are not always reported.

Taking all into consideration, I am convinced that, with designs as described above and with good quality components, the probability of having an accident due to a safety circuit is low enough to be disregarded.

14.1.3: Protective devices.

14.1.3/a
There is no clause 14.1.3 in the EN-81 but I want to call the attention on the devices required in:

EN/7.5.2.1.1.3	landing door protective device.
EN/8.7.2.1.1.3	car door protecting device.
EN/8.7.2.2	vertical door speed control.
EN/8.8	sill protection for doorless cars.
EN/10.6.2	slipping rope control for traction lifts.
EN/12.4.2.3	mechanical brake operation.
EN/12.7	stopping of machine and checking.
EN/14.1.1.1(j)	phase reversal.
EN/14.2.1.2(b+c)	speed control during levelling operation.
EN/14.1.1.3	protection against earthing.
EN/14.2.1.3(d)	speed control during inspection.
EN/14.2.1.4.6	speed control during emergency operation.
EN/14.2.1.5(c)	speed control during docking.

14.1.3/b:
The code does not require electric safety devices but, since the above devices are somehow related to the safety of people, I recommend using at least one of the simple designs of safety circuits.

In my opinion, this is especially important in all cases of speed control and in the case of the stop.

If it is not possible to use safety devices, be sure to use good quality components with the longest possible lifetimes and instruct the maintenance mechanic to check the component regularly (the visit of the mechanic would be considered as a cyclic check).

14.2: Controls

14.2.1.1: Normal operation
The Standard ISO 4190/5 "Control devices, signals and additional fittings" is based on the EN-81 requirements and gives many more useful details on the buttons, switches and signals suitable for each type of control.

Following this Standard is by itself a factor of safety because passengers will progressively get used to these devices, understand the meaning of the signals and use the lifts properly.

The proposed signals use ideograms to avoid language barriers.
If additional signals are necessary, it is highly recommended to avoid, as much as possible, written messages for the same reason.

14.2.4:
See also the ISO 4190/5.

NOTICES and OPERATING INSTRUCTIONS

15.1: General provisions

For the reasons already exposed in HB/14.2.1.1, the application of the standard ISO 4190/5 is recommended.

16.1.1:
As required in EN/16.2.1, the technical information to be recorded in the register when a lift is brought into sevice shall comprise:

- the basic characteristics of the lift,
- the characteristics of the ropes,
- the statements of conformity for interlocks, doors, safety gears, buffers and overspeed governor.

This is practically all the elements listed in EN/appendix C.

To avoid duplication of work, the Technical Dossier, if required for analysis prior to signing the contract, should be the first part of the Register which will be required later.

16.1.2
By the same token, the information to be supplied to the Inspector at the acceptance of the lift is exactly the data required in the register, so the logical thing to do would be to prepare the Register in due time for the reception and use it later to record the events related to the lift.

There is no required format for this Register but you can find in the Appendix APP:10/HB the translation of a document prepared by the Belgian association FABRIMETAL (lift section) as guide-lines for the lift contractors. The aim was to make it easy by presenting the data always in the same orderly way, so that nothing is forgotten and anyone can easily find the information he seeks.

The first 2 pages of this document along with the Attachment 1 (layout) would constitute the dossier needed when applying for the preliminary authorization (if required). In Belgium, the electric schematic diagrams and the tests certificates are usually not required at this stage. They may be required in other countries or the customer may want them.

The document required by the Inspector prior to the acceptance of the lift would consist of the pages 1 to 5 with the attachments:

1 Layout (see the requirements in EN/Appendix C parag C.2)
1A Simplified diagrams of the power and lighting circuits
1B Simplified diagram of the safety devices wiring
2 Justification of the suspension ropes
3 Traction calculations
4 Justification of the car guide rails
5 Justification of the counterweight guide rails
6 Copy of the test certificate for the doors interlocks
7 Copy of the fire test certificate for the landing doors
8 Copy of the test certificate for the car safety gear (with declaration of the adjustment as the case may be)
9 Copy of the test certificate for the ctw safety gear (if any) (with declaration of the adjustment as the case may be)
10 Copy of the test certificate for the car overspeed governor
11 Copy of the test certificate for the ctw overspeed governor (if any)
12 Copy of the test certificate for the car buffer(s) (if hydraulic) or justification of buffer(s) according to EN/10.4.1 (if spring)
13 Copy of the test certificate for the ctw buffer(s) (if hydraulic) or justification of buffer(s) according to EN/10.4.1 (if spring).

16.2: Register

To make up the register, you take the previous document and you attach:

- a copy of the acceptance certificate,
- a section for making notes of all routine maintenances, incidents, accidents, repairs, replacements, modifications,
- a section for filing a copy of the inspection reports.

This format of Register is suitable for any traction lift but its first part, the one containing the technical data, can be simplified for appartment lifts and most of it can be preprinted for standardized lifts.

The copies of the test certificates may be replaced, in the Register, by the characteristics of the five corresponding elements.

16.3: Servicing

The lift contractor who installed the lift is definitely the one who is best qualified for maintaining the lift. Moreover, allowing anybody else to meddle with the lift during the warranty period would alleviate his responsibility in case something went wrong.

Even if the lift contractor does not assume this responsibility throughout the liftetime of the lift, the maintenance should be made by a reputable firm, having a competent technical staff, a solid financial base and a good insurance contract.

A lot of microscopic firms or even individuals are nowadays offering to maintain lifts at undercut prices. They have a natural tendency to use makeshift solutions instead of using genuine parts for replacement and, in case of an accident, they often do not have the financial strength or an insurance sufficient to face their responibilities.

APPENDIX A to EN 81/I

EN/Table 4: Conditions for use of electric safety devices

The concept of "safety circuits" was new for most experts in the WG 1 of EN/TC 10. and there was little experience with these devices.

Wisely, the experts decided to limit the application to the cases where the use of safety contacts was impossible or at least questionable and where the safety circuits were an elegant solution.
Anyhow, a good safety circuit is often more expensive that the corresponding safety contact where the latter can be used.

Beside listing the allowed applications for the safety circuits, the tabulation of Table 4 is a convenient reminder of all the safety devices to be provided in the lift.

The list of the protective devices to be provided can be found in this Handbook under the reference HB/14.1.3/a.

The list of the required lighting appliances and socket outlets can be found under the reference HB/13.6. (see also Fig 27)

The list of the switches required in the machine room can be found under the reference HB/13.1.1.(see also Fig 27)

APPENDIX B to EN 81/I

EN/Figure 4: Unlocking triangle

This unlocking triangle, which is already normalized in other trades, was selected to make things easy for the firemen when they are called to rescue people trapped in a lift car.
It is hoped that they all will have an "elevator key" and refrain from hacking the door down instead of opening it.

So far, this unlocking triangle seems to be satisfactory in most countries. Reports from repeated abuses came from 2 countries only (United Kingdom and Sweden). Considering that any system can be abused by malevolent people, I do not think this is a bad record.

APPENDIX C to EN 81/I

Technical Dossier

More and more complete technical information has to be suplied gradually by the lift contractor from the time he submits his proposal for examination by the prospective customer or his consulting engineer to the time of commissioning the lift when a complete "Lift Register" must be put at the disposal fo the lift owner.

The logical thing to do, is to build gradually this Register by successive additions to an original few pages.
This is already explained in this Handbook under the references HB/16.1.1 and HB/16.1.2 and you will find Appendix APP:10/HB the translation of a document prepared by the Belgian manufacturer's association FABRIMETAL as guidelines for their adherents.

APPENDIX D to EN 81/I

Examinations and tests before going into service

As specified in EN/16.1.2, these examinations and tests shall be carried out by "a person or organization approved by the public authorities".
In some countries, the lift contractor himself is entrusted with this responsibility but, in any event, he should be present when these tests are carried out. We will call "Inspector" whoever is allowed to commission the lift and sign the acceptance certificate.

D.1: Examinations.
Whether a "preliminary authorization" was required or not, the Inspector will need the "Technical Dossier" which, as explained earlier (see HB/C and HB/16.1.2) is the first part of the "Register" which is to stay with the lift throughout its lifetime.

The EN/81 code does not require that the Inspector compares the lift with the commitments of the contract signed between the customer and the lift contractor. It is the subject of another examination but is sometimes carried out at the same time.

D.2: Tests and verifications.
The tests described in this paragraph are the most severe that can be carried out without endangering the equipment.
The lift contractor should object if more severe tests were requested by the Inspector and he should be relieved of his responsibilities as regards the possible future consequences.

The tests are described in the Code and I will comment only on:

- the testing of the overspeed governor,
- the testing of the safety gear.

D.2.(i)(1): The code does not say how to test the tripping speed.

For variable voltage lifts, the lift speed can be increased gradually up to the tripping speed of the governor.
For other lifts, some governors are provided with an auxiliary sheave, smaller than the regular one, which make the governor rotate 30 or 40% faster than the normal speed when the lift is running at contract speed.

Sometimes, the governor can be driven by a friction wheel mounted on an electric drill.

However, since the code says, in EN/9.9.6.7, that "the rope shall be easily detachable from the safety gear", you might also, if you can rely on the performance of the overspeed governor, use the alternative testing procedure described later in this Handbook under the reference HB/F.4.2.2.2 for a better evaluation of the performances of the governor.

For doing so, the car should be parked high in the well and the rope detached from the safety gear.
After attaching the proper mass to the rope, let it go down below the car. The governor assembly will accelerate like an Attwood machine and trip when the tripping speed is reached. If you have the proper recording equipment, you can read the tripping speed and calculate the pull on the governor rope by analysing the deceleration of the mass.

D.2.(j) Test of the safety gear before going into service

It cannot be stressed enough that this test is meant for checking the mounting, the liaison with the governor, the guide rails and their fixing to the building <u>and not the safety gear itself</u>.

D.2.(j).(1): Instantaneous safety gears
The test with rated load at rated speed is especially important in the case of instantaneous safety gears because, as explained in HB/Clause 5.Note 1/c, part of the kinetic energy of the eventually falling car would be dissipated in the guide rails, their fixing and the building itself.

Conducting the test with the brake open and the machine running is not very important here because the rate of deceleration imposed by the instantaneous safety gear is so high that, even if the machine was disconnected from the mains and if the brake applied, the ropes would still have to slip backwards during the operation (generally in this case the machines have a heavy flywheel and the machine deceleration, converted to translation, will be in the order of magnitude of 1m/s despite the braking due to the backwards slipping of the ropes).

D.2.(j).2: Progressive safety gears
In this case, the testing procedure would be important if the sliding distance were prescribed. But, since the reason for the test is only to verify that the guide rails system can withstand the maximum force applied by the safety gear shoes, it is also rather indifferent.

Indeed, one must remember that the force applied by the safety gear is <u>absolutely independent of the load in the car and of the forces applied to the car by the ropes</u>.
For a given gliding speed, it depends only on the material used for the braking shoes and on the force applying them to the rails.
Moreover, for the usual cast iron shoes, the maximum force occurs when the speed is nearing zero so making the test at low speed is fully justified. With other materials or with controlled braking force, this force would ideally be constant and a test at low speed would also be suitable.

Requiring a test at high speed is, not only completely meaningless, but also damaging for the lift. Each time the safety gear sets, it loses part of its ability to stop the lift the next time. Some designs are only capable of a few operation, as recognized in the Type testing procedure.

Making sure that the machine keeps running and increasing the load and the setting speed does not change the forces applied to the guide rails. It is consequently meaningless.

It would make sense however, if a sliding distance was specified. In that case, a test on the site with counterweight attached could, to a certain extent, be used as a substitute to the test in free fall. It would still be damaging and is not recommended outside of test towers, but this testing procedure could be used by manufacturers for orientation tests before presenting a safety gear to the Laboratory for type testing.

In the Appendix APP:11/HB, the formulae for calculating the gliding distances are developed. Tabulations give the values which should be expected in usual applications.

In the Appendix APP:12/HB, the formula used for expressing the friction factor as a function of the speed is justified.

In the Appendix APP:13/HB, the validity of the sliding distances allowed by the ANSI code are questioned. They seem too high.

APPENDIX E to EN 81/I

Periodical examinations and tests

They are very important because, for many components, they are the only way of detecting excessive wear or deterioration. This is the case for all the mechanical components with the exceptions listed in the Basic Assumption N°3 (see HB/Gen'1-0.7/a). Even for the listed components, it is of course much better to anticipate the coming failure and replace the faulty component before failure.

APPENDIX F.1 to EN 81/I

Landing door locking devices

F.1.1.1 : Field of application.
Emphasising the requirements of this clause, I call the attention to the comment I made in HB/7.7.3 where I wrote that, in my opinion, the whole door system contributes to the "locking" function and that a lock should never be tested without the door to which it is to be associated.

F.1.2.2.2 : Static test.
In the case of multipanel doors where advantage is taken from the interpretation N°100, the ropes (or belts, or chains) are used only for driving the panels and not for locking them.

If they fail during the endurance test, it does not mean that the locking device should be rejected. If they fail too often, it would be a sign of poor quality but it has no impact on the safety.

To control the adequacy of the locking components, the stops or hooks which ensure that all the panels are locked should be submitted to the static and dynamic tests after removing the connecting ropes, belts or chains.

APPENDIX F.2 to EN 81/I

Please read the comments made under the reference HB/7.2.2.
Let us hope that the CEN/TC 10/WG 3 will come up with a draft proposal.

Directives for the approval of safety gears

F.3.2.4: Determination of the total permissible mass
(instantaneous safety gears)

F.3.2.4.1
Some of the symbols need to be explicited.

- Total permissible mass:
  The certificate will give the total permissible mass corresponding to a given rated speed.
  It is for each application that the following condition must be verified:

$$P1 + Q + P2 + P3 \leq \text{permissible mass}$$

- Tripping speed of the safety gear
  Logically, the maximum tripping speed allowed by EN/9.9.1 should be used here. This means that it will be:
    - 1.0 m/s for the captive roller type
    - 0.8 m/s for all other types.

This, however, is true only for car safety gears. There is a loophole in the code as regards the use of instantaneous safety gears for counterweights (nominal speeds up to 1 m/s - see EN/9.8.2.3).

The Interpretation N°90 did not propose any solution.
I suggest that, if the safety gear is to be applied to a counterweight with a nominal speed of 1 m/s, the following tripping speed be considered:

- 1.5 m/s for the captive roller type
  (the other ones should not be used at that speed)

It is only for lifts with very heavy loads and low speeds that another tripping speed should be considered (EN/9.9.2.2).

In the formula for calculating the distance of free fall, the code did not include the additional travel of the mass during the energy absorbtion phase, i.e. after the clearance has been taken up and until the stop of the car, travel needed by the roller to open the jaws.
For the large safety blocks used for big freight elevators at low speeds, this additional travel is in the order of 0.04 m. It is by no means negligible by comparison with the other terms of the addition.

F.3.2.4.2: Total permissible mass
The assumption is that all the kinetic and potential energy has to be dissipated in the blocks. This is certainly on the safe side, but acceptable, for plain instantaneous safety gears as explained earlier in this Handbook under the reference HB/Clause:Note 1c.

This assumption is completely wrong in the case of instantaneous safety gears with buffered effect. The required buffer is designed to dissipate all the energy and it would be reasonable to admit that the blocks will take only a portion of the energy to be dissipated.
In fact, the blocks open just what it takes to create the force needed to counteract the reaction of the hydraulic buffer(s). In the process, and almost as a by-product, they dissipate some energy.

If the need arises, I suggest that an Interpretation be requested.
In my opinion, the permissible mass should be twice the one allowed for a plain instantaneous safety gear.

When analysing the problem, one should not forget that the potential energy lost during the operation corresponds to a distance greater than the 0.04 m mentioned earlier because of the stroke of the buffer.

F.3.3: Progressive safety gears

F.3.3/a
Let us recall here what the code requires in EN/9.8.4:

> "Retardation. For progressive safety gear the average
> "retardation in the case of free fall with rated load in the
> "car shall lie between 0.2 Gn and 1.0 Gn.

The Clause 9.8 of the code requires nothing else.
Although not specified, the retardation envisaged is the mean retardation based on time, not the one calculated from the sliding distance.
The Clause is mute as regards the instantaneous values of the peaks.

You will note that the testing procedure imposes much narrower limits for the mean retardation and sets a limit for the peak instantaneous deceleration. This is a little odd.

The explanation is that the procedure was elaborated on the theory of a constant friction factor with possible erratic deviations up to 20% (plus or minus) from the normal value.

Aiming at 0.6 Gn, meant a retarding force of 1.6 times the weight of the falling mass. Reckoning with a possible deviation of 20% for the friction factor meant that the retarding force could be anything between 1.2 times and 2.0 times the weight of the falling mass.

A braking force of 1.2 times the falling mass corresponds to a deceleration of 2 m/s2 which was considered as the minimum acceptable to ensure that the safety gear would start decelerating at all.

So, aiming at less than 0.6*Gn would have not been safe and checking that the instantaneous values of the braking force were never dropping below 1.2 times the weight of the falling mass did make sense.

I am not sure why a maximum was set; probably by comparison with the requirement for the hydraulic buffers.

F.3.3/b
However, in the course of years, I became convinced that the friction factor depended on the rubbing speed (for cast iron on steel).

You can find, in the Appendix APP:12/HB, the data and information collected. A formula for expressing the value of the friction factor as a function of the speed is proposed and compared with the data collected.

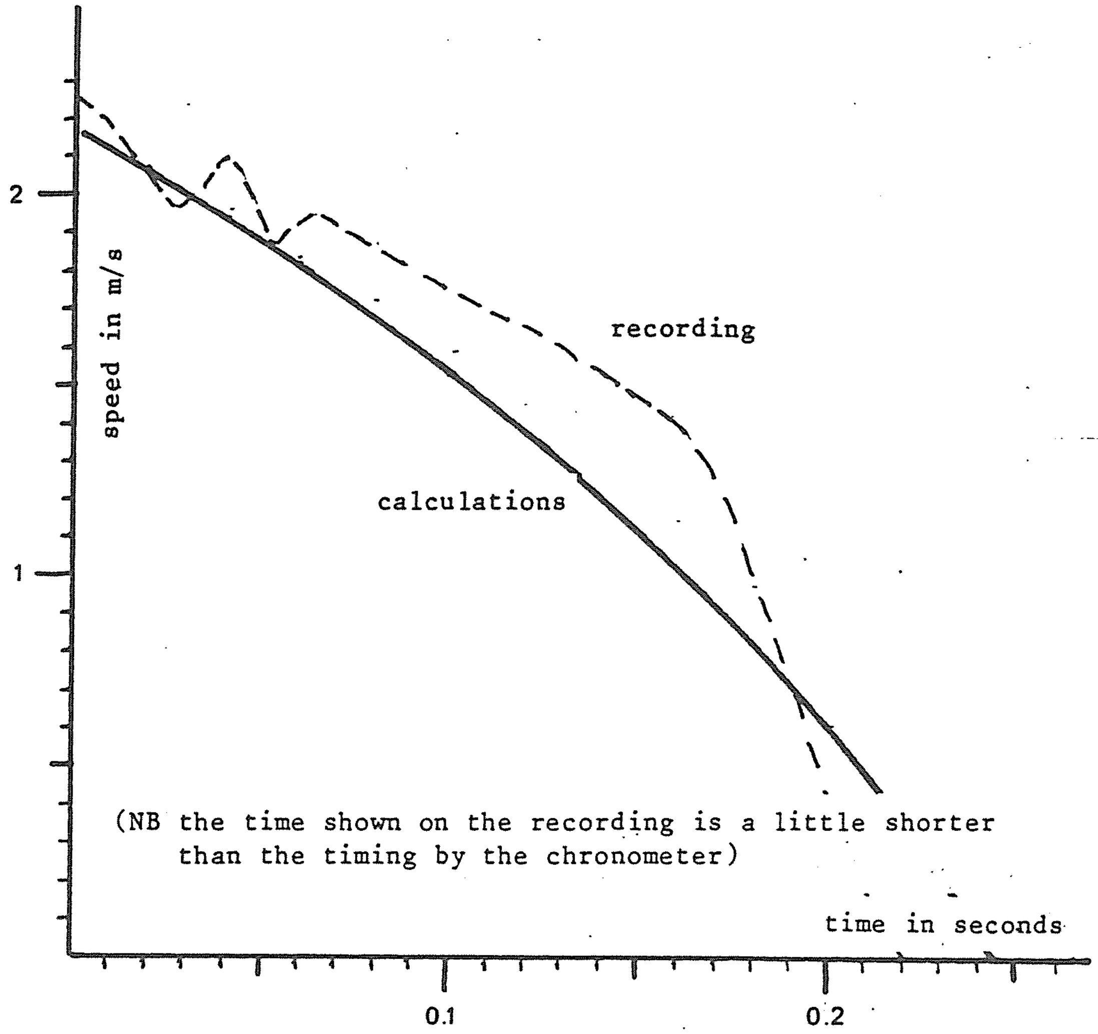
Using this formula, we can draw the speed/time curve which can be expected for a given triggering speed and a given initial deceleration.

Let us, for example, draw the curve corresponding to the actual test illustrated at the Fig 2 of the APP:12/HB.

Fig 29 Comparison between the values given by the formula and the values recorded during an industrial test.

Solid line: values calculated with the formula $\mu = \dfrac{0.4}{1 + 0.3*V}$

Dotted line: values taken from the recordings.



For this test, the setting speed was 2.17 m/s.
During this test, besides recording the speed, other values were measured and can also be compared with the corresponding values calculated by the computer using the formula with an initial deceleration of 5.5 m/s2.

Values		reported	calculated
Time	sec	0.240	0.245
mean deceleration	m/s2	9.04	8.82
peak deceleration	m/s2	15.7	15.4
distance	m	0.38	0.31

(the deceleration was higher than required by CEN).
The concordance of the values is satisfactory, especially considering that the test was made in an industrial test tower. Recordings made in a laboratory might be more accurate. But, at the same time, the curves show that some deviations from the theoretical values are to be expected.

F.3.3/c

For any combination of setting speed and initial deceleration, we can anticipate the behaviour of the safety gear by using the formula for the friction factor in the applicable equations.

To aim at the mean deceleration of 0.6*Gn required by the testing procedure, the initial deceleration must be selected in relation to the setting speed. The following formula has been developed empirically:

$$\boxed{G\ \text{initial} = \frac{-\,0.58 * Gn}{1.4 + 0.2*V}}$$

The validity of the formula has been verified by calculating the mean deceleration resulting from the application of the formula for setting speeds ranging from 1 m/s to 10 m/s.

More exact values were researched by having the computer successively testing increasing initial decelerations until the mean deceleration of 0.6*Gn was reached. The following tabulation (fig 30) shows that the formula gives acceptable values, especially for speeds exceeding 2 m/s.

Fig 30

Calculation of initial deceleration for meeting CEN requirements
Evaluation of the formula Go=Gn*0.58/(1.4+0.2*V)

	Exact values		Approx values by formula	
V m/s	Go m/s2	Gmean m/s2	Go m/s2	Gmean m/s2
1	4.31	6.06	3.56	5.13
2	3.56	6.15	3.16	5.63
3	2.94	6.06	2.84	5.94
4	2.54	6.02	2.59	6.02
5	2.32	6.06	2.37	6.13
6	2.14	6.03	2.19	6.15
7	1.98	6.01	2.03	6.11
8	1.90	6.08	1.90	6.08
9	1.78	6.02	1.78	6.02
10	1.72	6.04	1.67	5.97

The above tabulation can be used for orienting the choice of the force pressing the shoes on the guide rails.

The formulae to be used are of course:

$$\text{Friction factor at initial speed} = \frac{Kp * 0.4}{1 + 0.3*Vinit}$$

$$(\text{force applying the shoes}) * \mu o = (\text{falling mass}) * (G\ \text{initial})$$

As explained in the appendix APP:12 (see APP:12/9), I do not have enough information to propose any way of evaluating the pressure factor Kp.
In a first approach, you may take Kp=1, then find out, after a few tests, the value corresponding to your design and to the material used.

F.3.3/d
Before going further, it must be stressed that all the mathematical developments related to the progressive safety gears are based on the formula relating the friction factor to the rubbing speed (see above).

Only the case of ordinary cast iron shoes applied with a constant force is envisaged here. Other possibilities will be mentioned later.

The choice of this formula has been justified in the appendix APP:12/HB, however one must remember:

- that actual values may deviate from the calculated ones, (deviations generally limited to + or - 10%, but exceptionally reaching 15% to 20%)
- that, for speeds exceeding 4 m/s, the only test reports came from Dr Feyrer (see Fig 4 and 5 in APP:12/HB)

Nevertheless, the formulae, the tabulations and the curves derived from this expression of the friction factor, give good indications on what should be expected when testing and may be used for orientation purposes when designing safety gears.

I hope that my mathematical developments are correct. I think they are, they have been cross-checked whenever possible.
However, before paying attention to the decimals, the results should have been confirmed by the experience which will be accumulated by the laboratories in charge of type testing the progressive safety gears.

F.3.3/e
For example, let us calculate what happens if the setting speed is 2 m/s.

Looking at Fig 30, we select a initial deceleration of 3.47 m/s2.

After calculating the values of the speed and of the deceleration in relation to the time, we can draw the curves (Fig 31 and 32 next page)

The mean deceleration based on time, 6.08 m/s2, is in line with the CEN requirements (0.6*Gn).

The instantaneous deceleration exceeds 1*Gn, but during a very short time: less than 0.03 seconds.

F.3.3/f
Using the formula of F.3.3/c, the initial deceleration can be calculated for the setting speeds ranging from 1 to 10 m/s.
It happens to be the minimum one, providing it is a deceleration.

With the initial deceleration, we can calculate the force needed for applying the safety shoe. This force remains constant.

When the speed is nearing zero, the friction factor value tends to 0.4. This corresponds to the maximum retarding force and consequently to the maximum deceleration which can be calculated.

The Fig 33, on page 92, shows between what extreme values the deceleration must vary in the course of the safety operation to achieve a mean deceleration of 0.6*Gn as required by the code.

(continued on page 93)

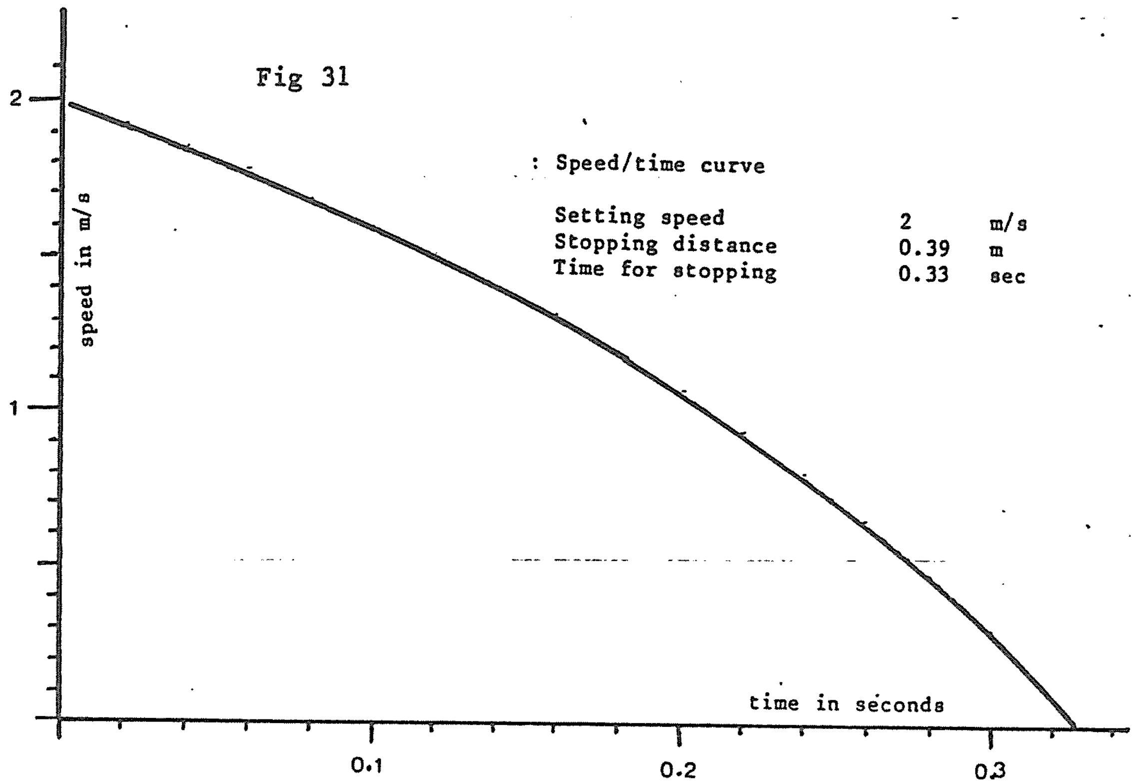


Fig 31
: Speed/time curve
Setting speed 2 m/s
Stopping distance 0.39 m
Time for stopping 0.33 sec
speed in m/s
time in seconds
2
1
0.1
0.2
0.3

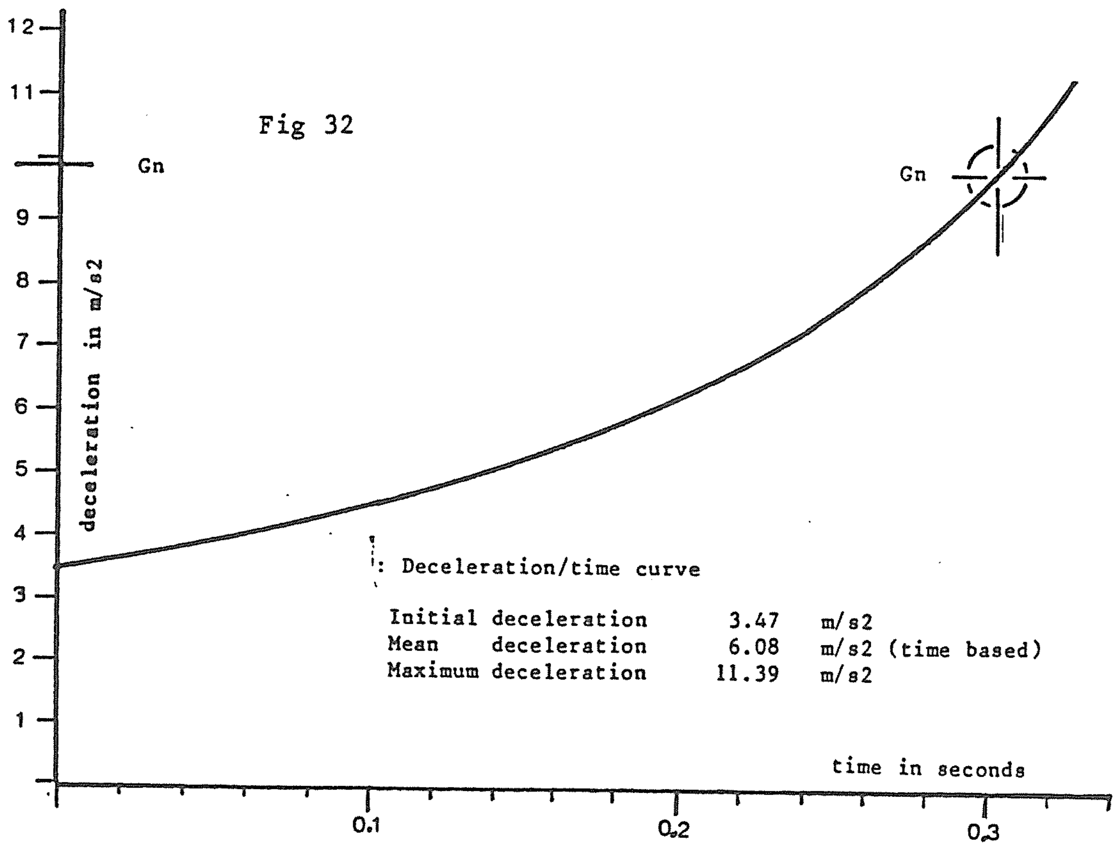


Fig 32
Gn
Gn
: Deceleration/time curve
Initial deceleration 3.47 m/s2
Mean deceleration 6.08 m/s2 (time based)
Maximum deceleration 11.39 m/s2
deceleration in m/s2
time in seconds
12
11
9
8
7
6
5
4
3
2
1
0.1
0.2
0.3

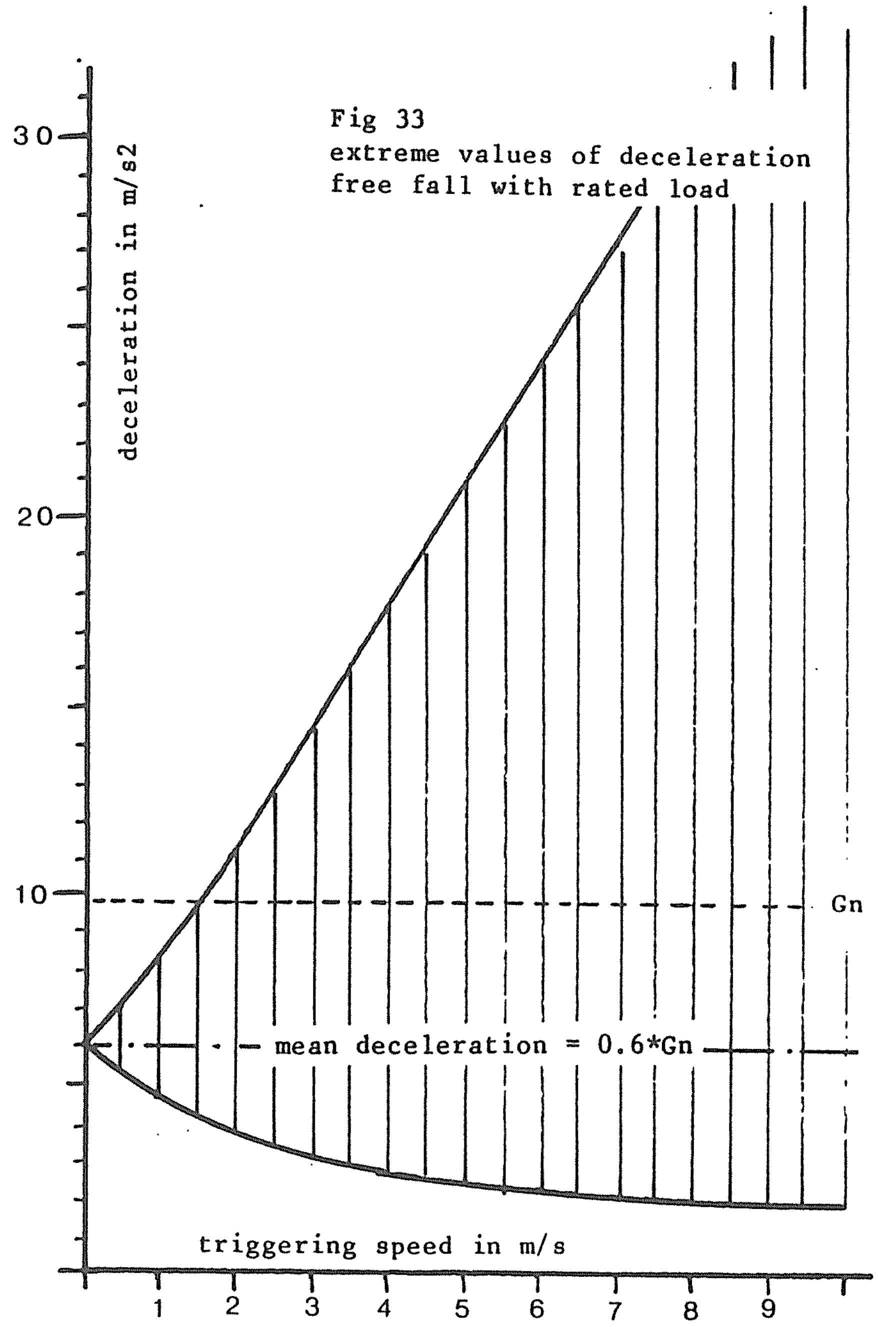


Fig 33
extreme values of deceleration
free fall with rated load

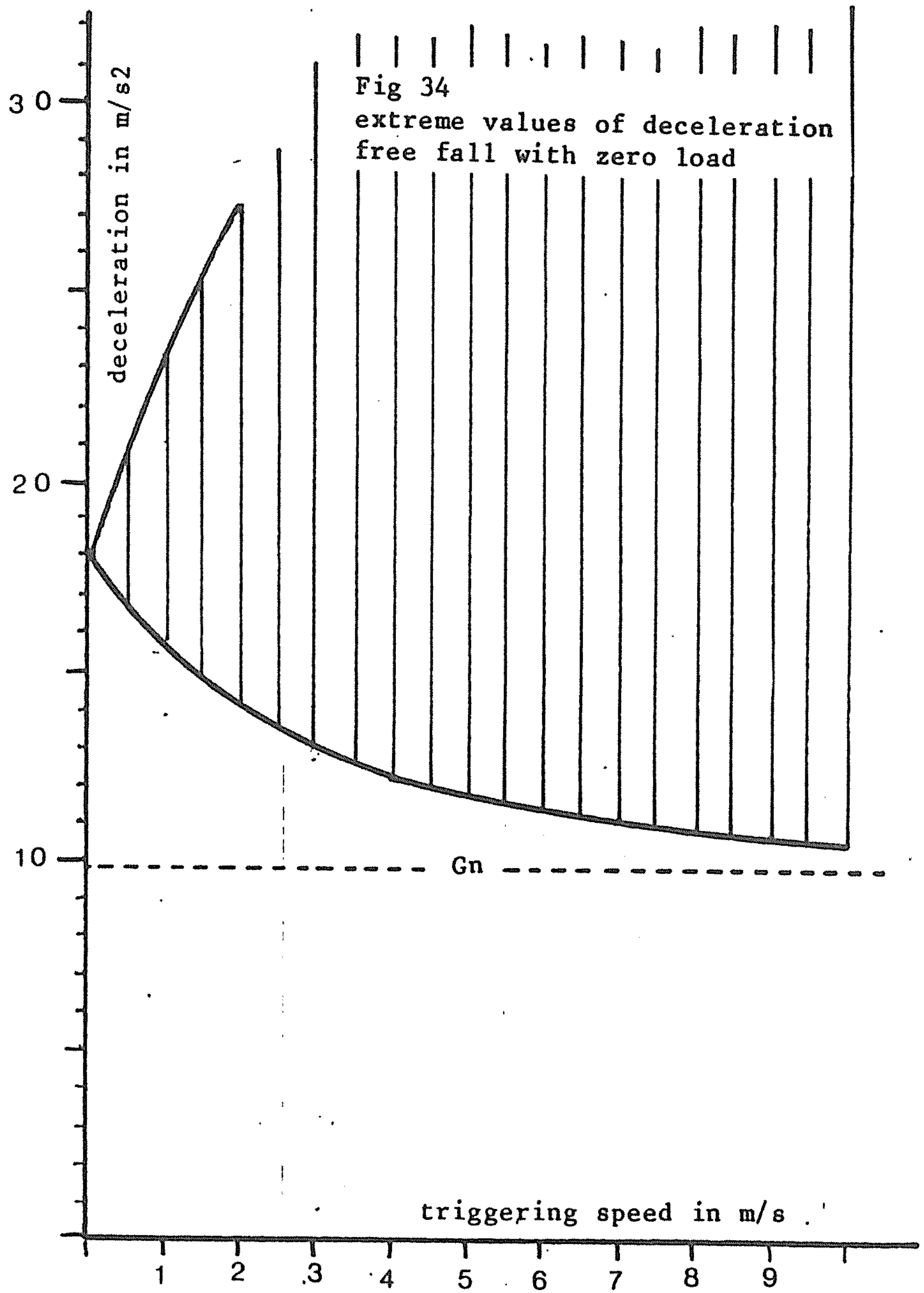


Fig 34
extreme values of deceleration
free fall with zero load

F.3.3/g
So, we can select the initial deceleration for aiming at a mean deceleration of 6 m/s2 (0.6*Gn).
But, for all triggering speeds of 2 m/s and above, instantaneous decelerations exceeding 1*Gn (9.81 m/s2) should be expected when testing the safety gears in free fall with the rated load (see Fig 32 and Fig 33).

This means that the instantaneous braking force will be more than 25% above the average. This is unavoidable with cast iron shoes.

~~It is contrary to the last alinea of EN/F.3.32.3.1 (testing procedure).~~
However, it is objectively not critical for the following reasons:

- the Clause EN/9.8.4 does not mention a limit for the peak deceleration
- the duration of the peak is limited
  At 2 m/s, the instantaneous deceleration exceeds 1*Gn only for a few hundredths of a second.
- in the specifications for the hydraulic buffers, a peak is allowed providing it does not exceed 2.5*Gn for more than 0.04 second.
  In the case of the progressive safety gear, such a specification would be met, even for the highest practical triggering speeds.
- at zero load (empty car), the decelerations will necessarily exceed 1*Gn during the whole time of operation (see Fig 34).
  Consequently, it would seem futile making a fuss because Gn is exceeded during a short time at full load.

Moreover, when the safety sets with counterweight attached, all the deceleration values are brought closer to Gn (either increased if they were lower, or decreased if they were higher). This smoothens the operation of the safety gear (see Fig 35 and 36 page 94).

The calculations are made assuming that there is a locked-down compensator and that the traction sheave does not interfere.
Incidently the same Fig 35 and 36 justify the recommendation for using a locked-down compensator at any speed higher than 2 m/s.
(see HB/9.6.1 and HB/9.6.2/a)

F.3.3/h
If a safety gear tested at a given triggering speed (test speed) has to be used at a lower speed (application speed) something has to be changed in order to still have a mean deceleration of 6 m/s2.

You can either change the force applying the shoe or change the suspended mass. Let us suppose that the adjustment of the force is not changed, i.e that the safety gear is used as tested .

To aim at keeping the mean deceleration around 6 m/s2, the suspended load must be increased by the ratios given Fig 37 (page 95).

Of course, because of the CEN rules, you need to have the gear tested with this new speed and new load. Such a tabulation is not recognized by CEN. It can be useful for orientation purposes only.

Perhaps after a few years of experience, such a tabulation will be recognized by CEN under certain conditions.
The TUV has been using something similar (see Fig 3 of APP:12/HB).

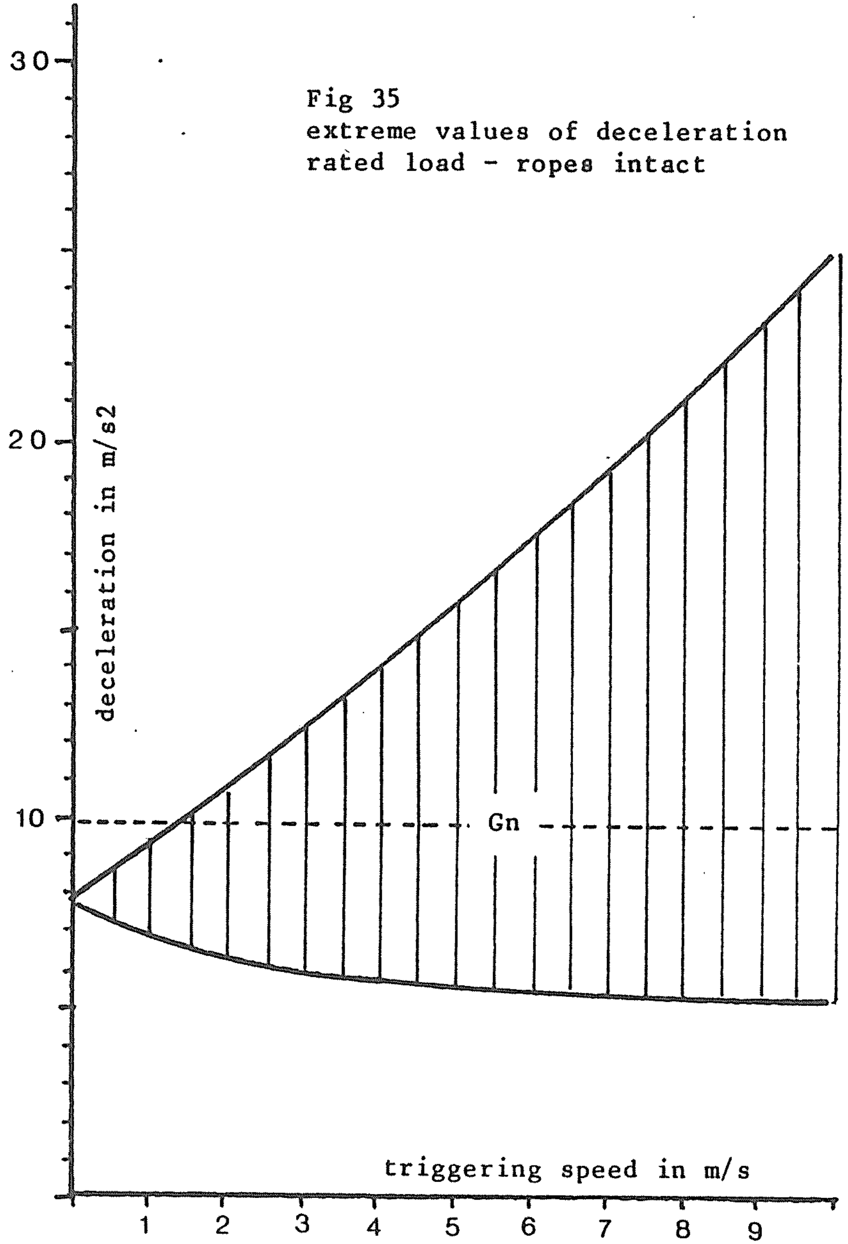


Fig 35
extreme values of deceleration
rated load - ropes intact

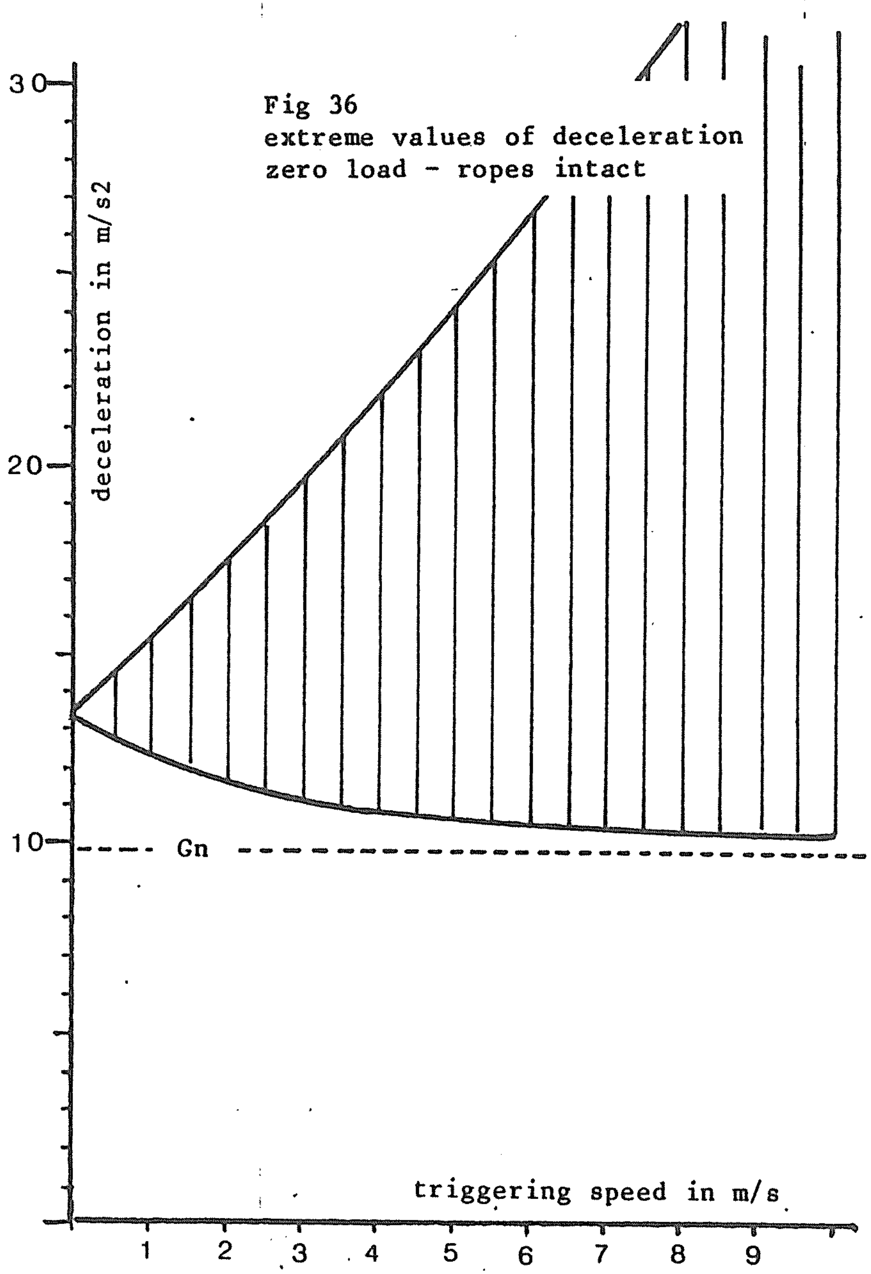


Fig 36
extreme values of deceleration
zero load - ropes intact

Fig 37: Indicative load ratios for using a given safety gear at speeds lower than the one used for testing

Speed at test	Speed of application in m/s 01	02	03	04	05	06	07	08	09
1	1.00								
2	1.19	1.00							
3	1.38	1.16	1.00						
4	1.57	1.31	1.13	1.00					
5	1.75	1.47	1.27	1.12	1.00				
6	1.93	1.62	1.40	1.23	1.10	1.00			
7	2.11	1.77	1.53	1.35	1.21	1.09	1.00		
8	2.29	1.92	1.66	1.46	1.31	1.18	1.08	1.00	
9	2.47	2.07	1.78	1.57	1.41	1.28	1.17	1.08	1.00

F.3.3/i
All the above considerations are meaningless if a better material than cast iron is used for the shoes.

Bronze has a more stable friction factor but the value is very low. Developing enough force for applying the shoes, would require a bulky mechanical arrangment.
Other materials with a friction factor almost independent of the rubbing speed have been developed by the brake manufacturers working for the automotive industry and for the railroads.

The considerations are also meaningless if the force applying the brake shoes is regulated to compensate, at least to a certain extent, the variation of the friction factor.

Of course, if such a design is used, things happen as originally envisaged by the CEN/TC 10/WG 1 but, such sophisticated designs are economically justified for high speeds only.

F.3.4.(b)
Although not required by the testing procedure, it would be advisable to measure or check:

- the minimum force needed to positively engage the safety gear,
- the maximum force which can be accepted by the mechanical liaison between the point where the governor rope is attached and the shoes of the safety gear. A reasonable safety factor should be selected.

F.3.5.2
The certificate should, in my opinion, also report:

- the minimum governor rope pull needed to engage the safety gear,
- the maximum governor rope pull which may be safely applied to the actuating mechanism.

This information is needed for selecting a proper governor.

APPENDIX F.4 to EN 81/I

Directives for the approval of overspeed governors

F.4.1: General provisions. (c) Anticipated value of the tensile force .

F.4.1(c)/a
Although not spelled out in the code, I think it is understood that the tensile force envisaged is the tensile force which would be generated by the overspeed governor if the car were not accelerating at all.

The additional pull needed to accelerate the rope and the rotating masses comes in addition to the force required from the overspeed governor. It constitutes an additional safety margin.

There are 2 reasons for not taking this additional pull into account.

- the safety gear might have to operate in cases other than free fall and, in such cases, the acceleration would be lower than Gn.
- the inertia will depend on the length of the governor rope which changes from lift to lift.

However, it is well to remember that this additional value is not negligible in the case of free fall. For example, let us estimate what it could be in the following case:

- travel	100	m,
- length of rope	210	m,
- diameter of rope	8	mm,
- mass of rope	48.3	kg

All the considerations developed in HB/9.8.3.1 do apply here.
Including the governor sheave and the tensioning pulley in the masses to be accelerated, the inertia would by itself generate a pull of 500 Newtons. It is not taken into consideration for the design of the governor but it should be for the design of the safety gear mechanism.

F.4.1(c)/b
The procedure assumes that a separate test will be conducted for each possible adjustment of the incorporated "braking system".

I think that there are 2 major "braking systems":

a) traction developed by the rope wedged in a narrow V groove.
   In that case, the governor sheave is blocked when the speed measuring device triggers.
b) separate brake applied on the rope.
   In that case the triggering device releases the device applying the brake shoe(s). The groove in the governor sheave is generally semi-circular and the sheave keeps turning, accompanying the rope.

In both cases, the braking is generated by the friction of the rope on another material. If that other material is cast iron, the braking force will vary during the deceleration process because of the variation of the friction factor with the rubbing speed.

In most speed governors, this is the case. This means that the braking force has a minimum and a maximum. The minimum must be above the minimum required by the code. The maximum, increased by the pull coming from the inertia, must be under the maximum accepted by the safety gear actuating mechanism.

For the traction formula (see APP:11/HB and APP:12/HB), we have selected the following formula for expressing the friction factor:

$$\text{Friction factor for ropes} = \frac{0.125}{1 + 0.125*V}$$

I have no reason for proposing anything else in the case of the speed governors except, of course, if a special braking material is selected instead of cast iron. If the material is judiciously selected, the friction factor could be independent of the rubbing speed (ex: linings developed for the brake industry).

In the case of the (a) braking system, the tensile force may be adjusted by changing the mass of the tensioning device in the pit.

In the case of the (b) braking system, the tensile force may be adjusted by changing the force applying the brake shoe(s) on the rope.

You will find, in the Appendix APP:14/HB to this Handbook, some more considerations on the subject of tensile forces.

F.4.2.2.2.NOTE 3
Because of the low acceleration required by this NOTE, the test cannot give any indication on the response time in case of free fall of the car.

The "response time" is in fact expressed as a "response distance" in the procedure for testing the instantaneous safety gears (EN/F.3.2.4.1) and is also materialised by an overtravel in the operation of the governor.

This "response travel" is made of 2 portions:

- a fixed portion (Fixed Delta Travel), which is the travel of the car from the time the pawl of the measuring device has started engaging a tooth of the ratchet-wheel until the time the braking system of the governor has started to pull the rope.
  This time depends on the design of the governor; for example, in some governors, the wheel must turn a given angle for rearming the spring applying the brake shoe.
- a variable portion (Variable Delta Travel) varying from zero to a maximum which corresponds to the travel from one tooth of the ratchet-wheel to the next which can be caught by one of the pawls of the speed measuring device.
  A pawl might just be in position to catch a tooth when the adjusted speed is reached: in that case, VDT is zero.
  At the other extreme, the pawl might barely miss the tip of one tooth when the adjusted speed is reached. In that case the governor sheave keeps turning until this pawl, or another one, is in position to catch a tooth and start the fixed portion described above.

Let us call TDT (Total Delta Travel), the total of FDT and VDT.
During this travel TDT, the speed keeps increasing if the car is accelerating. In the case of free fall of the car, the acceleration is that of gravity, Gn=9.81 m/s2. If the overspeed is due to a defect in the speed control of the car, the acceleration might be much lower and the testing procedure requires that it be as low as possible.

The maximum tolerated by the code for this overtravel TDT is 0.10 m, to be consistant with EN/F.3.2.4.1. This applies only to instantaneous safety gears but it seems logical to require the same limit for progressive safety gears which are used at higher speeds.

Let us assume that the 0.10 m limit applies in all cases.
Let us call Vadj = the adjusted triggering speed
Vmax = the speed reached at the end of the "response travel"
Gn = 9.81 m/s2
Gamma = acceleration in other cases

$$Vmax = \sqrt{(Vadj)square + 2*(0.10)*Gn}$$

which will be the highest possible value in operation.

$$Vmax = \sqrt{(Vadj)square + 2*(0.10)*Gamma}$$

which will be the upper limit for any other possible acceleration.
If the "response travel" is under the 0.10 m, the maximum speed will of course be lower than the Vmax given by the above formula.

However, in the EN testing procedure the acceleration is zero, which means that the Vmax would be equal to Vadj whatever the "response travel". (the differences due to the friction in the measuring device should be very small. If they are not, the governor is not performing well).

In the alternative testing procedures described in APP:15/HB, the test is made with an acceleration. Providing you calculate the value of the acceleration, you may consider that, if the highest of the recorded triggering speeds (amongst the 20 tests) is lower than the Vmax calculated with the formula, the "response travel" is acceptable.

The tabulation of Fig 38 illustrates dramatically how important it is to limit the "response distance" for the low and moderate rated speeds.
For a rated speed of 1 m/s, the code recommends selecting a triggering speed close to the upper limit (1.5 m/s).
If the "response distance" is 0.10 m, the speed reached before actuating the safety gear will be, in the case of free fall, 2.05 m/s.

Fig 38 Speed reached after the "response distance" in the case of free fall

Vrated = rated speed
V1 = minimum triggering speed as per EN/9.9.1
V2 = maximum triggering speed as per EN/9.9.1

V5 = speed corresponding to V1 after a 0.10 m response distance
V6 = speed corresponding to V2 after a 0.10 m response distance

Vrated	V1	V2	V5	V6
1	1.15	1.50	1.81	2.05
2	2.30	2.63	2.69	2.98
3	3.45	3.83	3.72	4.08
4	4.60	5.06	4.81	5.25
5	5.75	6.30	5.92	6.45
6	6.90	7.54	7.04	7.67
7	8.05	8.79	8.17	8.90
8	9.20	10.03	9.31	10.13
9	10.35	11.28	10.44	11.36
10	11.50	12.53	11.58	12.60

Although nothing is required by the code in the case of progressive safety gears, it would be wise to limit the response distance to 0.10 m.
Let us remember that the test certificates for progressive safety gears states the maximum speed of engagement.

Contrary to what most people would expect, this "response distance" is not as critical for high rated speeds.

The laboratory should at least check, by reviewing the drawings, if the geometry of the mechanism is such that the car will normally not travel more than 0.10 m after reaching the triggering speed.

It may also be tested by turning, by hand, the governor wheel until one of the hooks of the speed measuring device is just in the right position for catching one of the teeth of the ratchet-wheel.
Make a mark on the rope and note the position.
Keep turning the governor wheel until reaching the next possibility of catching one of the teeth of the ratchet-wheel, engage the hook in the ratchet-wheel and keep turning until the "braking system" starts acting on the governor rope. Note the new position of the mark on the rope.
The travel of the mark represents the "response travel" of the governor.

(see also, in this Handbook, the considerations reference HB/9.9.7)

F.4.3.2 (f)
The certificate must state the minimum tensile force when the rope is sliding but the procedure does not indicate how it shall be measured.

If it is measured statically, by holding the governor in the "braking position" and hanging gradually increasing weights until the rope starts slipping, the measured force should be multiplied:

a) in the case of a separate "braking system", by the ratio of the friction factor at the triggering speed to the friction factor at zero slipping speed, see HB/F.4.1(c)/b above.

b) in the case of a cast iron traction pulley,

$$\text{by the ratio} = \frac{e^{C3*3.14*Mu(\text{at gliding speed})}}{e^{C3*3.14*0.125}}$$

The value of C3 can be found in HB/APP:05. For an angle Gamma 30°, it is 3.86 in a new groove.
The friction factor formula is the same as above.

F.4.3.2 (g)
Although not required by the procedure, the laboratory should (in my opinion again) make a remark if, after examination of the drawings or after the simple test proposed above, it had found out that the geometry was such that the car could travel more than 0.10 m after detection of the overspeed. (see above).

The laboratory should also note if the tensioning pulley is locked-down.
For applications with gearless machines, this pulley should be locked-down each time a locked-down compensator is required.

F.4.4 : Alternative testing method.
This method is described in the Appendix APP:15/HB to this Handbook.
It is based on the principle of the Attwood machine and has the following advantages over the procedure of the code:

- it gives the values of the braking force with the rope sliding as in a normal safety setting.
- it gives an indication on the response time of the governor

It may be used on the site for the acceptance test (see EN/D.2.(i)) or for the periodic tests (see EN/E.1 and 2)

It may also be used by the lift contractor for orientation purposes.

APPENDIX F.5 to EN 81/I

Directives for the approval of energy dissipation buffers

F.5.3.1
Energy accumulation type buffers with buffered return.

See the comments in this Handbook under the reference HB/10.3.4.

F.5.3.2
Test of energy dissipation buffers.

Read the comments made earlier in this Handbook (ref: HB/10.4.3)

The Laboratory should remember:

- that the most important requirement relates to the peak retardation of 2.5*Gn which may not be exceeded for more than 0.04 sec

- that the mean deceleration should be calculated on the time base,

- that a mean deceleration slightly exceeding 1 Gn does not mean that the buffer is bad, although contrary to the requirements,

- that if the mean deceleration is too low (for ex. below 0.4*Gn), the buffer is not excellent, even if acceptable for the code.

APPENDIX G to EN 81/I

Recommendations for fire protection

Although this appendix of the EN/81 did not have much success, some of the considerations developed are worth being recalled here.

The operation of a lift becomes uncertain if the temperature exceeds:

(a) 40° C in rooms where control cubicles are located,
(b) 70° C on the outer face of the landing doors or in rooms where machines or pulleys are located.

This applies as well to the firemen's lift and specifications requiring that the Firemen's Lift be able to rush through the fire like a salamander should be objected to. The operation is also problematic if the landing doors are sprinkled with water.
The protection of the Firemen should come from a judicious arrangment of the building and not from mythical characteristics of the lift.
Also, the operation of the Firemen's Lift should be always the same to avoid errors and hesitations in case of emergency.

The inter-connexions of the lifts with the fire detection devices and with the emergency power system (if any) should be standardized.

COMMERCIAL ASPECTS of the CODE IMPLEMENTATION

Most of the requirements of the code relate to the lift equipment itself and meeting them is the responsibility of the lift contractor.

Some other requirements clearly involve other trades, such as the building contractor for making the walls, the floors etc..for the well, the machine room and the pulley room.

But some requirements can either be fulfilled by the lift contractor or by another contractor, as, for example, the lighting of the machine room and of the well, the iron work for handling the equipment etc..

For simple lift installations, there is, in a given country, a generally accepted consensus on the division of responsibilities.
However the consensus is not the same in each country.
For important lift installations, there are generally detailed specifications which clearly divide the responsibilities.

In order to avoid disputes and misunderstandings, the "Institut Belge de Normalisation: IBN" has issued a document proposing a division of the work between the various contractors. This document is not always enforced, but it has the merit of calling the attention to the possible areas of discussion. It is overruled by any bilateral agreement.
A translation of this document "NBN E 52-017" is given in the appendix reference APP:16/HB. All references to other Belgian standards have been removed. This translation is made only to list the possible areas of discussion. It should be specially useful when dealing with foreign countries where the local usage might be different.

In any event, the lift contractor is responsible for advising the other trades involved of the requirements of the lift code.

APPENDICES

TO THE EN 81 HANDBOOK

A. LEENDERS

1986

1 APP 01 to APP 08

LIST OF APPENDICES

to the HANDBOOK and COMMENTS on the EN 81/Part I SAFETY CODE
by Andre Leenders

The 101 pages of the HANDBOOK are bound separately

Andre Leenders - 234/B3, Avenue de la Lanterne, F-06200 Nice

1986

INTERNAL RULES OF CEN/TC 10/WG 1

when acting as

INTERPRETATION COMMITTEE

I am giving below excerpts of the Doc N 59 E of CEN/TC 10 dated April 1979. They have been selected with the view of laying down:

- what can be expected from the Interpretation Committee,
- how to get an interpretation or an opinion.

The details about the internal organization have been omitted.

---

1 - TASK OF THE WG 1

When acting within the mandate as defined above WG 1 has to :

a) give an interpretation of a clause or a series of clauses of the standard,

b) give a comment on the standard on subjects where it seems to be useful, for a better understanding of the text,

- to summon back the basis of thoughts or calculations that led to a given requirement,
- to make the text clearer if experience has shown that it is not clear enough.If necessary examples can be given ; these examples should not prejudge other possible designs.

c) to express its opinion on requests for deviation,

d) to establish a list of the items which have to be reconsidered during the next revision of the standard.

---

2 - "INTERPRETATIONS" AND "COMMENTS" PROCEDURE

2.1 Introduction of "interpretation" or "comments" requests

2.1.1 The only requests which will be considered are those introduced by

a) The national committees of CEN member countries ;
b) the CEN Central Secretariat,
c) FEM - CIRA - CENELEC.

The national committees will study requests sent in by their own countries and transmit them if they consider it necessary

The Central Secretariat of CEN will do the same with requests sent in by E.E.C or from national committees of contries which are not members of CEN.

2.1.2 A copy of requests which have not been transmitted by a member committee to the WG 1, including those dealing with clauses allowing national deviations, will be communicated to the secretariat of the WG 1 with a copy of the answers made by the national committee.

2.3 Publication

The interpretations or comments are transmitted by the secretariat of WG 1 to the secretariat of TC 10 which then sends them to :

a) - national committees of CEN members who will be responsible for distribution and publicity in their own countries.

b) - Central Secretariat of CEN who may transmit them to anyone of their choice.

c) - E.E.C.

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3 - PROCEDURE FOR OPINION ON A REQUEST FOR A DEVIATION

3.1 General

The WG 1 is not entitled to permit deviations. It can give its opinion to the authority entitled to do so either on a national level in one of the 15 member countries of CEN or on the level of E.E.C, when this authority has asked for it.

3.2 Introduction of requests for opinion

The only requests which will be considered are those introduced by :

a) national committees when national authorities are concerned.
b) the CEN central secretariat when it is E.E.C.

3.3 Giving an opinion

An opinion on a request for a deviation shall not be established by correspondance and shall be examined during a WG 1 meeting. The procedure is that given in 2.2.2

3.4 Publication of opinions

The opinion will be sent via the secretariat of TC 10 to the questioner with copy to :

a) national committees of other CEN members
b) CEN Central Secretariat
c) E.E.C via CEN Central Secretariat.

NOISE REDUCTION IN BUILDINGS

Hints for architects

APP:02/1 Most important factors in the building design

Apparently, the most important factors to be taken into consideration when designing a building are:

- location of the critical rooms away from the machine rooms and lift wells whenever possible,
- making floors, ceilings and walls heavy enough,
- avoiding carefully all solid "noise bridges" across insulating spaces whether filled with insulating material or not.

We will call "critical rooms", the rooms such as bedrooms and living rooms where the noise is especially objectionable.

APP:02/2 Floors and ceilings

In all the examples given below, the machine room floors are supposed to have a thickness of at least 0.25m of concrete.

All other floors, and all the ceilings, are supposed to have a thickness of at least 0.20m of concrete.

APP:02/3 Walls

Four different types of walls are considered in these examples:

LW	light weight	less than			350kg/m2
NW	normal weight	between	350kg/m2	and	450kg/m2
HW	heavy weight	between	450kg/m2	and	550kg/m2
EHW	extra-heavy weight	more than	550kg/m2.		

APP:02/4 Examples of layouts suitable for appartment buildings

Although it is always better to have the critical rooms away from the lifts, their location in relation with the machine room and the shaft depends on the type of walls.

In the following illustrations, "GOOD" means that the location is acceptable for one of the critical rooms whereas "BAD" means that they should never be located there.

APP:02/4/a: Layouts to be avoided

On the right side of Fig 1, is shown the most usual arrangment: the walls are all of the same normal thickness.
Locating critical rooms adjacent to the lift should be avoided.

On the left side of Fig 1, you can see that, if the dividing walls of the appartment are too light, even the rooms next to the adjacent ones should not be used as critical rooms.

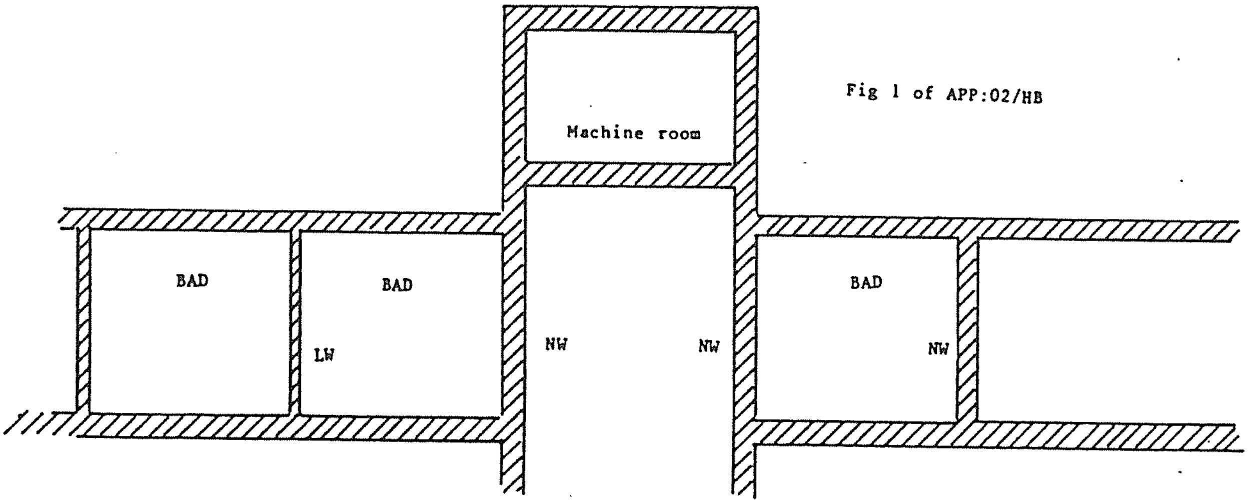


Fig 1 of APP:02/HB

APP:02/4/b: Acceptable layouts

On the left side of Fig 2, is illustrated the usual arrangement with all the walls of normal thickness. In this case, the spaces next to the ones adjacent to the lift may of course be used for critical rooms.

As illustrated on the right side of Fig 2, they might also be used in the case of light dividing walls providing that the walls of the spaces devoted to the lift be extra-heavy walls.

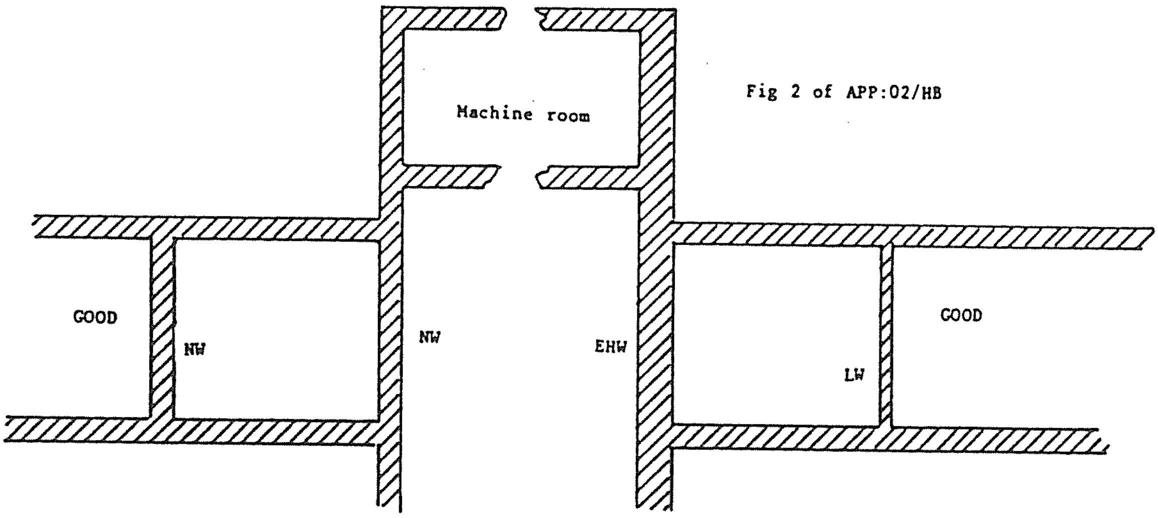


Fig 2 of APP:02/HB

If locating one of the critical rooms adjacent to the lift is unavoidable, the walls of the lift spaces should of course be extra-heavy but, in addition, at least the complete floor of the machine room should be carefully sound-insulated from the rest of the building (e.g. by cork).

This insulation does not replace the regular elastic supports of the machinery. The natural frequencies of the insulation and of the elastic supports must be different.

This case is illustrated Fig 3, the right side of the picture being an alternate of the left side.

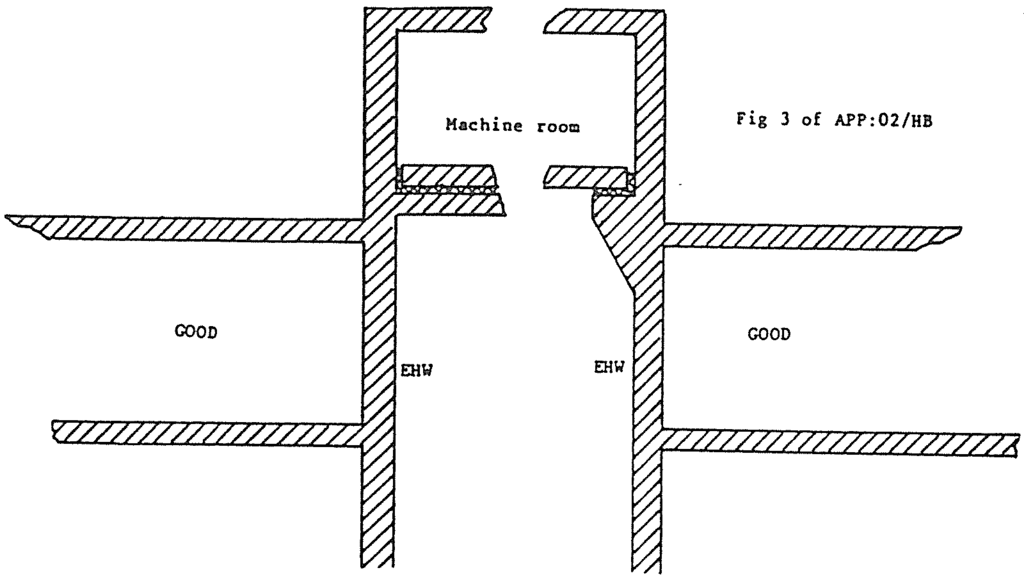


Fig 3 of APP:02/HB

However, if having critical rooms adjacent to the lift(s) is unavoidable, the best solution is to install the lift(s) in a core completely insulated from the building as illustrated Fig 4.
Normal walls may be used. The space between the lift walls and the other walls need not be filled by any sound-insulating material but, where closing the gaps is necessary, sound insulating material must be used. This is illustrated Fig 4.

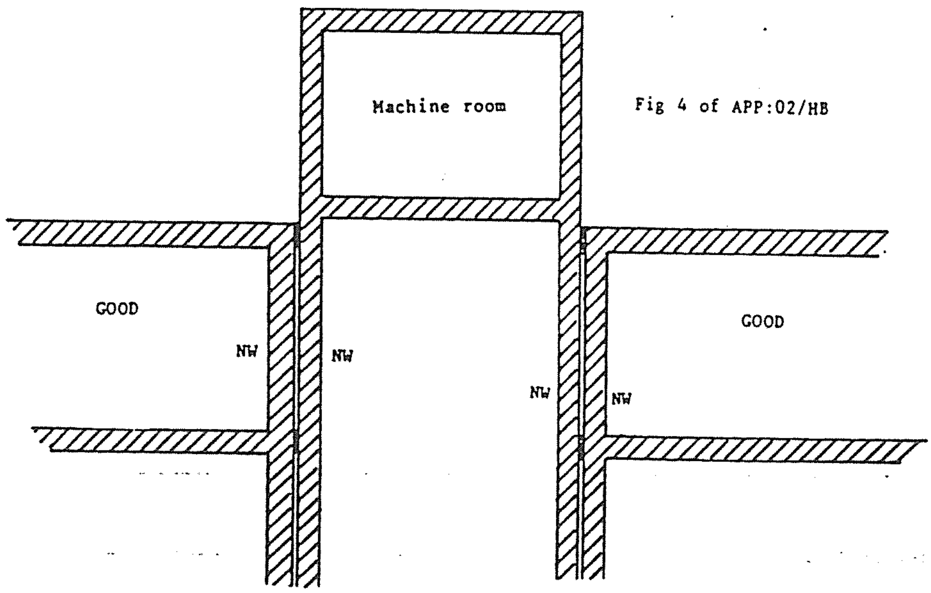


Fig 4 of APP:02/HB

APP:02/4/c: Case of the machine room not located at the top.

From the point of view of lift technique, this is definitely not recommended and not even envisaged in the ISO 4190 Standard.

If this is nevertheless planned:

- the walls of the lift spaces should always be of the heavy type,
- all other walls should be at least of the normal type,
- no critical rooms should ever be adjacent to the lift whether laterally, above or below.

APP:02/5 Higher class sound-proofing

In hotels, hospital, etc, the combination of several of the precautions illustrated above may be necessary. For example, use the construction illustrated Fig 4 and locate no critical rooms adjacent to the lift.

APP:02/6 Protection against airborne noises

All the above refers to the avoiding of the transmission of vibrations and noises through solids (walls, floor etc..)

Precautions should also be taken to dampen airborne noises: wall coverings, doors, etc..

BEHAVIOUR OF THE LIFT SYSTEM

DURING OPERATION OF THE SAFETY GEAR OR OF THE BUFFERS

APP:03/1
The lift system is illustrated in Fig a and the masses to be considered are identified with their respective symbols in accordance with the ones proposed in EN/4.2 and the additional ones proposed in HB/4.2.

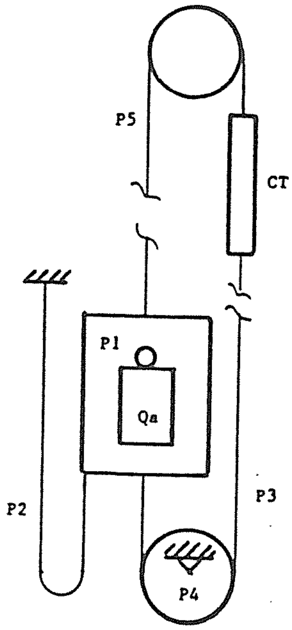


Fig a

Masses in kg:

Qa	actual load in the car
CT	counterweight
P1	empty car including sling, door, etc..
P2	travelling cable(s)
P3	compensating ropes (or chains)
P4	compensator
P5	hoisting ropes

x is the fraction of travel (H), counted from bottom to top identifying the position of the car in the well.

APP:03/2
When either the safety gear or the buffer is operating, an emergency stop is automatically initiated, unless some of the safety switches have been purposely short-circuited (ex: ANSI acceptance test).

For not overcomplicating the formulae:

- we will generally consider that, if there is a compensator, it is locked-down,
- we will disregard the elasticity of the ropes,
- we will generally disregard the kinetic energy coming from the rotation of pulleys,
- we will disregard the kinetic energy of the overspeed governor with its rope and tensioning pulley,
- we will disregard the frictions in the well.

As illustrated in Fig b, the lift system can then be seen as a space craft consisting of 2 masses connected by a liaison and having an initial speed with the corresponding kinetic energy. If there were no forces applied to the system it would continue travelling at constant speed.

But the following forces have to be considered:

- the force resulting from the action of the gravity on the car and everything inside or attached to it (including ropes). This force tends to accelerate the car.

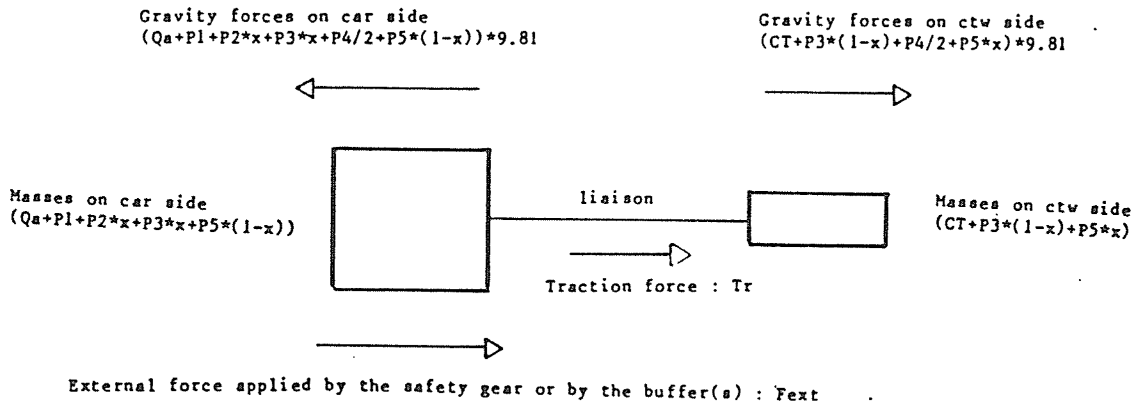
- the force resulting from the action of the gravity on the counterweight and everything attached to it (including ropes). This force tends to decelerate the car

- the external force resulting from the action of the safety gear or of the buffer(s). This force tends to decelerate the car.

- the force resulting from the friction of the hoisting ropes in the traction sheave if the rope speed differs from the tangential speed of the sheave and if there is any pull in the hoisting ropes. This force can act either way.

- the forces resulting from the action of the gravity on the compensator. These forces are equal on the car side and on the counterweight side and balance each other. They influence only the calculation of the pull in the hoisting ropes and, as a result, the calculation of the traction force.

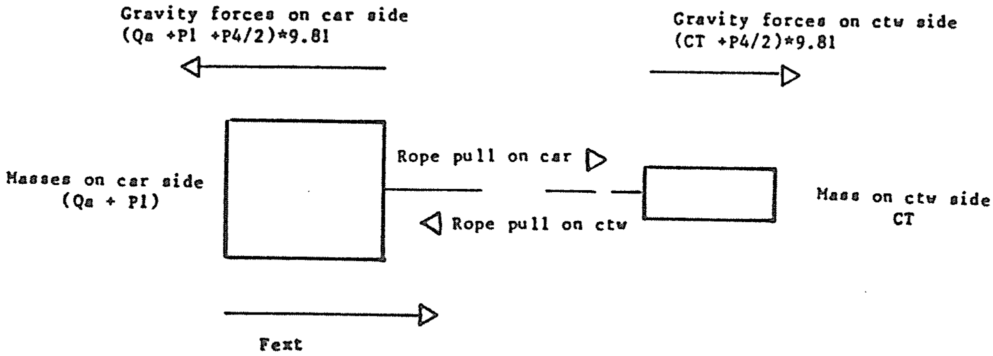
Fig b : Forces acting on the masses constituting the lift system



You will note that the mass of the compensator is not included in the travelling masses.

APP:03/3
For all the calculations I made for this Handbook, I used the following additional simplification: I considered that P2, P3 and P5 were equal to zero on the grounds that these values were generally small by comparison with P, Q and CT and that it was meaningless to try to have very accurate data on one hand whilst, on the other hand, many simplifying assumptions were introduced.(See Fig c)

Fig c : Simplification of the Fig b



This is certainly good enough for orientation.

For a better approximation, you may introduced the real values with x=0.5 (which means that the car is supposed to be at mid-travel).
Only for the extremely high rises will you need to consider various values for x (from 0 to 1).

Multiple roping can be assimilated to single roping by considering that the total mass of the ropes moves at the speed of the car.
(see HB/Note 1 to Clause 9 a.2)
The translation inertia of the reeving pulleys is accounted for in the masses of the car and counterweight. If you want to take the rotational inertia into account as well, refer to HB/Note 1 to Clause 9 a.2.

The external force, (Fext), will be a function of:
- the position of the car in the case of spring buffers,
- the position and the instantaneous speed of the car in the case of hydraulic buffers,
- the instantaneous speed of the car in the case of progressive safety gears.

Some formulae will be proposed in the relevant chapters of this Handbook but they cannot apply to all possible designs.
You will have to develop the formulae suitable for your specific design following the same principles.

APP:03/4
For analyzing the phenomenon, we must use the basic formula:

Force = Mass * Acceleration

Let us call:
"Gamma" the instantaneous acceleration
TR the limit traction ratio resulting from the traction calculations, (generally between 1.7 and 2)

Let us consider that:
- the speed is positive when the car goes down
- the acceleration is positive when the speed increases
- a force is positive when it pulls the car down

We will of course use the kg, m, m/s, m/s2 and Newtons as units.
9.81 is the acceleration of gravity (m/s2)

I think the easiest approach is to apply the basic formula independently to the car and to the counterweight, adding a third equation expressing the rope pull on the ctw side in relation to the rope pull on the car side and remembering of course that the car and the counterweight have the same acceleration (whatever the conditions with the locked-down compensator)

(Qa + P1 + P4/2)*9.81 - Fext - (Pull car side) = (Qa + P1)*(Gamma)

(Pull ctw side) - (CT + P4/2)*9.81 = CT*(Gamma)

(Pull ctw side) = (Pull car side)* (Factor)

with Factor = TR if the rope speed exceeds the sheave speed
= 1/TR if the sheave speed exceeds the rope speed
= 1 if the rope speed is equal to the sheave speed
= 1 if the rope pull is zero or negative

These conditions have to be introduced into your computer program.

To know the sheave speed, you must of course select a value for the acceleration of the sheave based on the type of machine, on the brake and on the characteristics of the electrical control.

The figure will be negative and can vary between:

(-1) if you have a geared machine with a heavy flywheel and a smooth mechanical brake

(-10) or (-30) if you want to figure a very hard braking due to the simultaneous action of the mechanical brake and of an emergency electrical brake (ex: gearless machine with immediate application of the mechanical brake and short circuiting of the armature).

(-1) or (-1.25) if, even in the case of an emergency stop, the normal deceleration is maintained by the electrical control and the mechanical brake is applied only after full stop.

If you want to assume that the machine has no influence just set TR=1.

If you want to consider the free fall, set CT=0 and P4=0.
This, because the suspension has failed and as soon as the speed of the car downwards exceeds the speed of the counterweight upwards, the compensator will sit on the pit floor and all liaison between car and counterweight disappears. It could be different in exceptional cases where the failure would occur at high speed down (the counterweight moving upwards at the same high speed) and the safety gear would immediately impose a deceleration of more than 1 Gn (empty car). Suspension failures are more likely to happen during loading, when starting or when braking.

APP:03/5 - Analysis of the forces developed in the liaison -

The liaison consists of the hoisting ropes and of the compensating ropes.

The formulae given in APP:3/4 allow you to calculate the "Rope pull car side" and the "Rope pull counterweight side".

If they are positive, the "Compensator rope pull" is equal to (P4/2)*9.81.

They will reach zero toghether and at this moment the "Compensator rope pull" will still be equal to (P4/2)*9.81.

But as soon as the calculations produce a negative figure for the pull in the hoisting ropes (Rope pull):
- the traction sheave has no effect anymore
(Factor = 1 in the formulae as already stated above),
- the actual pull in the hoisting ropes remains equal to zero both on the car side and on the counterweight side,
- the actual pull in the compensator ropes is equal to:
(P4/2)*9.81 + (absolute value of "Rope pull")

To illustrate this let us first assume that we can, in the formulae, use:
- TR = 1 which means that the driving machine is neutral
- P4 = 0 which means that the mass of the compensator is negligible (aluminum wheel, no added masses) but nevertheless locked-down.

The variation of the traction forces in the hoisting ropes and in the compensating ropes is illustrated Fig d.

Let us now load the compensator. What happens is illustrated in Fig e.
As long as the deceleration is lower than 1 Gn, the traction is increased by (P4/2)*9.81 both in the hoisting ropes and in the compensating ropes.
The moment at wich the traction in the hoisting ropes is reduced to zero is retarded, but after that all the values are identical to the ones of Fig d.

Fig d : Traction on ropes as a function of the car acceleration with a weightless locked-down compensator

mass of rated load	Q
mass of ctw (CT)	1.8*Q
traction factor	1
mass of compensator (P4)	Q

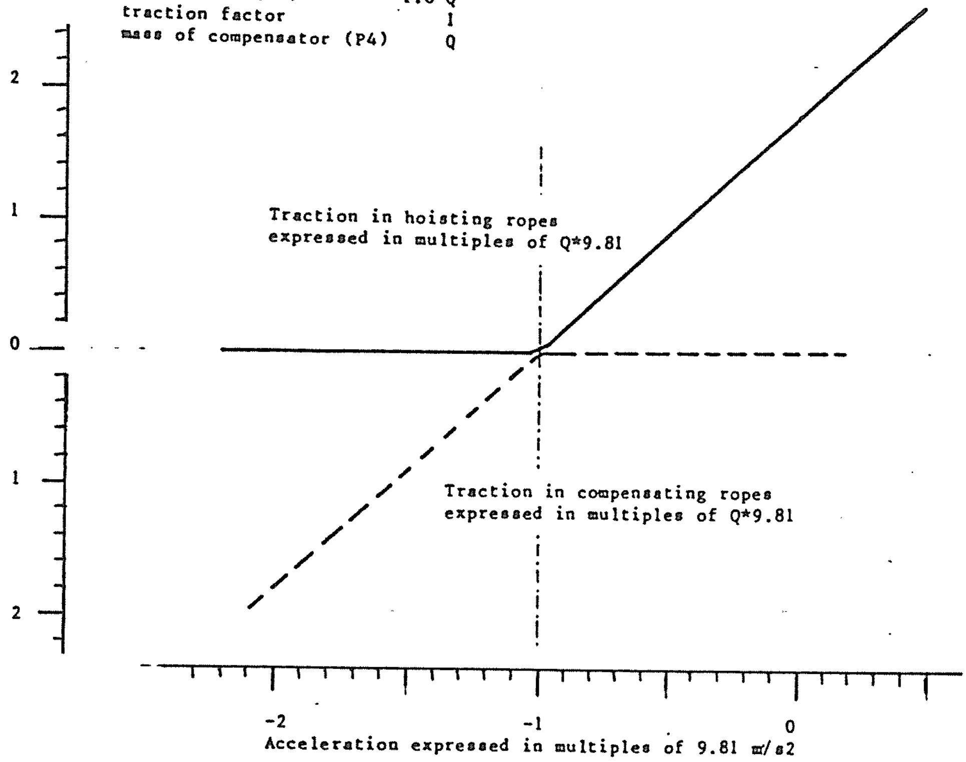
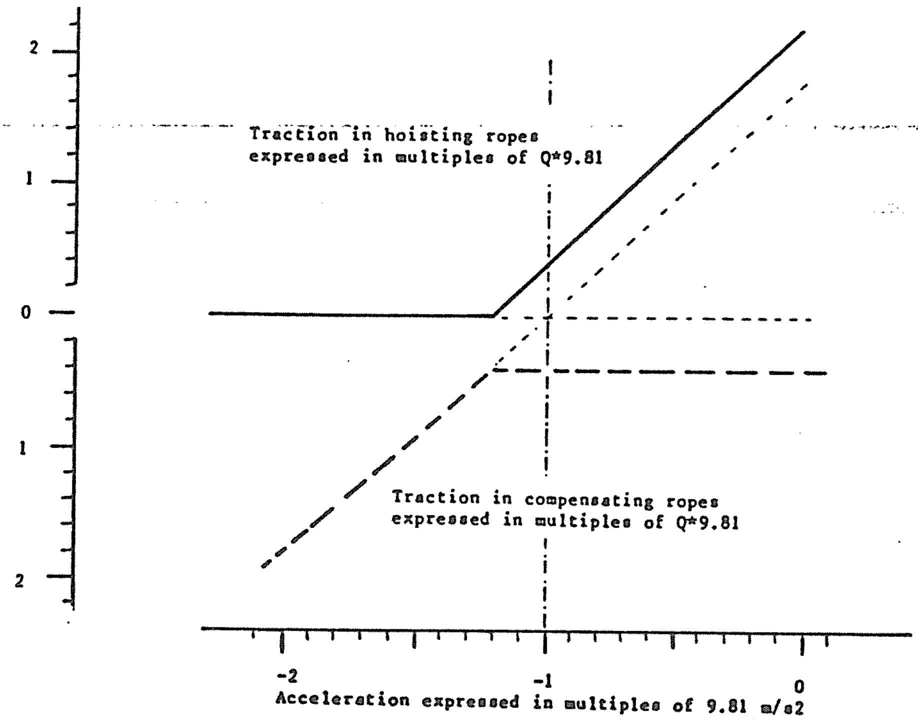


Fig e : Traction on ropes as a function of the car acceleration with a normal locked-down compensator

mass of rated load	Q
mass of ctw (CT)	1.8*Q
traction factor	1
mass of compensator (P4)	0.8*Q



APP:03/6 - Case of the compensator which is not locked down -

The behaviour of the system is exactly the same as long as the pull in the hoisting ropes remains positive.

Then, in your program, you should introduce the condition that this pull remains zero as it reaches zero and consider that, from then on, the car and the counterweight are not linked together anymore.

You can evaluate the jump of the counterweight (slack in the hoisting ropes) by making the difference between the travel of the car and the travel of the counterweight.

Considering that the compensator will move upwards by half of the difference of travel of the car versus the counterweight, you can evaluate the expected jump of the compensator.

I used the word "evaluate" because to be exact the pull in the compensator ropes should be increased by what it takes to accelerate the mass upwards but I do not believe that this refinement is worth the trouble.

APP03/7
- Case where there is no compensator -

Set P4 = 0, then proceed as per APP:03/6

SPECIAL REQUIREMENTS FOR THE SAFETY GEARS OF HOSPITAL LIFTS

APP:04/1) It has been said many times that safety gears do more harm than good. But, in the case of hospital lifts, it becomes more critical.

Indeed, although the gear characteristics are selected for the possibility of free fall with full load, the probability of a suspension failure is practically zero (it is zero as far as I am concerned!).

On the contrary, the probability of an inopportune safety setting with ropes intact and partial load in the car is not negligible.

For analyzing the behaviour of various arrangments, I will take the example of the first hospital lift listed in the standard ISO 4190/1-1980:

Rated load	1600 kg
Mass of empty car (with all accessories)	2000 kg
Mass of counterweight	2800 kg
Rated speed	1.00 m/sec
Overspeed contact adjusted at	1.15 m/sec
Governor triggering at	1.35 m/sec

APP:04/2) Progressive safety gears

It has, so far, been considered as good practice to use progressive safety gears for hospital lifts but let us see what happens in the case of a safety setting in various conditions.

The EN/81 requirements are for the free fall with rated load.

It is certain that the friction factor on which is based the operation of a progressive safety gear is, contrary to the assumption made in the ANSI code and implicitly accepted in the EN/81, varying with the rubbing speed when cast-iron brake shoes are used (which is generally the case).

The analysis of informations gathered from various sources led me to propose the following formula for relating the friction factor (cast iron on steel) to the figure expressing the rubbing speed when the units are m/s

$$\mu = Kp*0.4/(1+0.3*V)$$

where Kp is a factor depending on the unit pressure of application of the brake shoes. (Note: this is a better approximation than the one I used in my "Considerations" for ISO/TC 178)

For the mass allowed by the Type Test Certificate, the mean deceleration would be 0.6*(gravity) but, as stated in EN/F.3.4.(a) the actual suspended mass may deviate from the test value by 7.5%, plus or minus.
The Figure 1a shows the instantaneous decelerations which can be expected in the theoretical case of a free fall with rated load in the car.

Let us see now what would happen when the lift is carrying an hospital bed with a wretched body and an attendant supervising the bottles injecting various liquids to keep the patient alive.
In that case, the actual load in the car would be in the order of magnitude of 250 kg and the Figure 1b shows the instantaneous decelerations which can be expected. They would exceed 1 Gn all the time.
The mean deceleration would be between 1.45 and 1.86 Gn.

The maximum deceleration would be between 1.98 and 2.45 Gn.
However, the probability of having a free fall is so low that it can be disregarded whereas the probability of having an untimely safety setting is much greater. The hoisting ropes being intact, the above figures indicate that, if there is no locked-down compensator, the counterweight will jump and, after having experienced these high decelerations, the car will be shaken when the counterweight falls back.

That jump can be avoided by using a tightly-locked-down compensator, so let us see what would happen in that case (knowing it would be worse without this special device).
It is illustrated in Fig 2a and Fig 2b and, again, this is the case with 250 kg load (Fig 2b) which is the critical one.
The action of the counterweight with the locked-down compensator improves the conditions a little but the instantaneous decelerations are still exceeding 1 Gn all the time.
The mean deceleration would be between 1.23 Gn and 1.41 Gn.
The maximum deceleration would be between 1.44 Gn and 1.65 Gn.

This is the best which can be done with a progressive safety gear meeting the specifications of EN-81 when cast iron brake shoes are used.

Even if, for the brake shoes, a material having a constant friction factor was used, the deceleration with a load of only 250 kg in the car would be between 1.17 and 1.34 Gn.

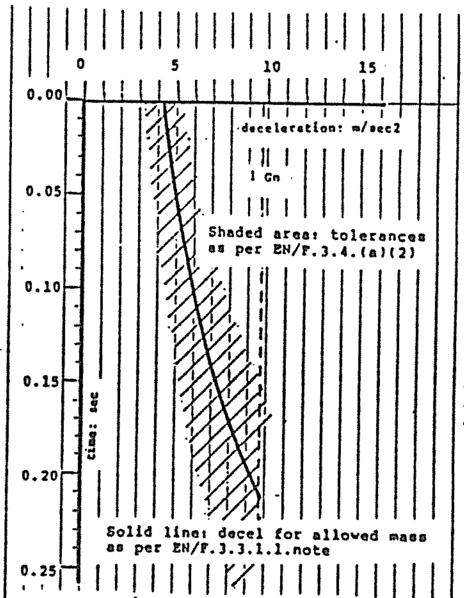


Fig 1a - PROGRESIVE SAFETY
Setting in free fall (1.35 m/sec)
with full load (1600 kg)

(Base of selection for prog. safeties)

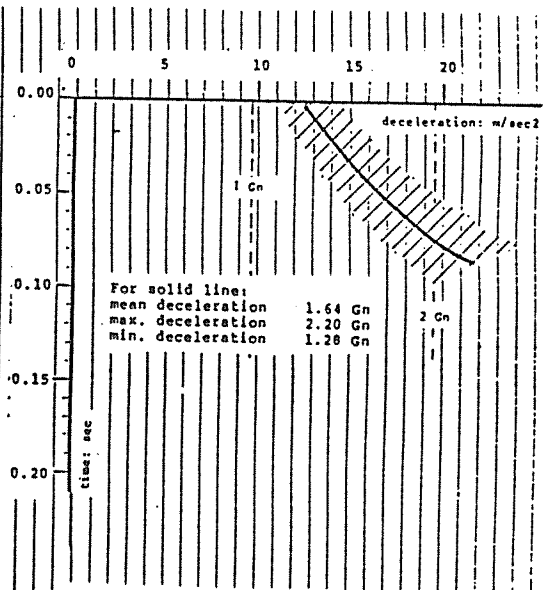
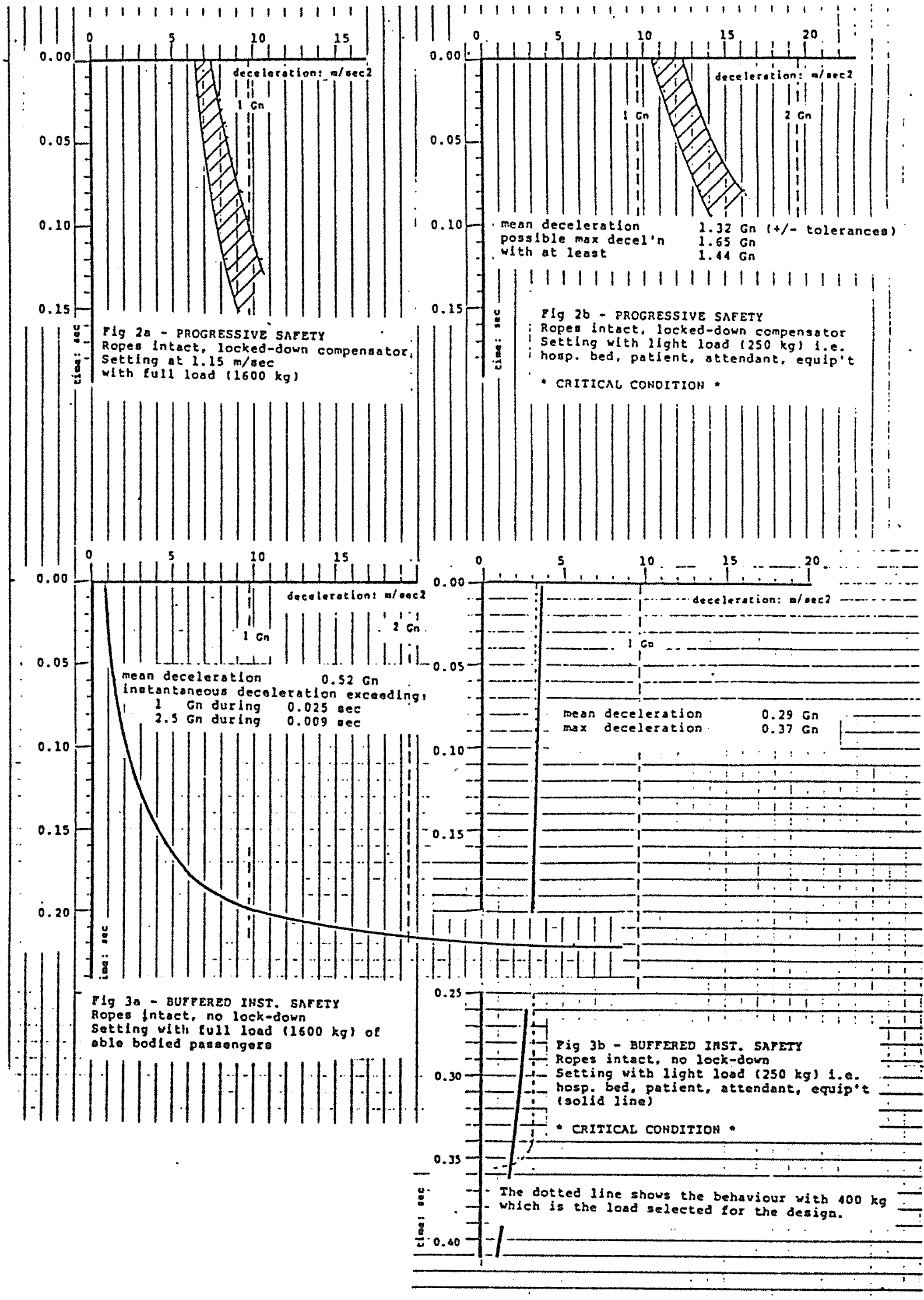


Fig 1b - PROGRESSIVE SAFETY
Setting in free fall (1.35 m/sec)
with light load (250 kg) consisting of:
hospital bed, patient, attendant, equipment


deceleration: m/sec2
1 Gn
time: sec
Fig 2a - PROGRESSIVE SAFETY
Ropes intact, locked-down compensator,
Setting at 1.15 m/sec
with full load (1600 kg)
deceleration: m/sec2
1 Gn
2 Gn
mean deceleration 1.32 Gn (+/- tolerances)
possible max decel'n 1.65 Gn
with at least 1.44 Gn
time: sec
Fig 2b - PROGRESSIVE SAFETY
Ropes intact, locked-down compensator
Setting with light load (250 kg) i.e.
hosp. bed, patient, attendant, equip't
* CRITICAL CONDITION *
deceleration: m/sec2
1 Gn
2 Gn
mean deceleration 0.52 Gn
instantaneous deceleration exceeding:
1 Gn during 0.025 sec
2.5 Gn during 0.009 sec
time: sec
Fig 3a - BUFFERED INST. SAFETY
Ropes intact, no lock-down
Setting with full load (1600 kg) of
able bodied passengers
deceleration: m/sec2
1 Gn
mean deceleration 0.29 Gn
max deceleration 0.37 Gn
Fig 3b - BUFFERED INST. SAFETY
Ropes intact, no lock-down
Setting with light load (250 kg) i.e.
hosp. bed, patient, attendant, equip't
(solid line)
* CRITICAL CONDITION *
The dotted line shows the behaviour with 400 kg
which is the load selected for the design.
time: sec

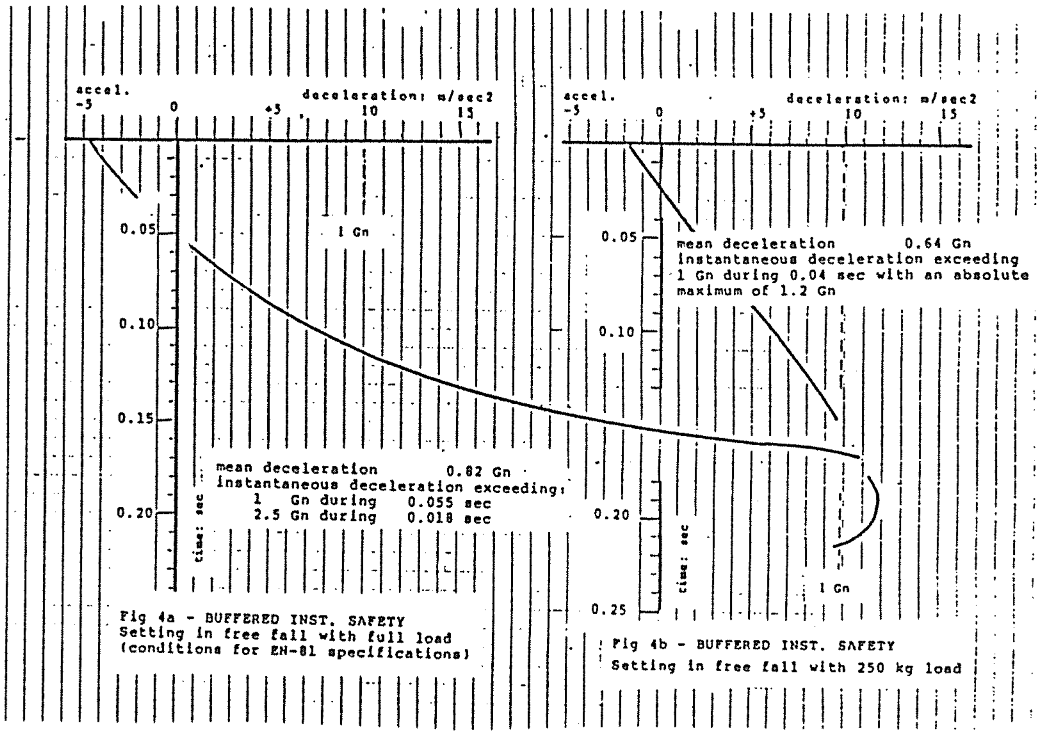


Fig 4a - BUFFERED INST. SAFETY
Setting in free fall with full load
(conditions for EN-81 specifications)

Fig 4b - BUFFERED INST. SAFETY
Setting in free fall with 250 kg load

APP:04/3) <u>Instantaneous safety gears with special buffered effect</u>

On the contrary, the provisions of the code give the possibility of designing that type of safety gear specifically for the critical conditions (250 kg load, hoisting ropes intact) and still meet the requirements for the theoretical case of the free fall under full load.

Indeed, providing all the suspended masses (car and counterweight sides) are known, the hydraulic buffer(s) can be designed for braking the car with 25% of rated load, the hoisting ropes being intact, with a uniform, low deceleration (see, att 1, the approach to the design).

For example, a deceleration of 1/3 of Gn can be selected for this load condition. The stroke has, ofcourse, to be 3 times the gravity distance corresponding to the speed when the operation is initiated.
Since the ropes are intact, the overspeed switch would prevent the speed of exceeding 1.15 m/sec and this is the speed we will consider.

The Fig 3b illustrates the behaviour of the system with 400 kg in the car and what would happen with the critical load of 250 kg: the mean deceleration would be 0.29 Gn and the maximum deceleration 0.37 Gn only.

Needless to say that this is much better than with the progressive safety whether with cast-iron shoes (Fig 2b just on top of 3b) or with an ideal braking material (last sub parag of paragraph 2).

The Fig 3a shows what would happen, still with ropes intact and locked-down compensator, when the car is fully loaded (in that case it would necessarily be regular able-bodied passengers).
The 2.5 Gn are exceeded only a very short time well within the limitations of the code and even 1.0 Gn is exceeded only a small fraction of a second.

The Fig 4a shows what would happen in the hypothetical case of free fall with full load and setting speed of 1.35 m/sec. Those are the conditions required by the code for approval of the safety gear and it can be seen that the deceleration does exceed 2.5 Gn less than the 0.04 sec mentioned in the code and that the mean deceleration is 0.82 Gn.

The Fig 4b shows that, in the also purely hypothetical case of free fall with an hospital bed in the car, the mean deceleration would be of 0.64 Gn with instantaneous values barely exceeding 1 Gn and this during 0.04 sec only, which is also much better than in the case of Fig 1b.

APP:04/4) <u>Conclusions for the application in the CEN area</u>

It is demonstrated that a smooth stop in case of an untimely safety setting can be achieved with this system. Consulting engineers should specify it for hospital lifts and, because of the problems of product liability, lift contractors would be wise to use it even when not required.

The extra cost would be minimal because additional buffer(s) would not be needed in the pit (nothing prevents hanging the buffer(s) from the car and this arrangment could probably save the return spring).
Moreover, a locked-down compensator would be of no avail because the hoisting ropes cannot become slack, except, to a small extent, in the case of safety setting with full load in the car and, in that case, a minijump cannot bother the able-bodied passengers.

In the example (1 m/sec), the "long" stroke would be of only 0.202 m and this would hardly necessitate increasing the pit depth.

Unfortunately, the EN-81 limits the application of instantaneous safety gears with buffered effect to a speed of 1 m/sec maximum.
But I am confident that, when the advantages of the system for hospital lifts have been demonstrated, the national enforcing authority would grant derogations for 1.6 m/sec or even 2.5 m/sec as allowed by the safety codes in the U.S.A. and Canada.

APP:04/5) <u>Possible applications for 1.6 and 2.5 m/sec</u>

As said above this would already be accepted in North America but in those cases the strokes becoming more important, it might be necessary to aim at 1/2 Gn instead of 1/3 Gn but, even so, it would be a big improvement because the higher the speed, the higher the instantaneous decelerations experienced with the progressive safety gears using cast-iron guide shoes.

The Fig 5 illustrates how the minimum and maximum instantaneous decelerations are related to the tripping speed when the suspended mass is exactly equal to the allowable one (i.e mean deceleration = 0.6 Gn in case of free fall with full load).

APP:04/6) <u>Guide lines for the design of ad-hoc buffers</u>

See the Attachment 1 to this Appendix

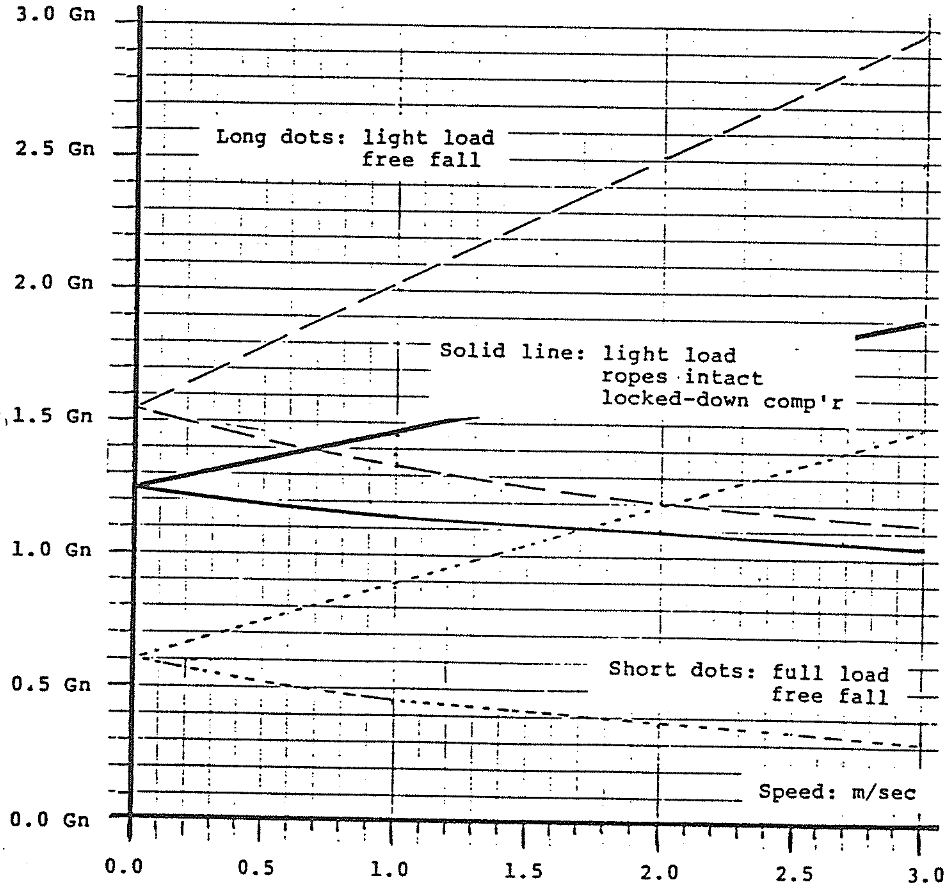


Fig 5 - Min and max instant. decel'ns during progressive safety setting for tripping speeds up to 3 m/sec and various conditions

GUIDE LINES FOR THE DESIGN OF AD-HOC HYDRAULIC BUFFERS

1) Select the maximum allowable pressure in the cylinder

This will depend more on the packing arrangments than anything else.

Example: let us take Max Press = 10,000,000 Newtons/m2 (100 kg/cm2)

2) Select the diameter of the piston

In view of the fact that the deceleration can not exceed 2.5 Gn during more than 0.04 sec, you may consider that the maximum deceleration will not exceed 3 Gn which corresponds to a maximum buffer reaction of:

Bm = 4*Gn*(P1+P2/2+P3/2+Q) Newtons

This is consistant with the reaction of the buffer in the pit as per EN/Clause 5.note 2/b. In the present application half the mass of the travelling cable and of the compensating ropes must be added for taking into account the safety setting in a medium position in the hatch.

The above is correct if there is a single buffer; the result has to be divided by the number of buffers as the case may be.

The section of the piston will then be

Approximate piston section = Bm/(Max Press) in cm2

Having the approximate piston section, you can deduce the approximate diameter then select the final diameter within the available ones. Then, going backwards, calculate the final piston section (Pist sect).

Example: we will use the same values as the ones selected in APPaa/HB and we will neglect P2 and P3.

Bm = 4*(2000+1600)*9.81 = about 144000 Newtons

Approximate piston section = 144000/10000000 = 0.0144 m2

Approximate diameter = Square root of (0.0144*4/3.14) = 0.1354 m

Selected piston diameter: Dp = 0.15 m

Actual section of piston: Sp = 0.01767 m2

3) Select the ratio of the stroke to the gravity distance

Calculate first the gravity distance corresponding to 115% of the rated speed (which is the maximum possible before actuating the overspeed switch)

Evidently, the longer the stroke, the lower the deceleration which can be expected but there are limits:

- there is probably no advantage in selecting a ratio higher than 3 because of the approximations inherent to the mathematical approach.
- the longer the stroke, the longer the buffer and the deeper the pit.

On the other end, the minimum is 1 because it is what the code requires for the buffers at the end of the travel.

Example:
115% of rated V = 1.15 m/sec
The gravity distance corresponding to 115% of rated V = 0.067 m
Let us take the ratio stroke to gravity distance Rs = 3
The stroke will then be Str = 0.202 m

4) <u>Calculate the constant deceleration aimed at</u>

Quite logically, this deceleration which we will call Gc is:

Gc = Gn/Rs

Gc will be between a maximum of 1*Gn and a probable minimum of 1/3 of Gn. Let us remark here that selecting 1*Gn would already be a significant progress because that constant deceleration of 1 Gn would be experienced with the critical load in the car (250 kg).

Obviously, the initial deceleration we have to achieve is equal to Gc.

Example:
The constant deceleration we are aiming at is 9.81/3 = 3.27 m/sec".

5) <u>Calculation of the required braking force</u>

Since we have defined that we want the deceleration to be limited when the safety gear sets with ropes intact and with the critical load in the car (250 kg), the formula to use is (for the case of 1 buffer):

(Buffer force)-(P1+P2+P3+250)*9.81+(CT)*9.81 = (P1+P2+P3+250+CT)*Gc

Note: Gc being <= Gn there will be no slack rope. The locking of the compensator (if any) is not useful.

Example: (Buf force)-(2000+250)*9.81+(2800)*9.81=(2000+250+2800)*9.81/3

(Buf force) = 1133.3*9.81 Newtons
Bf = 11117.7 Newtons

6) <u>Initial section of the hole through which the liquid flows</u>

6.1) The buffer force comes from the pressure building up below the piston. The actual section of the piston being Sp m2, the pressure must reach:

Pr = Bf/Sp in Newtons/m2

This pressure is the one needed to force the flow through the hole.

Example: in our example, Pr = 11117.7/0.01767 = 629,185.06 Newtons/m2

6.2) Once the car hits the buffer, the flow is necessarily linked to the speed of the car and the section of the piston

Flow (m3/sec) = V * Sp (m/sec*m2)

Example: Flow = 1.15 * 0.01767
= 0.0203205 m3/sec

6.3) But, at the same time, it can be calculated by the formula:

Flow = K * Sh * Square Root(2*Pr/UM)

where	Flow is expnesed in	m3
	K is a reduction factor depending on the shape of the hole	-
	Sh is the section of the hole	m2
	Pr is the pressure in the cylinder	Newtons/m2
	UM is the unit mass of the liquid (unit or specific mass)	kg/m3

You will find indications on the values of K in the Note 1.
Those values are only approximations and the formula is assuming that the viscosity of the liquid is close to the one of water.

6.4) Introducing the Flow calculated by (6.2) in the formula of (6.3), you can calculate the initial section of the hole.

This value, for the reasons mentioned before, is only an approximation and the exact value required to achieve the desired initial deceleration will have to be adjusted by trial and error after a few tests.

Example: 0.0203205 = 0.61 * Sh * Square Root(2*629185/950)
with K= 0.61 (see note 1)
UM = 950 (for water, UM =1000)

Sh = 0.000939075 m2
= 939 mm2

6.5) Then to keep the pressure constant along the stroke, despite the diminishing speed, the hole section must vary according to the formula:

(Sh current) = (Sh initial) * (Stroke-Travel)/(Travel)

Depending on the design, the linear dimensions must be calculated to meet the above condition.

7) You must now check the behaviour of the system under the conditions specified in the code i.e.:
- full load
- setting at the governor tripping speed (although this is not stated in the code, I think it is wise to select this speed rather than the 115% rated speed specified for the end-of-travel buffers).

If there is a problem, you might have to recalculate the buffer with a "light load" a little higher than 250 kg as was the case in the selected example. It appeared that, with this Rs of 3, the behaviour was acceptable with a "light load" equal to 25% of the rated load (400 kg)(Fig 4a).

You must then check the behaviour of the system with the critical 250 kg load to make sure that the declerations will still be low enough (Fig 3b).

8) As already mentioned, tests will be necessary to adjust the design:

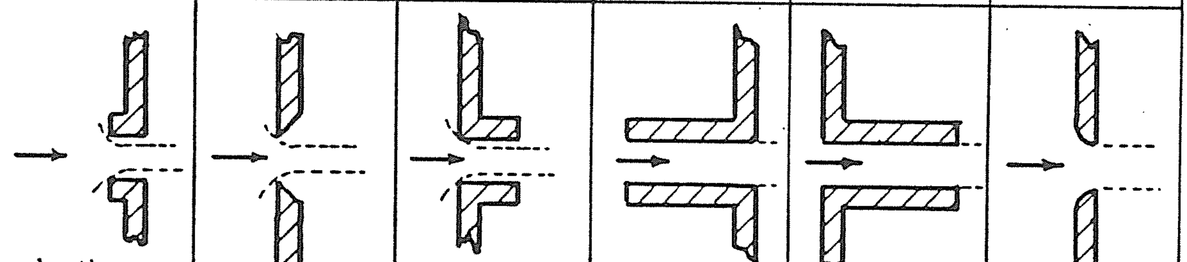
- because the value of K is not only an approximation but it may vary slightly along the stroke with the change of geometrical dimensions,
- because of the viscosity of the fluid selected

Because of one of the simplifying assumptions I made in the mathematical developments, the results of the tests should be more favourable than what those which the calculations indicate.

Indeed, all the calculations are made as if all the energy was dissipated in the buffer. However, to be able to support the buffer reaction, the instantaneous safety gear must have its rollers opening the jaws of the blocks to a certain extent and this means that some energy is accumulated in the safety blocks.
I do not think it improves significantly the deceleration curves in the case of the critical light load where the buffer reaction is minimal. On the contrary, it will smooth the peak deceleration in the case of safety gear setting with full load (Fig 3a and 4a).

9) The above theoretical developments have not been given a practical test.

Note 1 The "Cameron Hydraulic Data" from Ingersoll Rand is proposing the following values for K in the case of water:



RE-ENTRANT TUBE	SHARP EDGED	SQUARE EDGED	RE-ENTRANT TUBE	SQUARE EDGED	WELL ROUNDED
length 0.5 to 2 diam.		stream clears sides	length 2.5 diam.	tube flowing full	
K = 0.52	K = 0.61	K = 0.61	K = 0.73	K = 0.82	K = 0.98

TRACTION CALCULATIONS

New and more accurate approach to the problem

APP:05/1 Introduction of the proposed formula

APP:05/1/a
In the Handbook, under the reference HB/Clause 9.Note 1.a3, you can read the following comments:

"Quote"
Let us however remark that the formulae in the code are only approximations for the following reasons:

- the formulae do not take into account the throat opening (theta) always existing in the circular grooves,
- the formulae do not take into account the clearance always provided on the radius of circular grooves,
- the formulae do not take into account the thinning of the ropes due to stretching under load (both permanent and elastic),
- the coefficient C2 is introduced in the wrong place in the formulae which leads to wrong conclusions when wide V's are used,
- the code does not set a limit related to C2 for the wear in tight V grooves, which can result in insufficient traction in old sheaves,
- the formulae do not cover the case of undercut V grooves (sometimes called Y grooves) which can provide a constant traction if the undercut is properly selected in relation to the opening of the V.

Because there is a hidden safety factor in the selection of 0.09 for the value of the friction factor µ, the approximation does not lead to dangerous situations. However, the real margin of safety is not known, very different unit pressures can develop, with subsequent different rates of wear, and, in extreme cases, rope slidings have been experienced during braking in normal operation.

"End of quote"

In the next pages, we will analyze what all these remarks mean and propose new formulae successively for the circular grooves and for the V grooves.

When talking about the V grooves, we will cover the case of the undercut V grooves (also called Y grooves).

We will also give indications for the design of "constant traction" grooves.

APP:05/1/b
In order to see clearly the influence of the various factors involved, we will use the following traction formula:

$$\frac{T1}{T2} < C4 * e^{(C3*\mu*alpha)}$$

where the meaning of the factors is as follows.

- T1 and T2 have the values explained in HB/Clause 9:Note 1
- C3 is the wedging factor by which the radial force exerted on the rope is to be multiplied to calculate the traction.
- µ is the actual friction factor.

- (alpha) is the angle of wrapping on the traction pulley.

- C4 is a selected safety factor, the same for all types of grooves.

You will note that the factor C2 has disappeared from the formula; it was used by CEN to compensate, to a certain extent, the loss of traction due to wear in V grooves.
Instead, we will analyze the behaviour of all types of grooves during wear, taking also into account the variation in the diameter of the ropes.
We will calculate the minimum and maximum values for the wedging factor C3 and use, in the traction formula, the value giving the greatest safety.

Finally, instead of using the unreal low value of $\mu=0.09$, we will propose a more realistic value but, at the same time, we will introduce a selected safety factor, C4, which will give the same degree of reliability whatever the selected groove.

All these questions will be analyzed in the following paragraphs:

<u>APP:05/2</u> analysis of the action of wear in semi-circular grooves, either plain or undercut.
<u>APP:05/3</u> analysis of the action of wear in V grooves, either plain or undercut.
<u>APP:05/4</u> justification of the value $\mu=0.125$ and introduction of a sensible safety factor.
<u>APP:05/5</u> calculation and tabulations of the minimum and maximum values for the wedging factor C3.
<u>APP:05/6</u> condition of use of round and constant traction grooves.
<u>APP:05/7</u> condition of use of the narrow V grooves.
<u>APP:05/8</u> recommendations for the design of grooves

Moreover, in the attachment APP:06 you can find guide lines for the selection of grooves as well as the formulae and checklist for the calculation of the traction conditions when using the improved approach developed in the present attachment.

APP:05/2 Case of the circular grooves

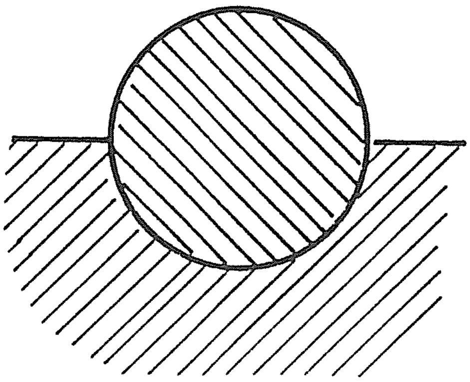
APP:05/2/a
The code formulae are based on the assumption that the rope is close-fitting in the groove up to the level of the center of the rope.

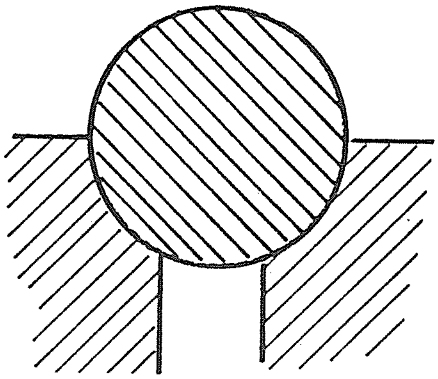
Fig 1a

dia rope = 100%
dia groove = 100%
(teta) = 0°
ß = 0°

Fig 1b

idem
idem
idem
ß = variable





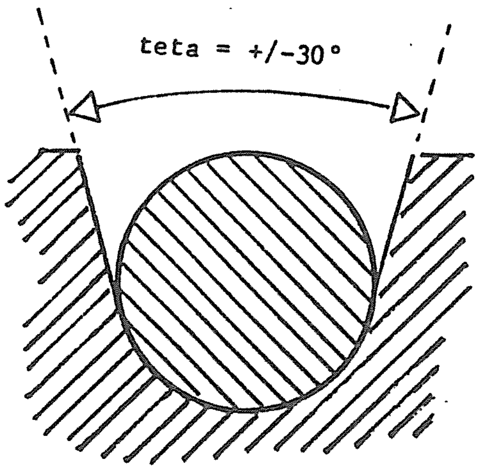
APP:05/2/b
But even if the rope fitted exactly in the groove, a throat of about 30° is provided for guiding the rope and allowing some misalignment

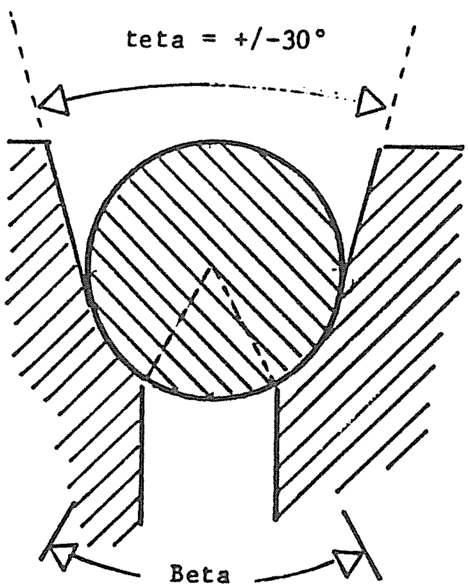
Fig 2a

dia rope = 100%
dia groove = 100%
(teta) = +/- 30°
ß = 0°

Fig 2b

idem
idem
idem
ß = variable



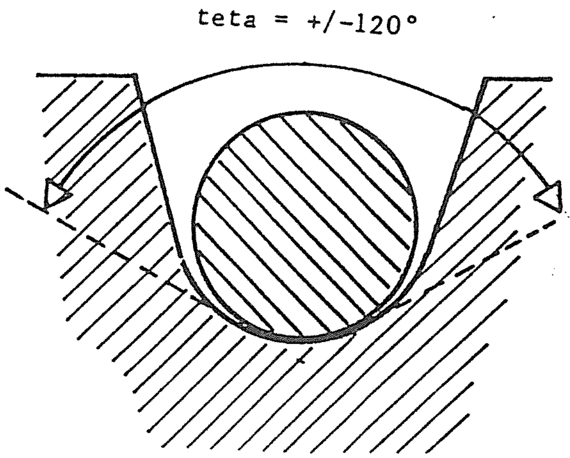


APP:05/2/c

But, there are tolerances on the diameter of the rope.
According to the Standard ISO/4344, the diameter of the rope at zero load must be between 102% and 106% of nominal (105% up to 10mm diam).
So, to make sure of accomodating the new rope without pinching, it is customary to select a groove diameter of 106% of the nominal diameter of the rope.

Moreover, due to elastic and permanent stretch, the rope thins rapidly, i.e. before the groove can wear significantly.
All liftmen know that the ropes stretch rapidly at the beginning of the lift operation. That stretching cannot go without a loss in diameter.
According to the book by Shitkow and Pospechow, the rope can lose up to 5% of its original diameter under load in the first quarter of its useful life and then, slowly, an additional 2%.
So, after some time of operation, the situation will be as follows:

Fig 3a		Fig 3b	
dia rope	= 95%	idem	
dia groove	= 106%	idem	
(teta)	= +/- 120°	(teta)	= +/- (180°-ß-60°)
ß	= 0°	ß	= variable



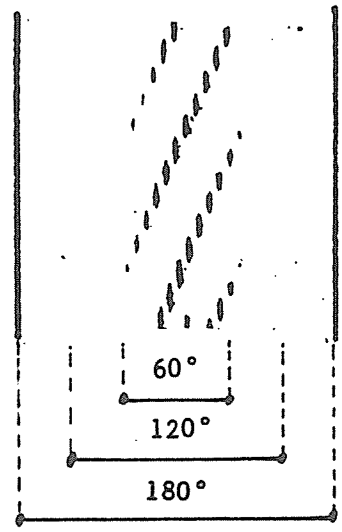
APP:05/2/d

The above effect is confirmed by inprints taken on a new sheave with a new rope.It shows that, even before stretching, the arc of contact is in the order of 60° when the rope is loaded with 1/12th of its breaking load.

Fig 4 : New rope, new groove

rope nominal dia: 13mm
groove dia   :   14mm
breaking load   : 7420kg
actual load   :   618kg

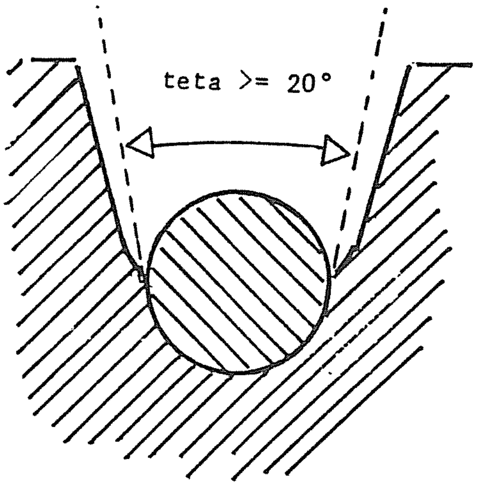
scale of picture: 2/1

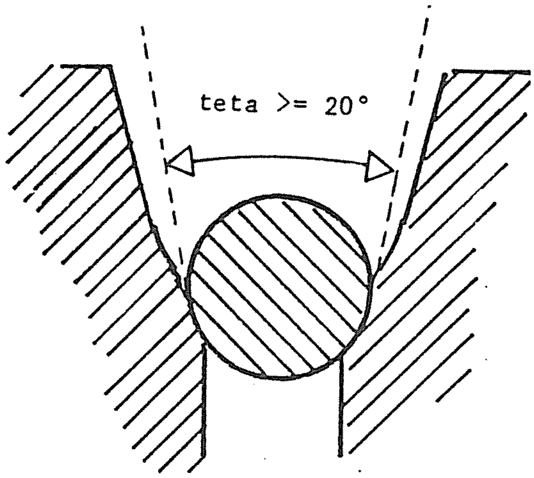


APP:05/2/e

Normally, the rope and the sheave will wear together and the rope will sink into the cast iron. The diameter of the worn groove will match the diameter of the worn rope but a throat opening will automatically be maintained because of the thinning process and because of the rope vibration. The minimum throat opening will be in the order of 20°.

Fig 5a		Fig 5b	
dia rope	= 93%	idem	
dia groove	= 93%	idem	
(teta)	= +/- 20°	idem	
ß	= 0°	ß	= variable





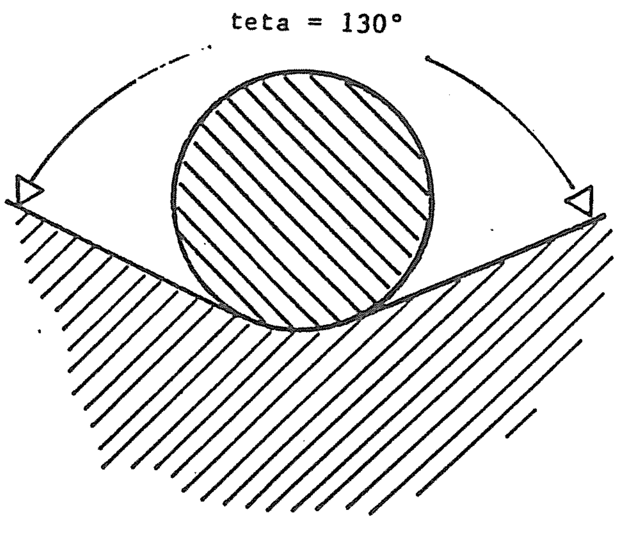
APP:05/2/f

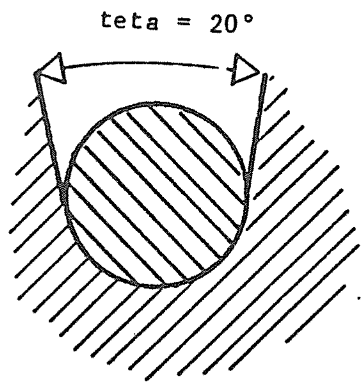
This means that, in the case of a full semi-circular groove, we have to calculate the traction in a new groove as if the throat opening (teta) was about 120°. To be on the safe side, we will use 130° because this is for calculating a value not to be exceeded.

In an old groove, the traction will be higher but, for the calculation, we have to reckon with a throat opening of at least 20°. To be on the safe side, we will use 20° because this is for calculating a value which must be exceeded (rope slipping with counterweight on the buffer).

It is interesting to note that the "constructional (teta)" has actually no influence on the calculations.

Fig 6a — traction new groove SMALLER THAN Fig 6b — traction old groove





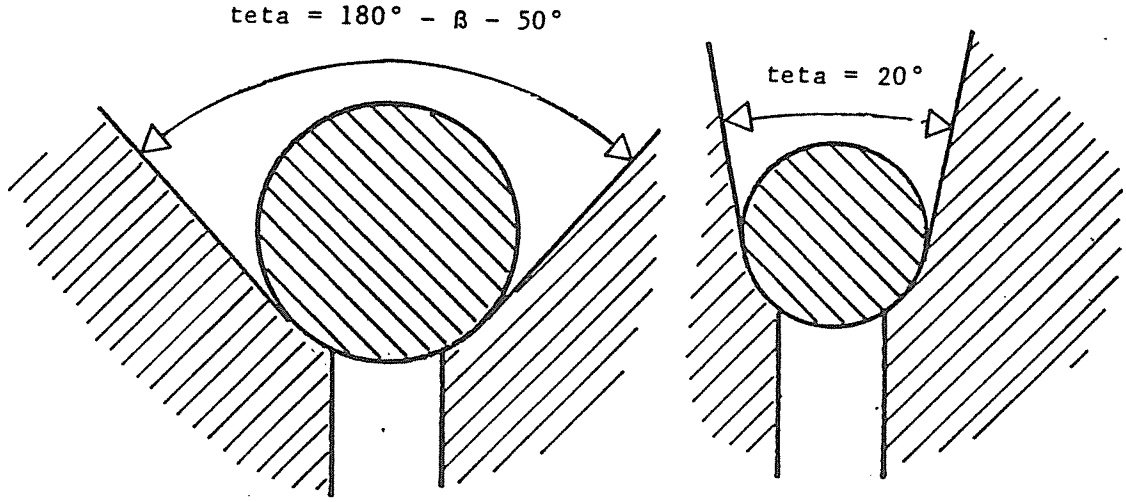
APP:05/2/g

In the case of undercut semi-circular grooves, we have similarly to consider that the rope is resting on the shoulders of the undercut of a new groove only on 25° on each side. For the calculation of the minimum traction, we will use:

$$(\text{teta}) = 180° - ß - 50°$$

Here also, the "constructional teta" has no impact except if it were, by any chance, greater than the above calculated value, which would be a bad design anyway (we will disregard this possibility).

Fig 7a traction new groove SMALLER THAN Fig 7b traction old groove.



APP:05/2/h

The formulae and the calculations will be developed in chapter APP:05/5 and APP:05/6 after reviewing the case of the V grooves and the real value of the friction factor μ.

APP:05/3 Case of the V grooves

APP:05/3/a

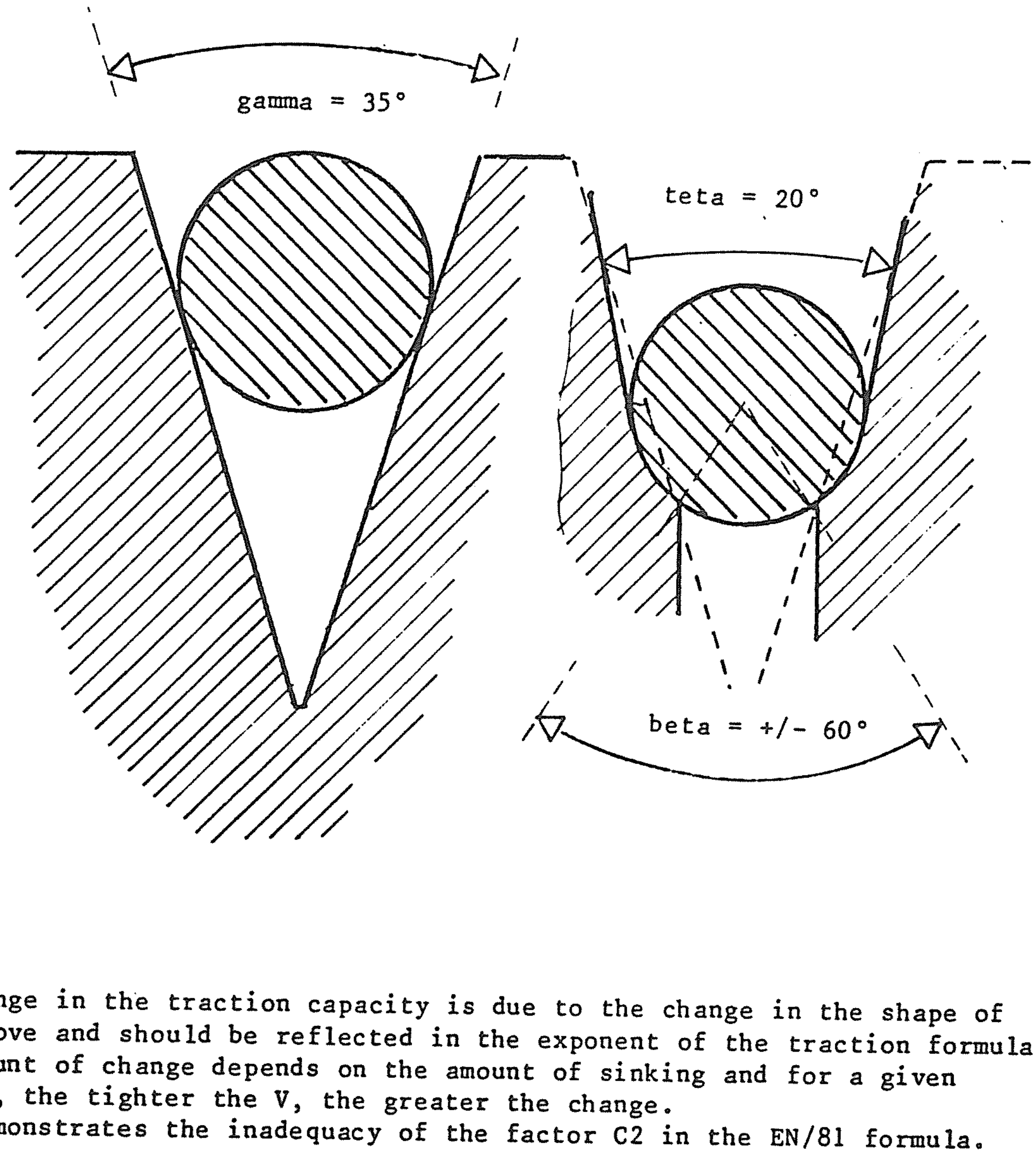
The usual "tight V" groove behaves according to the V groove theory when it is new but, after sinking in the groove, it behaves exactly as a worn undercut circular groove.

The value of the angle (beta) depends on how much the rope has been allowed to sink whereas the angle (teta) will always be around 20° providing the sinking reaches at least the value d*tang(0.5*gamma) in addition of course to the sinking due to the thinning of the rope.

The example Fig 8a and 8b shows what happens to a (gamma) = 35° V groove after the rope has sunk the equivalent of 1 rope radius.

Fig 8a
traction new groove GREATER THAN

Fig 8b
traction old groove



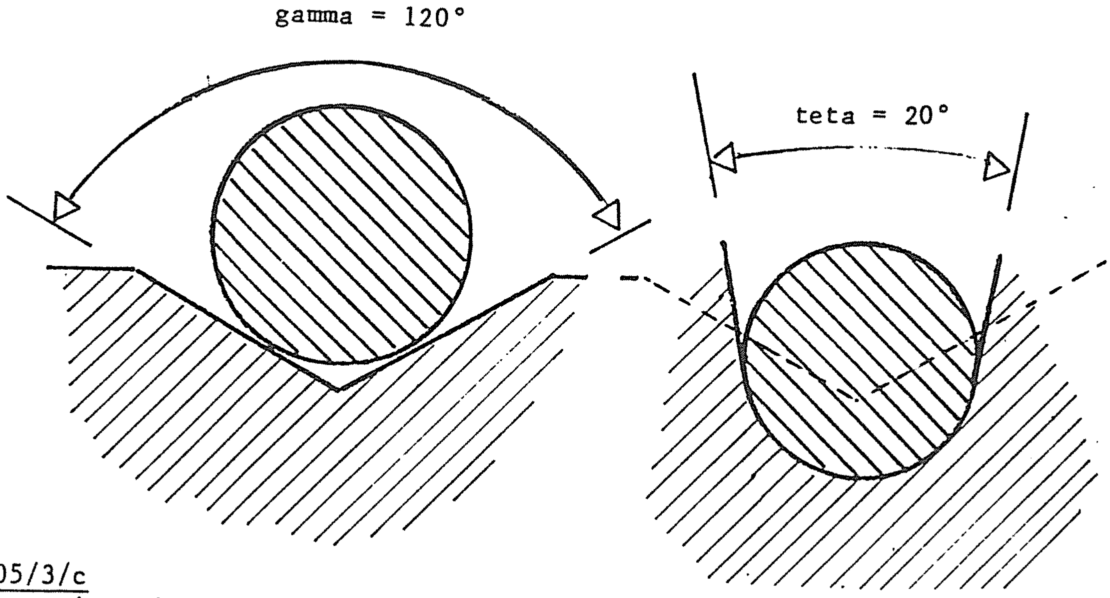
The change in the traction capacity is due to the change in the shape of the groove and should be reflected in the exponent of the traction formula.
The amount of change depends on the amount of sinking and for a given sinking, the tighter the V, the greater the change.
This demonstrates the inadequacy of the factor C2 in the EN/81 formula.

APP:05/3/b

Let us now see what would happen with a wide open V groove.
The reasoning is exactly the same but in this case, the traction in the "new" groove is obviously smaller than the traction in the "old" groove as illustrated in the example of Fig 9 where the angle (gamma) is 120°.

Fig 9a　　　　　　　　　　　　　　Fig 9b

traction new groove　SMALLER THAN　traction old groove



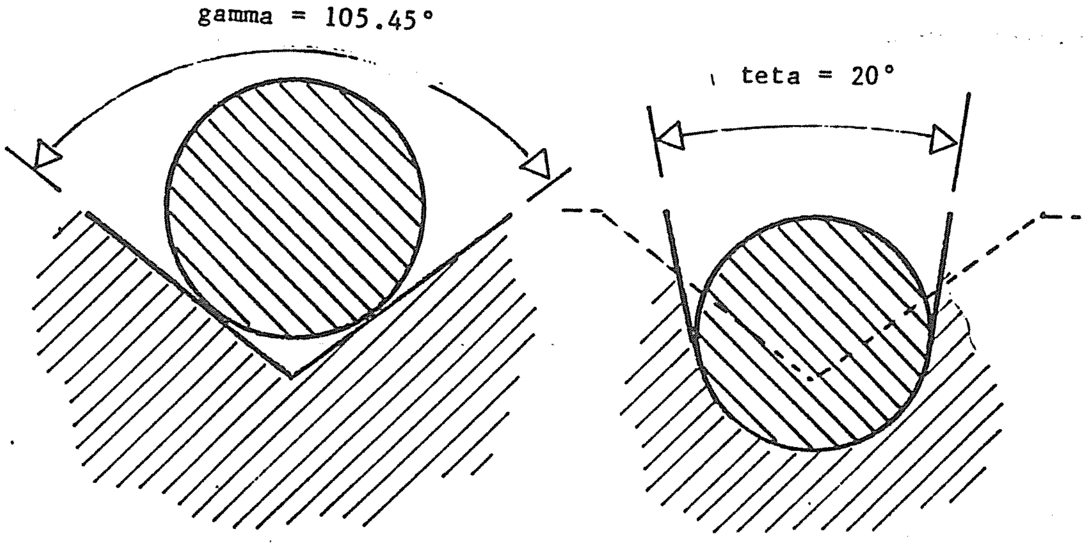
APP:05/3/c

It comes immediately to the mind that, between the tight V and the wide open V, there must be a value of the angle (gamma) for which the traction "new" would be equal to the traction "old".

For the plain V, this happens for (gamma) = 105.451 degrees.
In this case (Fig 10), the traction "new" is equal to the traction "old" and, as the Fig 10b is exactly alike the Fig 6b, is also equal to the traction of the full semi-circular groove after wear. It has even the advantage that the same value is maintained from the beginning

Fig 10a　　　　　　　　　　　　　　Fig 10b

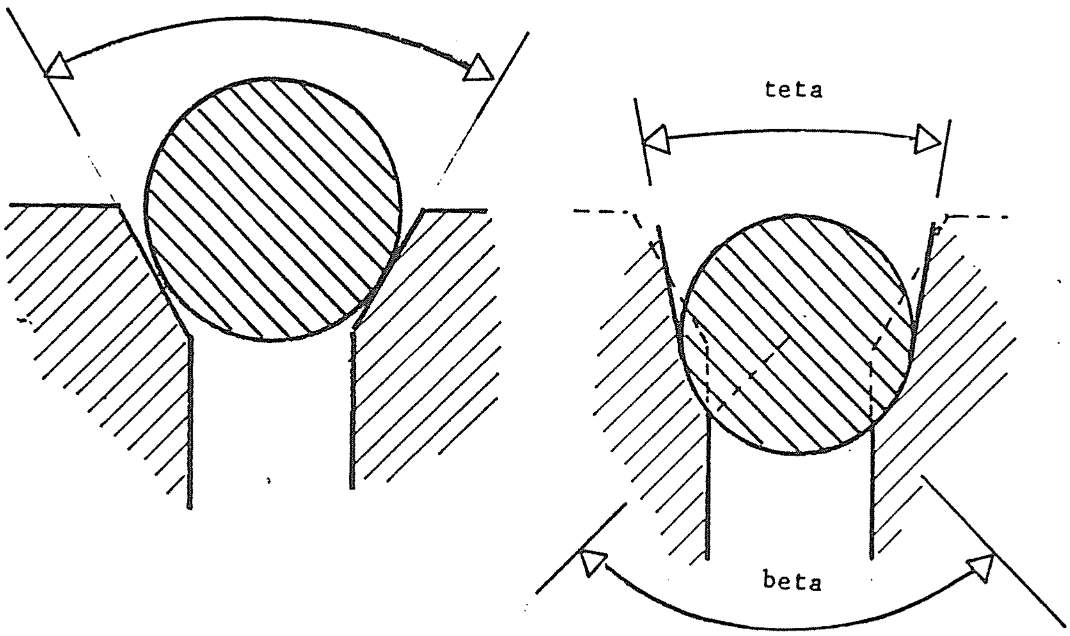
traction new groove　EQUALS　traction old groove



By undercutting a V groove you can limit the drop of the traction capacity to that of a semi-circular groove having the same undercut.
Following the same reasoning as in APP:05/3/a to c above, you will come to the conclusion that to each value of the possible undercuts (beta), therecorresponds a value of (gamma) such that the traction "new" is exactly equal to the traction "old".

Fig 11a — Fig 11b

traction new groove — EQUAL — traction old groove



The values of (gamma) as a function of (beta) are given by the formula:

$$\frac{1}{\sin(0.5*\text{gamma})} = \frac{4*\cos(0.5*\text{teta}) - 4*\sin(0.5*\text{beta})}{3.14 - (\text{beta}) - \sin(\text{beta}) - (\text{teta}) + \sin(\text{teta})}$$

where all the angles must be expressed in radians.

Using for (teta) a value of 0.35 Rad (20°) for the reasons explained in the preceeding paragraphs, you can prepare the table Fig 12.

You will remark that it is impossible to have "constant traction" V grooves for angles (gamma):

- smaller than 52° because you already reach the practical maximum of 105° for the angle of undercut (beta),
- greater than 105° because the undercut is already down to zero which corresponds to the full semi-circular groove.

By undercutting to 105° the V grooves with (gamma)'s smaller than 52° you will, to an extent, limit the traction loss. To what extent will be explained later.

Fig 12: Combinations of Gamma's and Beta's for constant traction

Final BETA degrees	GAMMA degrees	Original BETA for anticipating the rope thinning to 95% of original diameter
0	105.45	0 (equivalent to full semi-circular groove)
5	103.68	4.75
10	101.80	9.50
15	99.81	14.25
20	97.72	18.99
25	95.53	23.73
30	93.26	28.47
35	90.91	33.20
40	88.47	37.92
45	85.97	42.64
50	83.40	47.34
55	80.76	52.04
60	78.08	56.72
65	75.33	61.39
70	72.55	66.04
75	69.71	70.67
80	66.84	75.27
85	63.94	79.85
90	61.00	84.40
95	58.04	88.92
100	55.06	93.39
105	52.06	97.82 (maximum acceptable undercut)

APP:05/3/e

The above table gives "traction new" = "traction old" but what happens in the intermediate stages of groove wear?
By preparing a suitable computer program, you can check that there are some variations of the traction capacity but these variations are small by comparison with the ones occuring in conventional grooves. In the beginning of the wearing process, the traction capacity drops gradually to about 92%or 95% of the original value (the greater the gamma angle, the lesser the drop) then the traction picks up to regain the original value when the sinking is about one half of the rope radius.

It is important to note that for calculating the width of the undercut, you must either use the angle of the first column of Fig 12 with a diameter reduced to 95% of its nominal value (thinning of the rope) or, alternatively, use the nominal diameter with the corrected angle of the third column.

This is specially important for the large undercuts because ignoring the thinning of the rope would lead to excessive traction and possible squeezing of the rope in the groove after thinning.

The values of C3 corresponding to each Gamma angle are given Fig 16.

Of course, if the rope supplier can guarantee a lower loss in diameter (for example in the case of prestretching) you should ignore the third column and use the guaranteed final diameter with the angles of the first column.

APP:05/4 Realistic friction factor and introduction of a safety factor

APP:05/4/a

What the code calls "coefficient of friction of the ropes in the grooves", for which the symbol "f" is used, is really an apparent friction factor.

The real friction factor is "μ" and practically does not depend on the shape of the groove.

To get the apparent friction factor, this μ must be multiplied by what we will call a "wedging factor" using the symbol "C3".

$$f = \mu * C3$$

The values to be retained for C3 will be discussed in APP:5/5.

APP:05/4/b

The EN-81 code imposes for μ the value of 0.09.

As said earlier, there is a hidden safety factor in this value because all reported experiments indicate higher values.

Some reports give only the apparent friction factor "f", some others give values for "μ" but, in both cases, the only thing which can be measured by the laboratories is the tractions on both sides of the sheave.
The value of "μ" can then be deducted from the formula:

$$T1/T2 = e \text{ exponent}(\mu * C3 * alpha)$$

where (alpha) is the angle of wrapping in Radians.

Many reports do not give the value used for the wedging factor "C3". Some others do not describe the exact conditions of the test. It is then to be feared that an erroneous expression has been used for the C3.

One of the most reliable reports is the "Treibfaehigkeit von geharteten Treibscheiben mit Keilrillen" from Dr-Ing Michael Mulkow because it describes exactly the conditions of the reported tests. It is from this publication that the Fig 13 has been taken (see next page).

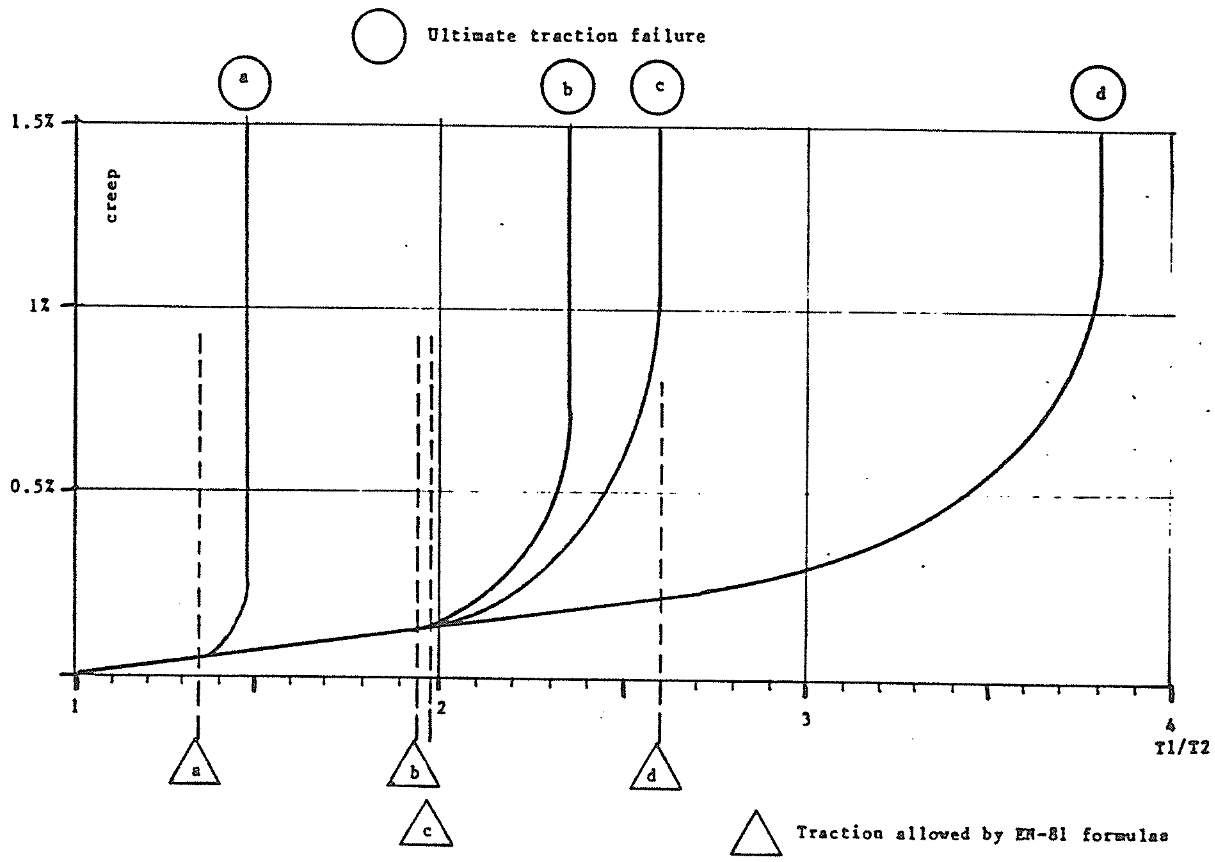
Let us first agree that the most interesting value of μ is the one calculated when the ropes start slipping (ultimate traction failure).
The ratios T1/T2 at which this ultimate traction failure does occur are taken from Fig 13. Using for (a) the value C3 = 1 and for (b), (c),(d) the values taken from the diagrams given in APP:5/5 for new grooves, you can deduce the value of μ for each of the four experiments.

You will find for (a) : $\mu = 0.123$
(b) : $\mu = 0.126$
(c) : $\mu = 0.126$
(d) : $\mu = 0.125$

These values are remarkably consistent.
These values are independant of the pressure in our range of applications. They would be slightly greater if the ratio D/d were notably above 40 but, on the other hand, they would be slightly smaller for rope diameters less than 16mm (still according to Dr-Ing Mulkow).

Fig 13: Experimental traction limits.
d = 16mm
D = 600 or 700mm
T1 = 6 kN
oil lubrication

(a) = cylinder (flat drum)
(b) = semi-circular undercut groove (ß=102°)
(c) = V groove with gamma = 50°
(d) = V groove with gamma = 35°



(continuation from previous page)
So the average value of μ = 0.125 is certainly meaningful.

APP:05/5/c
When using 0.09 for the value of μ, you introduce a variable factor of safety which amounts to:

for T1/T2 limits of 1.4 (EN formulae) : factor of safety = 1.14
1.8 (EN formulae) : factor of safety = 1.26

There is no justification for a variable factor of safety so we propose to use μ = 0.125 but to introduce a fixed factor of safety of 1.25 in the formula (i.e use, as a limit, only 80% of the ratio T1/T2 which would lead to complete loss of traction).

By taking this safety factor, you will:
- have values consistent with the ones resulting from the present EN-81 calculations but with a better knowledge of your real margin of safety,
- have a creep inferior to 0.5% which is important for limiting wear.

In normal operation, the creep will be still lower because the lift will run most of the time with only a fraction of the rated load.

APP:05/4/d
If you want to calculate the possibility of slipping, you must of course eliminate the safety factor.

Then, if you want to analyze what happens after the slipping has started, you must remember that the friction factor will drop with increasing gliding speeds. The following formula is proposed:

$$\mu = \frac{0.125}{1 + 0.125*Vg}$$

where Vg is the figure expressing the gliding speed in m/s.
This would be, for example, in the case of a brutal emergency stop.

APP:05/4/e
All the above considerations apply in the case of cast iron or steel grooves and steel ropes.

If another material is used, the manufacturer must supply his own data with supporting documents and get a derogation from the proper authorities.

Reportedly, with polyurethane grooves the friction factor µ is about twice the one expected from cast iron.

APP:05/5 Wedging factors values

APP:05/5/a

In the case of semi-circular grooves, the wedging factor will be lower than the one which would be indicated by the EN-81 formula when the groove is new and will then gradually increase as the rope sinks into the cast iron. (see APP:05/2 from b to g)

In the case of undercut semi-circular grooves, the angle (beta) increases as the rope gets thinner because the width of the undercut is constant.

Fig 14: - the column N° 1 gives the angle (beta) calculated with the nominal diameter of the rope,
- the column N° 2 gives the wedging factor C3 for the new rope in the new groove,
- the column N° 3 gives the actual angle (beta) when the diameter of the rope has been reduced to 95% of its nominal value due to the stretching,
- the column N° 4 gives the wedging factor for the thinned rope in the worn out groove.

Of course,if you have evidence that the rope will not lose so much in diameter, you have to calculate your own values for columns 3 and 4. The column N° 5 gives, between brackets, what the final wedging factor would be if the rope were not thinning at all; in that case, the angle (beta) would also remain constant as in the column N° 1.

It is not recommended to select grooves where the "Beta final" exceeds 110° which means that the "Beta initial" should not exceed 100° with the usual ropes where a thinning of at least 5% is to be expected.

FIG 14: For full and undercut semi-circular grooves, tabulation of initial and final wedging factors.

Beta init'l	C3 init'l	Beta final	C3 final	idem if d constant)
0°	1.03	0.0°	1.26	(1.26)
5°	1.04	5.3°	1.27	(1.27)
10°	1.06	10.5°	1.29	(1.29)
15°	1.07	15.8°	1.31	(1.31)
20°	1.09	21.1°	1.33	(1.33)
25°	1.11	26.3°	1.36	(1.35)
30°	1.13	31.6°	1.38	(1.38)
35°	1.16	36.9°	1.41	(1.40)
40°	1.19	42.2°	1.45	(1.43)
45°	1.22	47.5°	1.48	(1.47)
50°	1.26	52.8°	1.53	(1.50)
55°	1.30	58.2°	1.57	(1.54)
60°	1.35	63.5°	1.62	(1.59)
65°	1.40	68.9°	1.68	(1.64)
70°	1.46	74.3°	1.74	(1.69)
75°	1.53	79.7°	1.81	(1.75)
80°	1.61	85.2°	1.89	(1.82)
85°	1.70	90.7°	1.98	(1.89)
90°	1.80	96.2°	2.09	(1.97)
95°	1.92	101.8°	2.20	(2.06)
100°	2.06	107.5°	2.34	(2.16)

In the case of plain V grooves, the initial value of the wedging factor will be exactly equal to the calculated one but, when the flanks of the groove start to wear out,:

- the wedging factor will deteriorate if the Gamma angle is smaller than 110°. (the smaller the angle, the greater the deterioration),
- the wedging factor will increase if the Gamma angle is between 110° and 180° (case of the flat drum),

The wedging factor will remain nearly constant for Gamma values around 110°

Fig 15: For plain V grooves, variation of the wedging factor C3 due to the wear of the groove flanks as a function of the sinking of the rope resulting from the combination of the wear of the groove and of a rope thinning as can be expected from a regular ISO rope.

Gamma	C3 new	C3 sinking 0.125*d	C3 sinking 0.250*d	C3 sinking 0.375*d	C3 sinking 0.500*d
25°	4.62	2.91	2.24	1.98	1.81
30°	3.86	2.57	2.03	1.80	1.66
35°	3.33	2.31	1.86	1.67	1.54
40°	2.92	2.12	1.73	1.56	1.46
45°	2.61	1.96	1.63	1.48	1.39
50°	2.37	1.84	1.55	1.42	1.33
55°	2.17	1.74	1.48	1.36	1.29
60°	2.00	1.65	1.42	1.32	1.26
65°	1.86	1.58	1.38	1.29	1.23
70°	1.74	1.52	1.34	1.26	1.21
75°	1.64	1.47	1.30	1.23	1.20
80°	1.56	1.42	1.28	1.22	1.19
85°	1.48	1.38	1.25	1.20	1.18
90°	1.41	1.34	1.23	1.19	1.18
95°	1.36	1.30	1.22	1.18	1.18
100°	1.31	1.27	1.21	1.18	1.18
105°	1.26	1.24	1.20	1.18	1.18
110°	1.22	1.22	1.19	1.18	1.19
115°	1.19	1.20	1.18	1.18	1.19
120°	1.15	1.18	1.18	1.18	1.20
125°	1.13	1.16	1.18	1.18	1.21
130°	1.10	1.14	1.18	1.19	1.21
135°	1.08	1.13	1.18	1.19	1.22
140°	1.06	1.12	1.17	1.20	1.23
145°	1.05	1.11	1.17	1.21	1.24
150°	1.04	1.10	1.17	1.21	1.24
155°	1.02	1.10	1.17	1.22	1.25
160°	1.02	1.09	1.17	1.23	1.26
165°	1.01	1.09	1.17	1.23	1.26
170°	1.00	1.08	1.17	1.24	1.26
175°	1.00	1.08	1.17	1.24	1.26
180°	1.00	1.08	1.17	1.24	1.26

Of course V grooves with large Gamma angles are meaningless, except from a theoretical point of view.

However, Gamma angles ranging from 50 to 110° present a special interest as explained in the next paragraph.

APP:05/5/c

By judiciously undercutting a V groove,

- the wedging factor can be kept nearly constant despite the wear for Gamma angles ranging from 110° down to 50°
- the deterioration of the wedging factor can be limited to a certain extent for smaller Gamma angles.

As already said for the circular grooves in APP:05/5/a, it is not recommended to use a wider undercut than the one corresponding to an ultimate Beta angle of 110°. This is the reason for the above limitation.

In Fig 16:- column 1 gives the selected Gamma angle
- column 2 gives the corresponding ultimate Beta angle
- column 10 gives the maximum value of the wedging factor C3
- column 11 gives the minimum value of the wedging factor C3
- columns 3 to 9 give the width of the undercut for each of the ISO rope diameters.

Fig 16: Optimum undercuts for V grooves

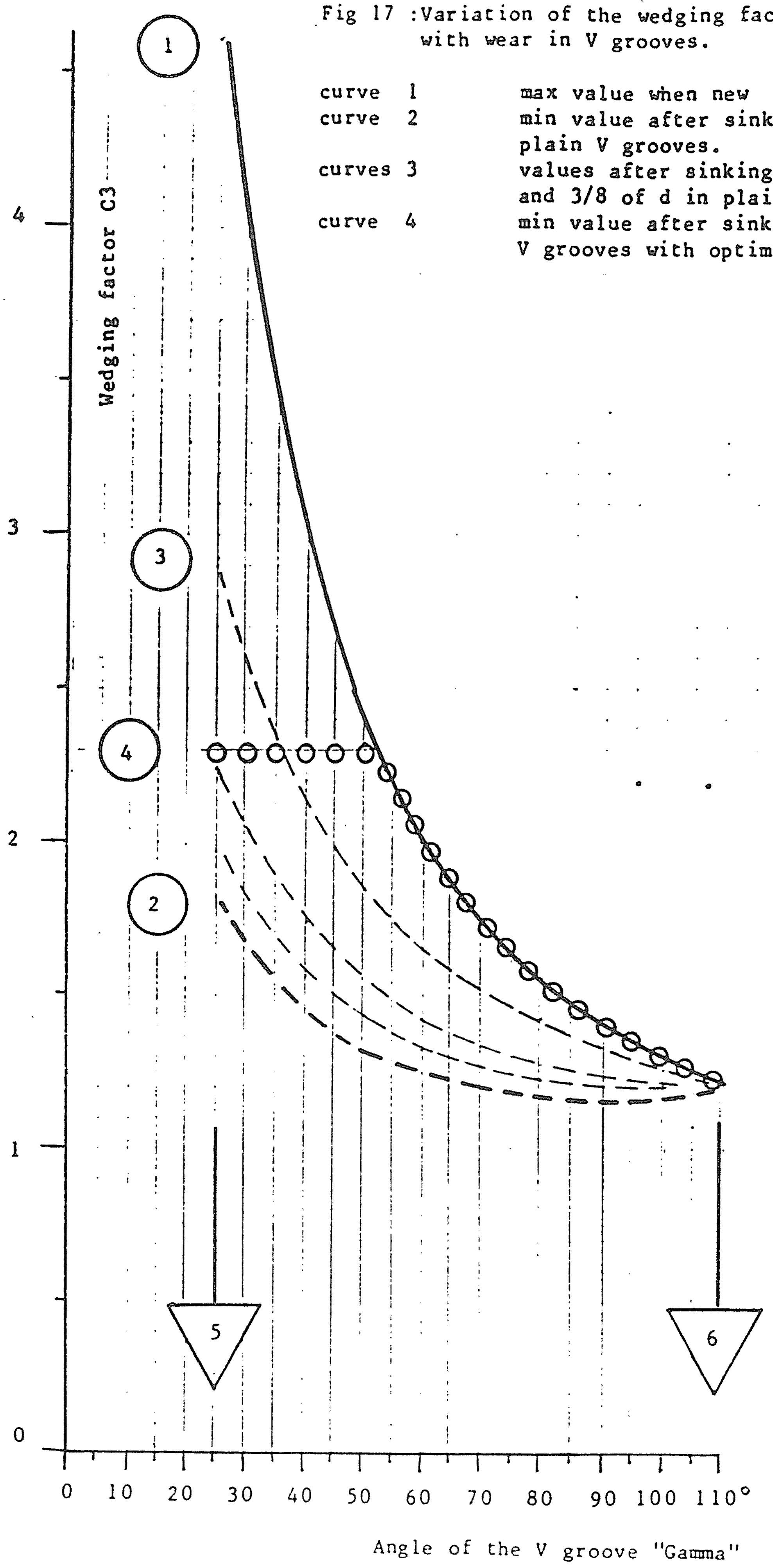
Gamma	Beta	Width of undercut for each ISO rope diameter 8 mm	10 mm	11 mm	13 mm	16 mm	19 mm	22 mm	Values of C3 max		min
105°	1.3°	0.1	0.1	0.1	0.1	0.2	0.2	0.2	1.26	Constant traction	1.26
100°	14.6°	1.0	1.2	1.3	1.6	1.9	2.3	2.7	1.31		1.31
95°	26.2°	1.7	2.2	2.4	2.8	3.4	4.1	4.7	1.36		1.36
90°	36.9°	2.4	3.0	3.3	3.9	4.8	5.7	6.6	1.41		1.41
85°	46.9°	3.0	3.8	4.2	4.9	6.0	7.2	8.3	1.48		1.48
80°	56.5°	3.6	4.5	4.9	5.8	7.2	8.5	9.9	1.56		1.56
75°	65.6°	4.1	5.1	5.7	6.7	8.2	9.8	11.3	1.64		1.64
70°	74.5°	4.6	5.8	6.3	7.5	9.2	10.9	12.7	1.74		1.74
65°	83.2°	5.0	6.3	6.9	8.2	10.1	12.0	13.9	1.86		1.86
60°	91.7°	5.5	6.8	7.5	8.9	10.9	13.0	15.0	2.00		2.00
55°	100.1°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	2.17		2.17
50°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	2.37	Narrow V gr	2.28
45°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	2.61		2.28
40°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	2.92		2.28
35°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	3.33		2.28
30°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	3.86		2.28
25°	105.0°	5.8	7.3	8.0	9.5	11.7	13.8	16.0	4.62		2.28

You will remark that for Gamma angles greater than 50°, the maximum and minimum values of the wedging factor C3 are equal.

On the other hand, for smaller Gamma angles, the final (minimum) value can not exceed 2.28 because it corresponds to the widest possible undercut (Beta = 110°). This mean that the traction condition will deteriorate if the groove is allowed to wear.

Fig 17, on the next page, illustrates the advantage of the undercut V grooves as compared to the plain V grooves.

Fig 17 :Variation of the wedging factor C3 with wear in V grooves.



The advantage of using the optimum undercut appears clearly in Fig 17.

For all Gamma angles from 110° down to 50°, there no deterioration of the C3 due to wear whereas, if we take the example of Gamma=55°, the value of C3 would drop from 2.2 to 1.3 after sinking of d/2 in a plain V groove.

For Gamma angles ranging from 50° down to 25°, all what the optimum undercut can do is to limit the drop to a C3 value of 2.3.
If we take the example of Gamma=35°, the drop will be from 3.4 to 2.3 with the optimum undercut whereas it would drop to 1.5 in a plain V groove.

Point 5 is the smallest Gamma angle which may be used for avoiding self clamping. This is an experimental limit.
In fact, the formula for calculating the self clamping angle is:

$$\tan(\text{Gamma}/2) \leq \mu$$

Which would mean that for the sideway sliding of a steel rope on a cast iron surface a value of $\mu = 0.22$ can be expected.

Point 6 is the limit above which the value of C3 would improve with wear instead of deteriorating.

APP:05/6 Conditions of use of round and constant traction grooves

APP:05/6/a Relation between the wedging factor C3 and the wrapping angle

First, let us call attention to the fact that the value of the ratio T1/T2 lies, in all usual applications, between relatively narrow limits:

$$1.4 < (T1/T2) < 2.0$$

Then, considering the formula:

$$T1/T2 < 0.8 * e^{(\mu*C3*alpha)}$$

where $\mu=0.125$ the value of (C3*alpha) is, for a given value of (T1/T2), a constant to be calculated by the formula:

$$C3*alpha = 8*Ln(1.25*T1/T2)$$

So, if you know the ratio (T1/T2) and the wedging factor C3, you can calculate the minimum wrapping angle alpha which will be needed for the required traction (and vice-versa).
The N.E. quadrant of Fig 18 gives the results of the calculations for various values of (T1/T2).

The value of (T1/T2) for a typical lift without compensation will be in the order of magnitude of 1.8. You can decrease this value by using a heavy car, then compensating ropes, then (still further) compensator. But all this increases the cost.

As for the wrapping angle "alpha", the simplest lifts try to do with values between 145 and 180°.
Exceeding 180° was, until lately, done exclusively by using the double-wrap arrangment which allowed alpha values up to almost 360° but is costly and generates high internal stresses in the installation.
Lately, KONE in Europe (under the name of ESW,"extended single wrap") and OTIS in the USA (LWT,"long wrap traction"), have introduced a system by which the rope passes only once on the traction sheave but the rope going to the counterweight crosses the rope coming from the car allowing wrapping angles up to about 230°.

APP:05/6/b Relation between the wedging factor C3 and the undercut "beta".

The following developments apply both to the conventional semi-circular grooves with undercuts varying from zero to 100° and to the constant traction type of grooves as recommended in APP:05/7.

Indeed, if you look Fig 14 for the case of the conventional semi-circular grooves, to each value of "C3 initial", which has to be used for the traction calculation, there corresponds a value of "beta".

The same, for the constant traction grooves, if you look at the second and the last columns of Fig 16.

The relationship between the wedging factor and the wrapping angle is illustrated in the N.W. quadrant of Fig 18.

APP:05/6/c Relation between the rope speed and the undercut "beta"

Any of the formulae used for evaluating the pressure between the rope and the flanks of the groove is conventional and valid only for comparison purposes. I have no reason to propose formulae other than the ones selected in EN 81-Note 2 to Chapter 9 but the formula proposed for V grooves should be limited to "Gamma" angles between 30° and 45°.

Experience indicates that when using the formulae proposed for calculating the pressure along with the formula proposed for calculating the limit, a satisfactory life time can be expected (if the sheave material is correctly selected of course).

Let us first remark that the minimum breaking load of the steel ropes according to the specifications of the ISO/4344 Standard can be expressed, in Newtons, by the formula:

519*d2 for the single tensile 1770 N/mm2 wires
440*d2 for the dual tensile wires (equiv. 1500 N/mm2).

where "d" is the diameter of the rope expressed in millimeters.

We now notice that (T/n) is equal to the minimum breaking load of 1 rope divided by the safety factor for which the minimum is dictated by EN/9.2.2 for the ropes.

Lastly, "D", the diameter of the traction sheave, can be expressed as d multiplied by a ratio which shall be greater than 40 according EN/9.2.1.

Putting all this together for the case of the single tensile ropes, the formula of the code can be written as:

$$p = \frac{519 * d2}{(\text{safety factor}) * d * (\text{ratio}) * d} * \frac{8 * \cos(\beta/2)}{3.14 - \beta - \sin\beta}$$

Calling X1 the product of (safety factor) by (ratio) and after simplifications, this will read:

$$p = \frac{4152 * \cos(\beta/2)}{X1 * (3.14 - \beta - \sin\beta)} \qquad (\text{N/mm2})$$

All the angles are, of course, expressed in Radians.

The lowest possible value for "X1" is 12*40 = 480 and it will generally not exceed 800 (which, for example, could be 16*50).

But the maximum allowed specific pressure being expressed as a function of the rope speed, the maximum for the angle ß can be directly related to the ropes speed for a given X1.

The results are illustrated in the S.W. quadrant of Fig 18.

The same diagrams may be used for the dual tensile ropes providing that the value of X1 be calculated by the formula: X1=(1.18)*(ratio)*(safety factor) In this case, the lowest possible value for X1 would be 566.

APP:05/6/d Directions for using the diagrams of Fig 18

(The Fig 18 can be found a few pages later)

Amongst all the above factors, the only value fixed by the lift contract is the speed of the lift.
You will have to decide, based on your knowledge of the available equipment, whether you use direct traction or reeving.

That being decided, then the rope speed is known. This will be the starting point on the South axis of Fig 18.

You draw an horizontal line to the selected value for X1 and the maximum value acceptable for ß can be read on the West axis of Fig 18.
You select the value which suits you best below this maximum.

Having decided to try either conventional semi-circular grooves or the constant traction grooves recommended in this Handbook, the value of the wedging factor C3 to be used for calculating the traction can be read on the North axis of Fig 18.

Having calculated the ratio (T1/T2), you can find the minimum wrapping angle "alpha" on the East axis.

If this value does not suit you, you can play around with:
-the traction sheave diameter,
-the number and diameter of the ropes,
-the type of groove,
-the ratio T1/T2,
selecting the best compromise for the use of your equipment.

First example:

- lift speed 1.6m/s
- direct traction
- let us try to use the smallest possible safety factor and smallest possible traction sheave, then X1=500 (12*41.5),
- let us try to use conventional round grooves,
- let us try to avoid compensating ropes and let us assume that T1/T2 is equal to 1.8 in this case,

The maximum undercut (ß) will be 74°.Let us use 70°.
The corresponding C3 is 1.46.
The minimum wrapping angle would be 255°.

But you probably do not want to exceed 160° for the wrapping angle so you will have to work on all the factors
- decrease T1/T2 by using compensation (for example 1.6),
- use the constant traction grooves,
- use a higher value for X1 (for example 40*15=600).

Now ß=85°, C3=2, (T1/T2)=1.8.

We are getting close to the limits but let us remember that, in the present more accurate calculation system, there remain a 20% safety margin on the traction factor.

With the calculation method of EN 81, this margin would be greatly reduced because a C3 of only 1.8 would seem adequate.

Second example:
Let us consider a 2.5m/s gearless lift with 2/1 roping which means a rope speed of 5 m/s.

We want to avoid the double-wrap but the the reeving pulleys can be arranged to have 180° wrapping on the traction sheave.

We will have compensation of some sort and let us assume we can reduced T1/T2 to a value of 1.6.

Knowing by experience that this will be a limit case, let us directly plan using the constant traction grooves.

The only thing you can still play with is the value of X1, so in this case you have to work backwards starting from 180° on the East axis of Fig 18. Going vertically to the curve corresponding to T1/T2=1.6, you find that a C3 value of 1.77 is required to get the proper traction.

Even using the constant traction grooves, the angle ß need to be 77° in order to have enough traction.

Since the rope speed is 5 m/s, you will have to select for X1 a value of at least 730 to keep the pressure under the required limit. This can be done by using, for example, a safety factor of 15 and a ratio D/d of 50.

Obviously, for such an application, we are working close to practical limits. With the ESW and LWT systems we would have a lot more freedom for selecting the parameters without having to accept the cost of a double-wrap. We could, for example, use lower values for X1.

Third example:
Let us take the case of a double-wrap with a 320° wrapping angle. Providing the value of T1/T2 does not exceed 1.7, we will not need a C3 higher than 1.03 which means that we may use a conventional full semi-circular groove with any speed and that we may use the minimum of 480 for the value of X1 because, using the diagram, we stay on the North then on the South axis of Fig 18. The ropes would even be at ease because for a speed of 6 m/s for example, we could use a ß angle up to 50° whereas we use an angle of 0°.

The advantage of the constant traction grooves, in this case, is that you could accept a value as high as 1.9 for T1/T2 because the minimum C3 value for this groove is 1.26.

(see Fig 18 next page)

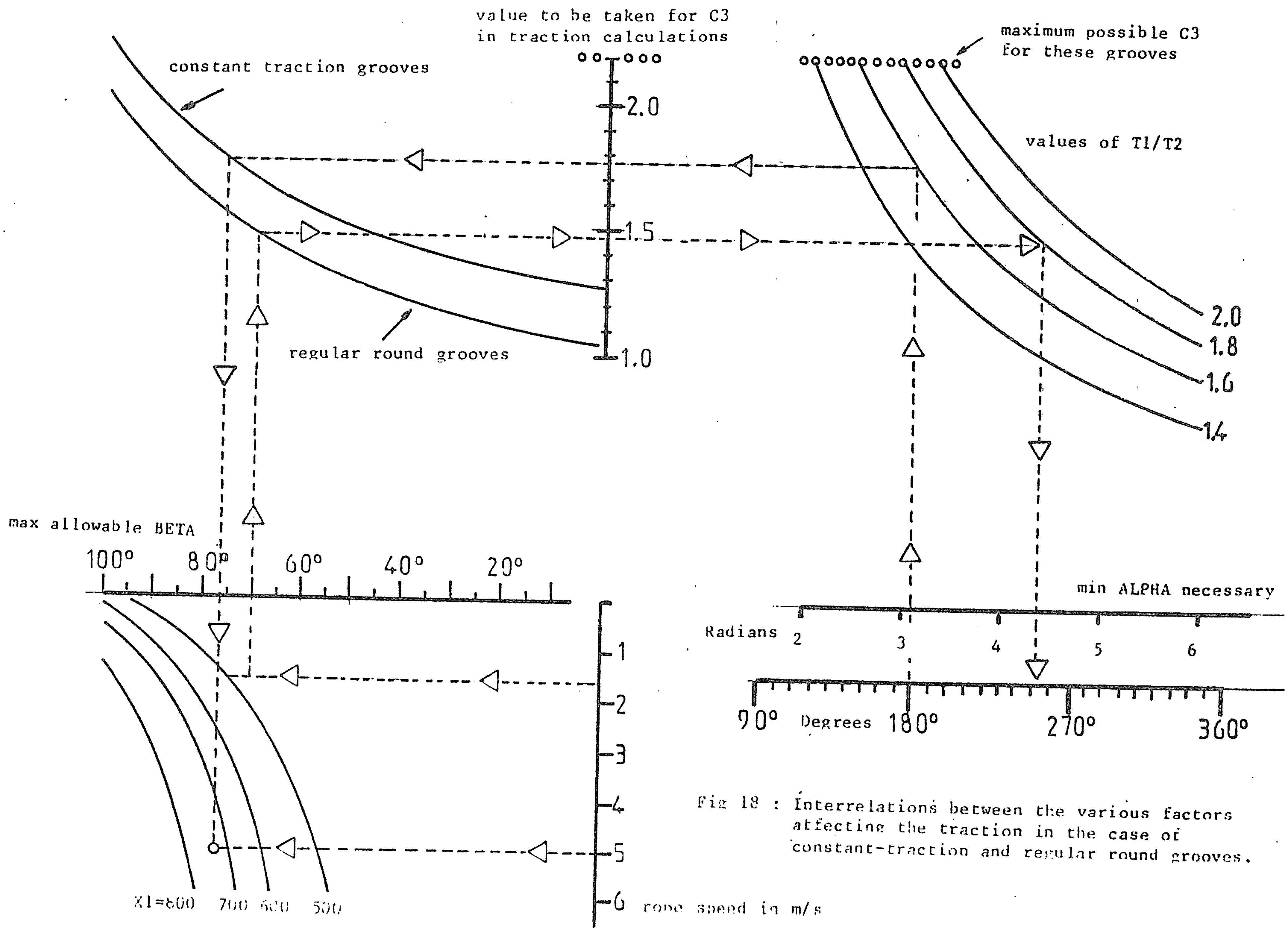


Fig 18 : Interrelations between the various factors affecting the traction in the case of constant-traction and regular round grooves.

APP:05/7 Conditions for the use of narrow V grooves

APP:05/7/a Reason for using narrow V grooves

In constant-traction grooves, the maximum value obtainable for C3 is 2.2 because the largest angle ß which may be used in undercut grooves is 100°. This correspond to a "Gamma" angle of about 52°.

For angles "Gamma" smaller than 52°, we will have to keep the undercut to the width corresponding to 105° whatever the "Gamma" angle. This means that, if we want to continue using for our traction calculation the lowest expected value for C3 we would be stuck to this 2.2 value and there would be no advantage in using "Gamma" angles smaller than 52°.

(see Fig 16 and Fig 17)

There are however cases where we would like to use a wrapping angle smaller than 180° for keeping the cost of the lift down.
This is specially the case in low cost appartment housing.

The only way to have a C3 value exceeding 2.2 is to compromise on the safety principle which was the base for our reasoning so far.

Let us assume that in a low speed, low activity lift the frequency of inspections will be relatively high enough to detect the signs of wear in the groove from the very beginning and that instructions will be given for regrooving the sheave as soon as the first signs do appear.

We may then consider that an high original C3 will not drop as low as 2.2.

The early signs of groove wear cannot be detected by watching the sinking because the sinking due to wear will be combined with the sinking due to the loss of diameter due to stretching.
The inspector will have to inspect visually the flanks of the grooves but when he sees something, it means that there was some sinking due to wear and, for calculation purposes, we will assume that the latter does not exceed 0.125*d.

The maximum allowable undercut should in any event be provided as ultimate residual safety and this means that, in Fig 17, the C3 would drop from the initial value of curve 1 to either the value of curve 3 or the value of curve 4 whichever is higher.

Analyzing Fig 17, one can see that there is no point in using "Gamma's" between about 35° and 50° because the minimum to be used for calculations will be 2.2 and it is as well to take advantage of the constant traction groove having this large undercut.

Experience has shown that 30° is about the smallest "Gamma" which can be used without running the risk of squeezing the rope in the groove.

So 35° is about the only angle which can give a small advantage.

APP:05/7/b Relation between the wedging factor C3 and the wrapping angle

All the reasoning and formulae developed in APP:05/6/a do apply here.
In this case, you have to look at the N.E. quadrant of Fig 19.

APP:05/7/c Relation between the wedging factor C3 and the angle "gamma"

Looking at Fig 15, you see that to each value of "gamma" in the first column, there corresponds a value for C3, after sinking of 0.125*d, in the third column.

This is illustrated in the N-W quadrant of Fig 19.

APP:05/7/d Relation between the rope speed and the angle "gamma"

Using developments similar to the ones of APP:05/6/c, you will find, for the single tensile wires rope:

$$p = \frac{519*d2}{(\text{safety factor})*d*(\text{ratio } D/d)*d} * \frac{4.5}{\text{Sin}(\text{gamma}/2)}$$

Calling X1 the product (safety factor)*(ratio D/d), and after simplifications, this will read:

$$p = \frac{2336}{X1 * \text{Sin}(\text{gamma}/2)} \qquad (N/mm2)$$

Let us remark that what we could call "pressure factor" is here equal to 4.5*C3 whereas, for round and constant pressure grooves it varies from 2*C3 for the full round grooves to more than 5*C3 for the large undercuts. The results are not directly comparable but they are in the same order of magnitude and the same limits may be used.

The results are illustrated in the S.W. quadrant of Fig 19

For dual tensile ropes also use X1=1.18*(safety factor)*(ratio D/d)

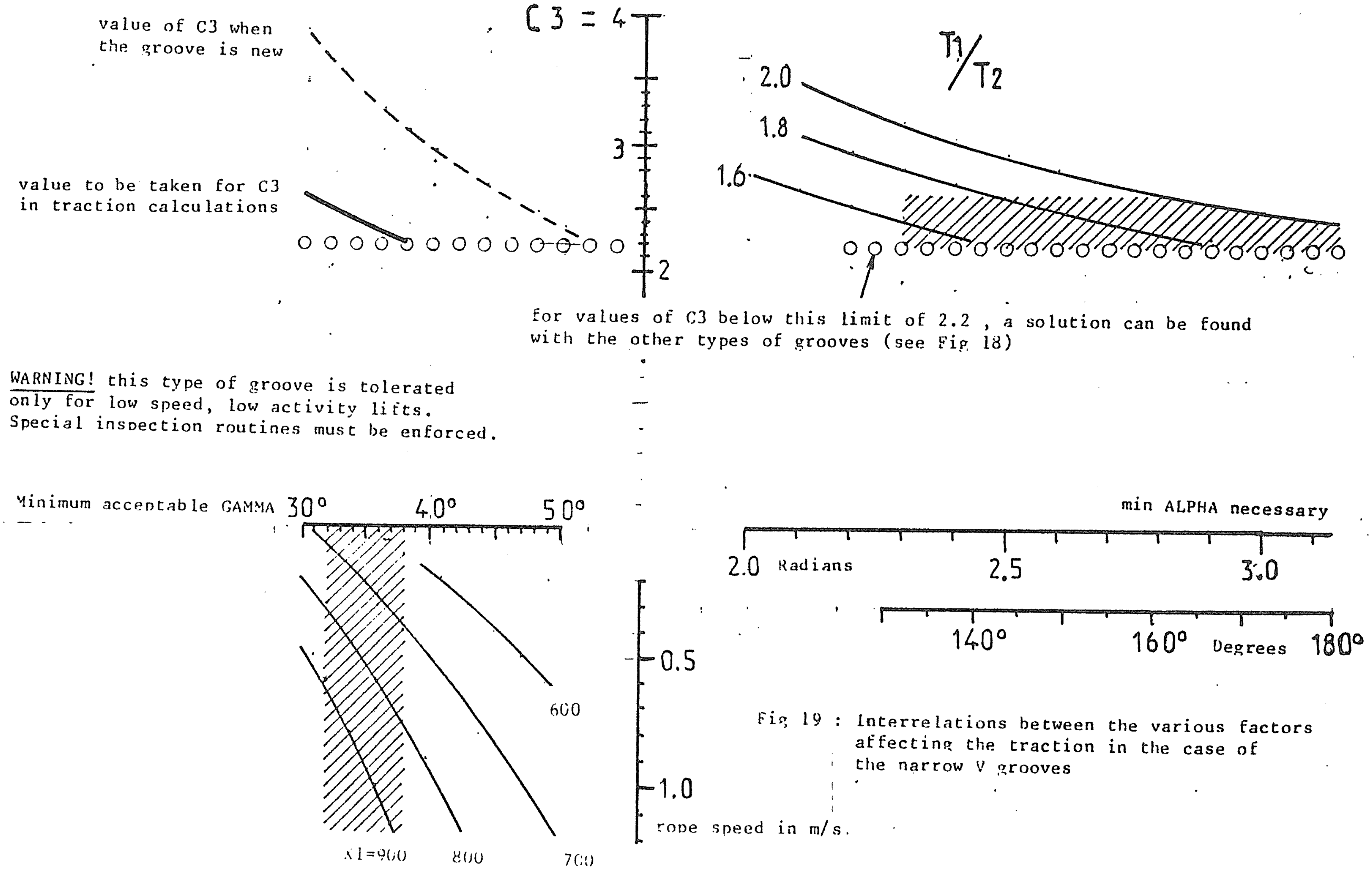
APP:05/7/e Directions for using the diagrams of Fig 19

The approach is the same as for Fig 18.

Again it appears that the only justified applications would be in the shaded areas and this calls for the following remarks:

- for speeds between 0.5 and 1.2 meters, the value of X1 will be 800 to 900 which means that a high safety factor and/or a high ratio D/d will be needed for meeting the code requirements,
- if T1/T2 is below 1.5, there is a solution by using the constant-traction grooves (see Fig 18).

This type of groove should be tolerated only for low speed and low activity lifts; constant-traction grooves should be preferred whenever possible.



WARNING! this type of groove is tolerated only for low speed, low activity lifts. Special inspection routines must be enforced.

Fig 19 : Interrelations between the various factors affecting the traction in the case of the narrow V grooves

APP:05/8 Recommendations for the design of grooves

APP:05/8/a

Let us first recall that:

- all grooves should be able to accomodate ropes having a diameter 5% over nominal before being under tension,
- for all normal ISO ropes, the diameter will drop to around 0.95% of nominal after stretching.

All the following illustrations show the rope having its diameter already reduced to 95% of nominal whereas the groove is still intact as if it had not time to wear at all. It is a little schematic but illustrates nevertheless very well the pro's and con's of each design.

We will, in the following paragraphs, give those pro's and con's for each of the possible designs, and qualify them as:
RECOMMENDED, ACCEPTABLE, TOLERATED, NOT RECOMMENDED

APP:05/8/b Conventional semi-circular grooves

The conventional undercut semi-circular groove is not recommended because, as shown in Fig 20/b, in the beginning of its life the rope rests on the edges of the undercut, which is not good for the rope and reduces the initial wedging factor C3 (see tabulation Fig 14)
On the other hand, if the shrinking of the rope is not taken into consideration for calculating the width of the undercut, the value of the wedging factor C3 for the worn groove will be higher than expected,especially for wide undercuts.

In full round grooves, the seating, although not as good as hoped for, is not bad (see Fig 20/a). The initial wedging factor (1.03) is considerably lower than the final one (1.26) but, in general, this type of groove is used in conjunction with the double wrap arrangment and the value of the wrapping angle is so high that the low C3 is good enough.
The advantage of the full round groove is that the tooling is a little simpler than for the constant traction groove illustrated Fig 22/a.

Fig 20/a ACCEPTABLE

Radius of groove	:1.06*d
Width of undercut	:0
Initial C3	:1.03
Final C3	:1.26

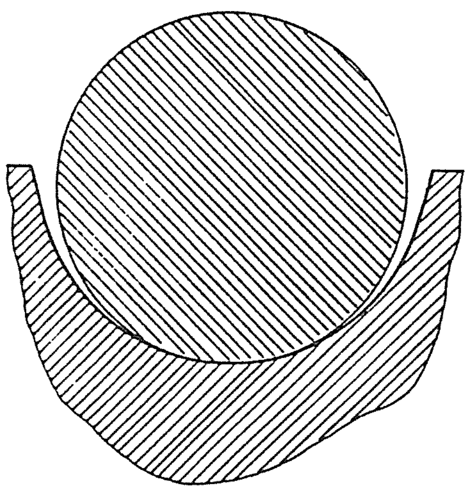
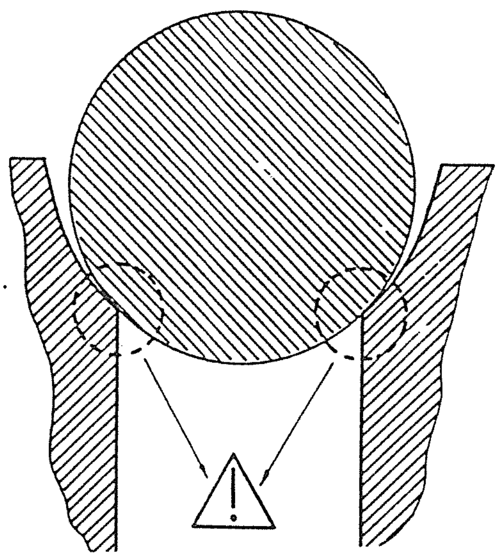


Fig 20/b NOT RECOMMENDED

Radius of groove	:1.06*d
Width of undercut	see Fig 14
Initial C3	" " "
Final C3	" " "



APP:05/8/c Simple "constant traction" grooves

As explained in APP:05/3/c and APP:05/5/c, the wedging factor of a V groove can be kept nearly constant if the angle of the undercut "Beta" is selected as a function of the V angle "Gamma", at least for "Gamma's" between 52° minimum and 105° maximum.

The advantages of this type of groove are:

- wedging factor C3 nearly constant
- insensitive to the tolerances on the rope diamenter (*)
- machining tools easy to make and easy to maintain.

(* all the ropes on the same lift must of course have the same diameter)

From the traction point of view, this groove is as good as any.
The disadvantage is that inspectors will probably be reluctant to use the formula based on the angle "Beta" for calculating the pressure instead of the formula proposed for V grooves.
This is disturbing only for Gamma's larger than 65° (see next parag.), and, for these cases, one must admit that the design proposed in APP:05/8/d (below) is undoubtly superior.

Fig 21/a ACCEPTABLE

"Gamma angle"	:105°
Width of undercut	: 0
Value of C3	:1.26

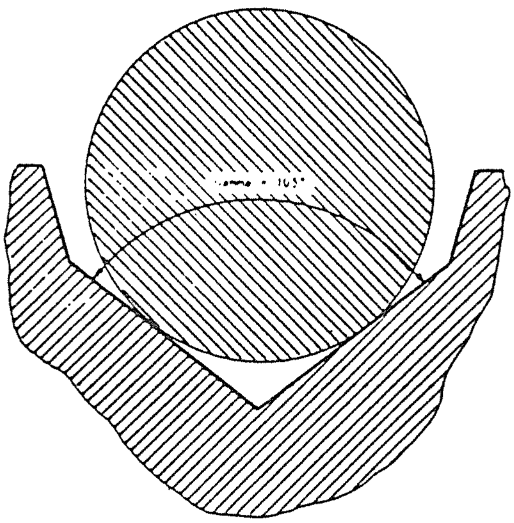
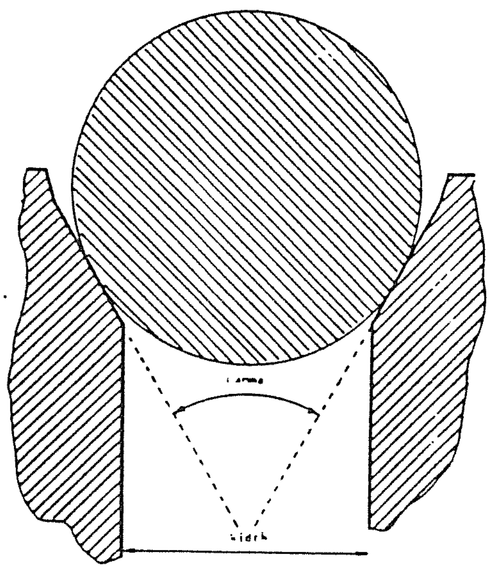


Fig 21/b ACCEPTABLE

"Gamma" angle, Width of undercut, and value of constant C3: see all legitimate combinations in the upper tabulation of Fig 16



APP:05/8/d Recommended "constant traction" grooves

Whilst retaining the principles of the simple constant traction groove above, it is modified in such a way that the application of the formula based on the angle "Beta" for calculating the pressure cannot reasonably be questioned by Inspectors.

Let us here remark that, for "Gamma's" between 50° and 65°, the V grooves formula gives pressures lower than the other formula (this corresponds to "Beta's" ranging from 100° to about 80°).

It is only for the very large "Gamma's" (> 65°) that the choice of the formula becomes important.

The advantages of this type of groove are:

- Wedging factor C3 nearly constant,
- Insensitive to the tolerances on the rope diameter(*)
- indisputable application of the formula based on "Beta" for calculating the unit pressure.

(* all the ropes on the same lift must of course have the same diameter)

The only disadvantage is that the tools for machining the grooves are a little more complicated but this cost is negligible as compared to the advantages. If this was really a problem, you could use the design of Fig 21/b for "Gamma's" ranging from 50° to 65° (see remark in the preceeding paragraph). For the Fig 22/b, the Gamma=60° with an undercut=68° of "d" was taken as example; you will appreciate how little it differs from Fig 21/b (especially in the active part of the groove).

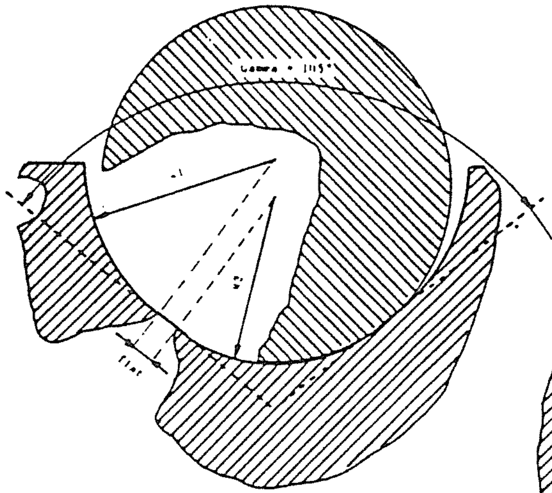
Fig 22/a <u>RECOMMENDED</u>

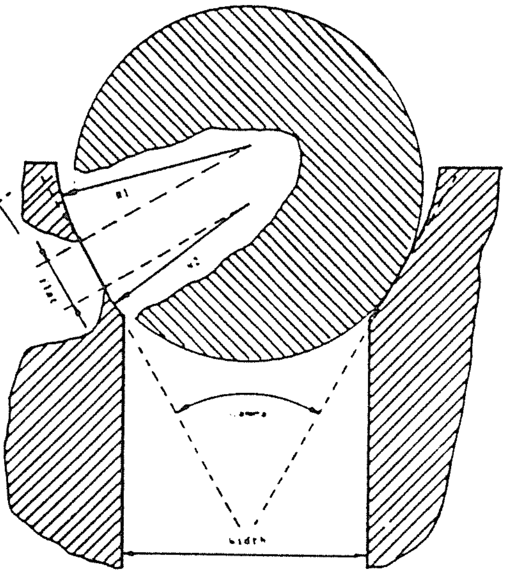
R1	= 0.53*d minimum
R2	= 0.45*d maximum
flat	= (R1-R2)/Tang(52.5°)
Gamma	= 105°
Width	= 0

FIG 22/B <u>RECOMMENDED</u>

R1	= 0.53*d minimum
R2	= 0.45*d maximum
flat	=(R1-R2)/Tang(Gamma/2)

Gamma and Width:
see Fig 16 (upper table)





<u>APP:05/8/e</u> Narrow V grooves tolerated under certain conditions.

The danger with this type of groove is that it is impossible to make an undercut wide enough to prevent the drop of the C3 value when the groove wears and allows the rope to sink.

So, instead of allowing the rope to sink the equivalent of 0.5*d as is possible with the constant C3 grooves, the traction sheave should be regrooved as soon as the first signs of wear are detected.

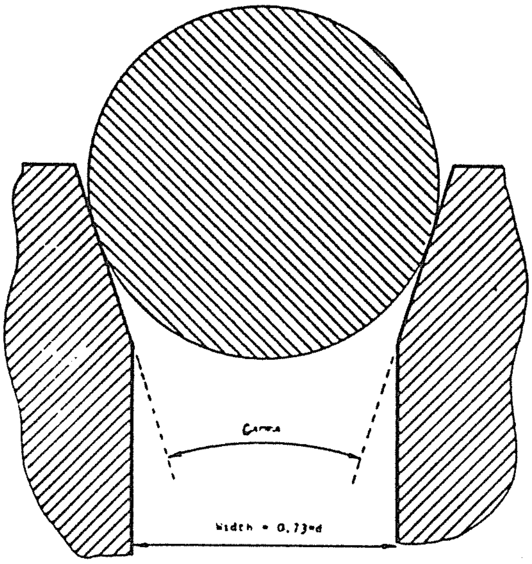
Because of the interval between inspections, the use of this type of groove should logically be limited to low activity lifts. Special instructions should be given to the inspectors for reporting the first signs of wear.

Because of the pressure limitations, they are, in any event, restricted to low rope speeds (1 to 1.2m/s)

As an ultimate protection in case of undetected wear, an undercut of 0.73*d should be provided in all cases (this corresponds to the widest acceptable Beta angle i.e. 105°). A C3 of 2.28 is guaranteed by this undercut. This is lower than the minimum used for the traction calculation but, combined with the safety factor, could save the situation in many cases.

For the reasons explained in APP:05/7, only angles ranging from 30° to 40° do make sense if narrow V's have to be used and the only reason for wanting to use them is that they can manage with wrapping angles smaller than 180° which results in less expensive lifts.

Fig 23 <u>TOLERATED under certain conditions</u>



30° < Gamma < 40°
For the traction calculations:
   min value of C3 see 3d column of Fig 15
   max value of C3 see 2d column of Fig 15

Width of undercut : 0.73*d
Ultimate value of C3
   in case of mishap: 2.28

Conditions of use:
- low speed
- low activity
- regrooving at first sign of wear

TRACTION CALCULATIONS

Guide-lines, Formulae and Check-list

APP:06/1 Data to be collected

APP:06/1/a Intangible data

There is a limited number of data which you cannot touch because they are in the lift contract, namely:

- contract speed	v	m/s
- travel	H	m
- mass of rated load	Q	kg

APP:06/1/b Semi-fixed data

To a certain extent, the other data can be selected (between limits) Of course, based on your experience, you will start with a tentative design but, to meet the traction requirements, you might have to adapt some of the following data:

- maximum acceleration of the car	a	m/s2
- rope speed (depends on the reeving)	Vc	m/s
- diameter of traction sheave	D	mm
- diameter of hoisting ropes	d	mm
- number of hoisting ropes	n	
- mass of empty car with sling, doors etc	P1	kg
- mass of travelling cable	P2	kg
- mass of compensating ropes (or chains) if any	P3	kg
- mass of compensator	P4	kg
- mass of the hoisting ropes	P5	kg
- mass of the counterweight	CT	kg

APP:06/2 Rough check of the adequacy of the tentative design

APP:06/2/a Pre-calculations

Using the data you have collected, calculate the following values which will be needed for using the diagrams Fig 18 and Fig 19.

$$n*T1 = (P1 + 1.25*Q + P5)*(9.81 + a) + 0.5*P4*9.81$$

$$n*T2 = (CT + P3)*(9.81 - a) + 0.5*P4*9.81$$

Using these two, you can get T1/T2

$$X1 = (D/d)*(\text{ropes safety factor})$$

APP:06/2/c Pre-check of the adequacy of the tentative design

You should always start by trying a solution with the constant-traction grooves and for that you go to Fig 18.

Starting from the rope speed point on the South axis of Fig 18, you go horizontally left to the curve corresponding to the value you have calculated for X1.

Then, going up vertically to the curve "constant traction grooves", you find the maximum allowable "beta" angle on the the West axis and you can read the value of C3 to take into account for the traction calculations on the North axis.

You keep going horizontally to the right until arrive at the curve corresponding to the value you have calculated for T1/T2. Then you can read on the East axis the minimum value required for the wrapping angle "Gamma".

If this minimum value is higher than the one provided for in your tentative design, you will have to go back to square one after playing around with the semi-fixed data.

You have to decide first which ones are the easiest to modify in your case.

You could, for example, increase the value of X1:
- by increasing the number of hoisting ropes,
- by increasing the diameter of the ropes,
- by increasing the diameter of the sheave,

You could also decrease the value of T1/T2:
- by adding masses both to P1 and to CT,
- by adding compensating ropes if there were none,
- by adding a compensator if there were none.
- by decreasing the acceleration "a" (*)

(* Note: it is not always possible and, because of the mechanical brake, it would be unreasonable to take less than 1 m/s)

If, doing this, you cannot find a suitable solution, you could increase the wrapping angle of your tentative design going eventually to the double wrap if necessary. In the case of low speed, low activity lifts, you could decide to try the narrow V grooves; in that case see APP:06/2/d.

Coming back to the constant traction grooves, if you have a double-wrap arrangment with an ample reserve in the wrapping angle, you might want to try using the conventional full-round grooves. In that case, go back to Fig 18 but use the curve "regular round grooves" in the N.E. quadrant.

Once you have identified a suitable combination go to step APP:06/3.
If you have the choice between several combinations, remember that the one leading to the lowest pressure is the best.

APP:06/2/d Pre-check for the application of narrow V grooves.

The use of constant-traction grooves should always be preferred, not only because the traction is guaranteed even after the wearing of the groove, but also because they lead to lower pressures.

The use of narrow V grooves may be contemplated only if:
- the rope speed does not exceed 1.2 m/s,
- the activity of the lift will be low,
- special inspection rules can be enforced,
- you do not want to increase the wrapping angle.

For narrow V grooves, use Fig 19 in the same way as described for Fig 18 above. Once you have found an acceptable combination go to step APP:06/3.

APP:06/3 Accurate calculations for the selected arrangment

APP:06/3/a Review of the conditions to be verified.

After going through the proceedings of APP:06/2, you have identified an arrangment which has every chance of being right and have now selected values for all the semi-fixed data listed in APP:06/1/b.

Using the complete formulae, you must now check that:

- you have enough traction for accelerating UP (or braking DOWN) the overloaded car at the lowest landing,
- you have enough traction for accelerating UP (or braking DOWN) the counterweight with the empty car at the highest landing,
- you do not have so much traction that you would keep lifting the empty car when the counterweight rests on the buffer(s),
- the unit pressure in the groove is below the required limit.

We can already calculate this required limit with the formula

$$(p\ limit) = (12.5 + 4*Vc)/(1+Vc)$$

We will, successively for each condition, review the appropriate formula and where to find the value to use for the wedging factor C3.

APP:06/3/b Traction with overloaded car at lowest landing.

The following inequation must be verified

$$T1/T2 < 0.8 * e^{(C3*0.125*alpha)}$$

where:
- $n*T1 = (P1 + 1.25*Q + P5)*(9.81 + a) + (0.5*P4)*9.81$
- $n*T2 = (CT + P3)*(9.81 - a) + (0.5*P4)*9.81$
- 0.8 = safety factor on the traction limit
- 0.125 = friction factor at slipping limit
- alpha = wrapping angle in Radians
- C3 = lowest wedging factor during considered lifetime.

If you choosed to use a constant-traction groove, the Fig 18 gave you the maximum allowable "beta". Using the 2 columns to the left of Fig 16 you can select the nearest suitable "gamma" angle and, going to the last column to the right, you will find the value to be used for C3.

If you choosed to use a conventional full-round groove the value to be used is C3 = 1.03.
(see 1st line, 2d column Fig 14. The other lines have no application)

If you choosed to use a narrow V groove you may use the value corresponding to the selected "gamma" in the 3d column of Fig 15.

> Note:
> Since we decided, for the narrow V grooves, to provi e the widest acceptable undercut (0.73*d) as ultimate protection in case of mishap, remember that, as soon as this C3 value drops under 2.2, you will be better off using the constant-traction groove based on 55° "gamma". This is a safer solution.

APP:06/3/c Traction of counterweight with empty car at highest landing.

The following inequation must be verified

$$T1/T2 < 0.8 * e^{(C3*0.125*alpha)}$$

where: n*T1 = (CT + P5)*(9.81 + a) + (0.5*P4)*9.81
n*T2 = (P1 + P2 + P3)*(9.81 -a) + (0.5*P4)*9.81
0.8 = safety factor on the traction limit
0.125 = friction factor at slipping limit.
alpha = wrapping angle in Radians
C3 = lowest wedging factor during considered lifetime.

For C3, use the same value as in APP:06/3/b.

APP:06/3/d Slipping with car empty and ctw on buffer(s)

The following inequation must be verified

$$T1/T2 > 1.25 * e^{(C3*0.125*alpha)}$$

where: n*T1 = (P1 + P2 + P3)*9.81 + (0.5*P4)*9.81
n*T2 = P5*9.81
1.25 = safety factor on the traction limit
0.125 = friction factor at slipping limit
C3 = highest wedging factor during considered lifetime

If you choosed to use a constant traction groove, you should use the same C3 value as in APP:06/3/b.

If you choosed a conventional full-round groove, use C3 = 1.26.
(see 1st line, 4th column Fig 14)

If you choosed to use a narrow V groove, you should use the value corresponding to the selected "gamma" in the 2d column of Fig 15.

APP:06/3/e Pressure in the groove below the required limit.

You must first calculate this required limit:

$$(p\ limit) = (12.5 + 4*Vc)/(1+Vc)$$

then calculate the pressure in the groove "p" and compare it to the limit.

The formula for calculating the pressure in the groove depends on the selected type of groove.

APP:06/3/e/1
The formula

$$p = \frac{T}{n*d*D} * \frac{8*Cos(beta/2)}{3.14 - beta - Sin(beta)}$$

has to be used in the following cases:
- full-round grooves illustrated Fig 20/a.
- recommended constant-traction grooves illustrated Fig 22/a and 22/b.

The formula

$$p = \frac{T}{n*d*D} * \frac{4.5}{\mathrm{Sin(gamma/2)}}$$

has to be used in the following cases:

- acceptable constant-traction grooves illustrated Fig 21/a and 21/b.
- tolerated narrow V grooves illustrated Fig 23

It should be noted that the pressure, calculated by this formula for the simple design of the constant-traction grooves, will be more favorable than the pressure calculated by the formula APP:06/3/e/1 for the sophisticated design of Fig 22/b for all "beta" angles greater than 80° (corresponding to "gamma's" ranging from 55° to 65°).
This is because the two formulae, although giving values commensurate enough to be compared to the same limit, cannot be used to compare one design with another.
It should be remembered that these values are conventional.

APP:07/1 Critique of the limitation of the mean deceleration to 1 Gn

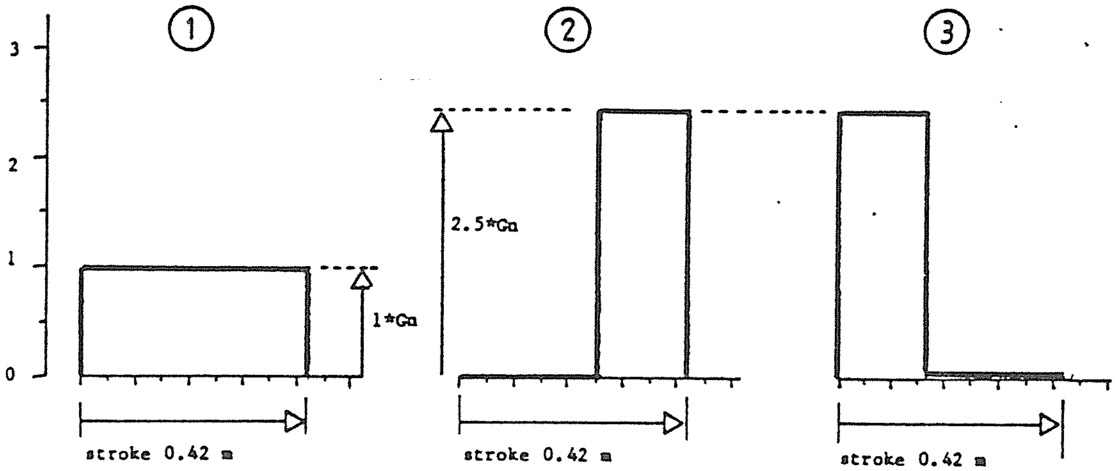
Let us review what happens, for a given load (let us say the contract load), with 3 possible hydraulic buffers, N°1, N°2 and N°3, designed for an impact speed of 2.875 m/s (lift rated speed of 2.5 m/s).

It is possible to make buffers very close to these if the action of the return spring can be neglected.

For all the 3 buffers in Fig 1, the stroke of 0.421 m (gravity distance) has been selected. This is the minimum required by the code.
The mean deceleration based on distance is of course 1 Gn for all 3 of them but it is so obvious that it cannot be the one meant by the code.

Their behaviour during operation is illustrated Fig 1.

Fig 1 of APP:07



The buffer N°1 has a constant deceleration of 1 Gn along the stroke.
The mean deceleration based on time is of course 1 Gn which is the upper limit accepted by the code although this buffer would be ideal!

The buffer N°2 does not decelerate at all during the first 0.253 m of the stroke then decelerates with a constant 2.5*Gn (maximum allowed by the code) during the remaining 0.168 m.
The mean deceleration based on time is 1.428*Gn and such a buffer is not acceptable by the code.

The buffer N°3, on the contrary, decelerates with the maximum allowed 2.5*Gn during the first 0.168 m of the stroke, then with only 0.06 m/s2 (0.0063*Gn) during the remaining 0.253 m.
From the passenger point of view, this buffer is neither better nor worse than the buffer N°2 but, the mean deceleration being in the present case only 0.1*Gn it seems to be an excellent buffer from the code point of view!

This is not logical.

APP:07/2: Ideal buffer

This requirement of the code prevents the design of what I would consider to be the ideal buffer i.e a buffer which would have:

- a constant deceleration of 1*Gn with 50% of the rated load,
- a moderate peak at the beginning of the stroke with empty car,
- a moderate peak at the end of the stroke with full load.

Using, for the typical lift mentioned in HB/FWD/12, the method outlined at the end of Appendix APP:04, you will find the values illustrated in Fig 2. The <u>instantaneous</u> peak decelerations would be very moderate (around 1.6*GN) but such a buffer is unacceptable from the code point of view because, under full load, the mean deceleration exceeds 1*Gn.

(Note: remember that the calculations can only give an idea of the buffer performances. Actual tests are needed to finalize the design)

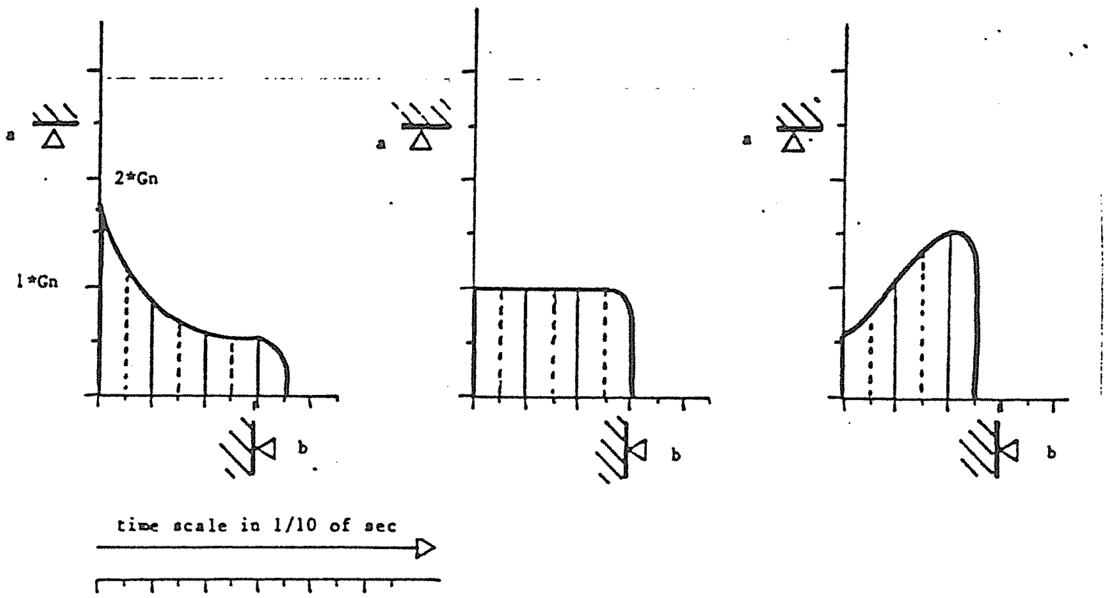


Fig 2 of APP:07

a = deceleration which shall not be exceeded for more than 0.04 sec
b = minimum time of operation for having a mean deceleration < 1 Gn

APP:07/3: Actual buffers based on the present code requirements

It is to be remarked that the only thing to do in order to have the mean deceleration below 1*Gn, is to design the buffer so that it will take more than the corresponding time to complete the stroke.

This time is easy to calculate. The deceleration being constant, the average speed is of course half the impact speed and, to get the time, you just divide the gravity distance by the average speed.

For the impact speed of 2.785 m/s, the gravity distance is 0.4213 m. The average speed is 1.3925 m/s and the time needed to slow down to zero speed with a deceleration of 1*Gn is 0.2956 second.

So the trick is to make sure that the buffer drags at low speed at the end of its stroke whatever happens at the beginning of the operation.

With the present code requirements, buffers can be designed having a range of applications much wider than the ideal one illustrated APP:07/2 Fig 2.

For the same impact speed of 2.785 m/s, the Fig 3 illustrates the behaviour of a typical buffer for test masses ranging from 1000 kg to 5000 kg.

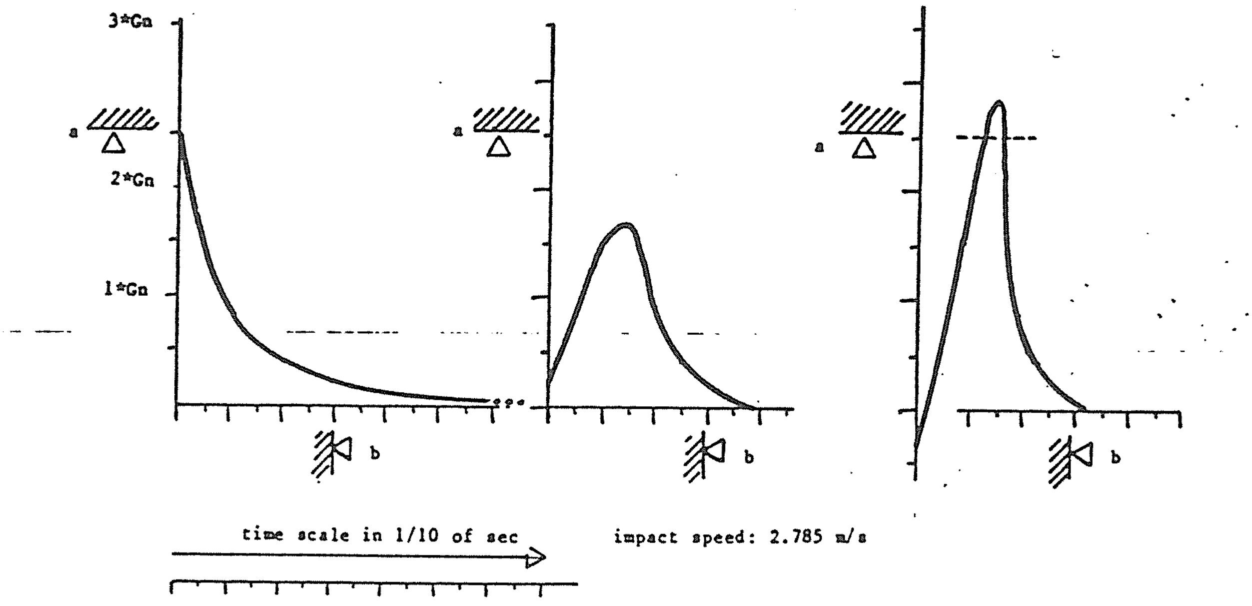


Fig 3 of APP:07

a = deceleration which shall not be exceeded for more than 0.04 sec
b = minimum time of operation for having a mean deceleration < 1 Gn

You will notice that, whatever the load, everything happens within the first 0.2 sec but, nevertheless, the peak is at the beginning of the stroke with the minimum load and much further down with the maximum load.

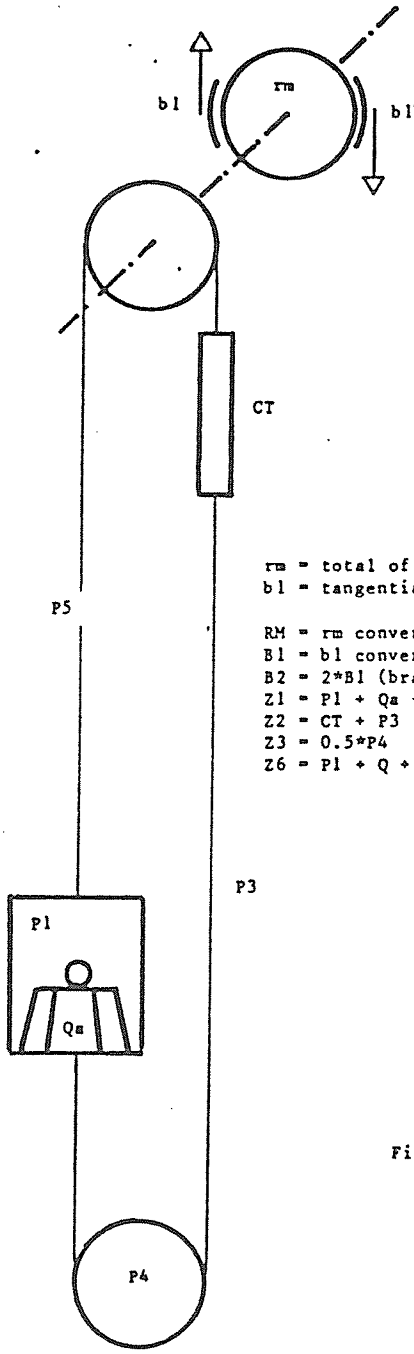
BRAKING CALCULATIONS

APP:08/1 Mathematical approach

The approach is basically the same as the one developed in APP:03/HB for analyzing the behaviour of buffers and safety gears.

However, in APP:03/HB, we neglected the forces due to traction and all rotating masses in the machines because the decelerations due to buffers and safety gears were so high (1 Gn or more) that ropes were surely sliding and the residual traction was disregarded.

Here, to the contrary, the aim is either to avoid the slipping of the ropes or to analyze the behaviour of the system if slipping cannot be avoided.



rm = total of the rotating masses
b1 = tangential force exerted by 1 brake shoe

RM = rm converted to translation
B1 = b1 converted to translation
B2 = 2*B1 (brake force with 2 shoes)
Z1 = P1 + Qa + P5
Z2 = CT + P3
Z3 = 0.5*P4
Z6 = P1 + Q + P5

Fig 1 de APP:08/HB

To the masses listed in APP:03/HB we must add the mass of the rotating parts of the machine and of the motor; but that mass is not subject to gravity. The external force coming from the buffer or from the safety gear is replaced by the external force coming from the braking system.
All the masses are moving jointly as long as the traction limit is not exceeded but when this limit is exceeded, the movement of the rotating mass must be analyzed separately.

Some new symbols are introduced because of the new items coming in the picture and also for simplifying the formulas by identifying sub-assemblies of masses of frequent recurrence.

rm	mass of the rotating parts of the machine and motor.
RM	rm converted to translation
b1	tangential force exerted by one brake shoe
b2	tangential force exerted by the two brake shoes
B1	b1 converted to translation
B2	b2 converted to translation
w	rotating speed of rm in Rad/s
Rg	radius of giration of rm

Z1	= (P1 + Qa + P5) travelling mass on the car side
Z2	= (CT + P3) travelling mass on the counterweight side
Z3	= 0.5*P4 used only in the traction calculations
Z4	= (P1 + P5) = value of Z1 for Qa equal to zero
Z5	= (P1 + 1.25*Q + P5) = value of Z1 for 25% overload
Z6	= (P1 + Q + P5) = value of Z1 for contract load

For the mathematical developments, the rotating mass is converted to translation, and the following values relate to translation.
When the ropes are not slipping:

V	= speed of all masses
G1	= deceleration with one brake shoe applied
G2	= deceleration with the two brake shoes applied
Gtr	= deceleration limit for the traction conditions
H	= distance covered during the analysis

When the ropes are slipping

Vc	= translation speed of all suspended masses
Vr	= "translation" speed of the rotating mass
Gc	= acceleration of the suspended masses
Gr	= acceleration of the rotating mass converted to translation
Hc	= distance covered by the suspended masses
Hr	= "translation" distance of the rotating mass

(NOTE: rope slipping is analyzed only with the 2 shoes operating)

APP:08/2 <u>Conversions from rotation to translation and vice-versa</u>

APP:08/2/a: Conversion of masses.
The combination with the respective linear speeds must be such that the kinetic energy remains the same.
The ratio of rm to RM is then inverse to the ratio of the square of (w*Rg) to the square of V:

rm*(square of w*Rg) = RM*(square of V)   (01)

APP:08/2/b: Conversion of forces

The combination with the respective linear speeds must be such that the power remains the same.

$$b2*(w*Rg) = B2*(V) \qquad (02)$$

APP:08/3 <u>Relation between braking force and traction factor</u>

Basically, the braking force should be such that the deceleration of the lift system does not exceed the deceleration (or acceleration) taken in consideration for the calculation of T1 and T2 in the traction calculations (see APP:06/HB).

In fact, in the traction calculations, it is an inequation and not an equation which has to be satisfied; the margin of safety might be considerable (as is often the case with double-wrap arrangments).

It is best to recalculate the maximum value of the deceleration which could be accepted by the existing combination of data (wrapping angle, type of groove, exact values of the intervening masses).

So, instead of the inequation mentioned in APP:06/3/b, you write it as an equation where the unknown quantity is (a), the acceleration.

You will, doing so, find the value of Gtr.

APP:08/4 <u>Calculation of the brake force needed</u>

APP:08/4/a: Gearless and geared machines with continuous speed control

In these 2 cases, the value of rm is fixed: this is the unavoidable minimum consisting of the masses of the motor rotor, the coupling (if any), the hand wheel (if any), the brake drum and the traction sheave. It is kept as low as possible to reduce the power and the starting torque required.

APP:08/4/a.1: Calculation of the B2 leading to a deceleration G2 = Gtr

The deceleration produced by the 2 brake shoes working toghether (G2) should ofcourse not exceed the Gtr calculated above if we want to avoid slipping of the hoisting ropes.

So the resultant of all forces acting on the lift system shall not exceed:

$$F\ resultant = G2 * (Z5 + Z2 + RM) \qquad (03)$$

Analyzing the component forces:

$$F\ resultant = B2 - Z5*9.81 + Z2*9.81 \qquad (04)$$

Combining the equations (03) and (04) you will find B2.
(NB: Z5 is used here because, in the code, braking with 2 shoes is related to 25% overload in the car).

APP:08/4/a.2: Calculation of the deceleration with 1 shoe and rated load

We will assume that the 2 shoes are braking with the same force:

$$B1 = 0.5 * B2 \qquad (05)$$

$$F\ resultant = B1 - Z6*9.81 + Z2*9.81 \qquad (06)$$

Having also:

$$F\ resultant = G1 * (Z6 + Z2 + RM) \qquad (07)$$

you can, combining (06) and (07) find the deceleration G1 with one shoe. (NB: Z6 is used here because the code relates the single shoe braking to the full load only)

The code sets no minimum for this G1, but if, in your opinion, it is too low to safely stop the lift, then you have to select the lowest acceptable value, calculate backwards the corresponding value for B1, B2 and G2. Considering then Gtr = G2, introduce this new value in your traction calculations and find new values for C3 (by changing the grooves) or for the wrapping angle (by changing the machine room arrangment).

APP:08/4/a.3:
As can be seen in HB/Fig 24 and 25, problems should be anticipated only with gearless machines and probably more with the 2/1 roping where the wrapping angle is generally 180° whereas in the 1/1 roping we have normally a double-wrap arrangment with a comfortable wrapping angle.

APP:08/4/b: Case of the geared machine with a simple 1 speed AC motor.

In this case, the value of rm is not fixed: you adjust it to the need by selecting the proper flywheel.

The combination "flywheel + mechanical brake" is selected so as to:
- limit the deceleration to a value acceptable both for the comfort of passengers and for avoiding rope slipping,
- achieve the required stopping accuracy.

APP:08/4/b.1:
Before starting the calculations, you should select:
a) the deceleration:
quite naturally, you should select the value which was used for the traction calculations (see APP:06/HB)
This deceleration (which is the one we decided to call G2) was defined with 25% overload and the 2 brake shoes working equally.

b) the stopping accuracy:
Let us call "stopping accuracy" the difference between:
- the distance covered during the braking in the down direction with the rated load in the car and
- the distance covered, also in the down direction, during the braking with an empty car.
This stopping accuracy will be designated by: StAcc.

APP:08/4/b.2:
You must also know one of the essential characteristic of the A.C. motor: the slip under load.

Indeed, when going down with the rated load in the car, the speed will practically be the synchronous speed whereas, when lifting the counterweight, the speed will be 5 to 15% lower.

Let us say that this reduced speed will be:

$$(\text{synchronous speed}) * (1 - MS1)$$

where MS1 is the motor slip.

APP:08/4/b.3:
Let us first calculate the distance needed for braking with rated load. We have all the necessary data.

$$\text{H full load} = \frac{\text{V. squared}}{2 * G2} \qquad (08)$$

Since we have selected the stopping accuracy, the distance for braking with an empty car will be:

$$(\text{H no load}) = (\text{H full load}) - (\text{StAcc}) \qquad (09)$$

Because we start from a lower speed, this same distance can be expressed:

$$(\text{H no load}) = \frac{\text{square of } (V - V*MS1)}{2 * G4} \qquad (10)$$

if we call "G4" the deceleration in this case.

Combining (09) and (10), we can find G4.

APP:08/4/b.4
Let us now apply the good old formula:

(resultant force) = (total mass) * (acceleration)

successively in the 2 cases "full load" and "no load".

For the full load:

$$B2 + Z2*9.81 - Z6*9.81 = G2*(Z6 + Z2 + RM) \qquad (11)$$

For the empty car:

$$B2 + Z2*9.81 - Z4*9.81 = G4*(Z4 + Z2 + RM) \qquad (12)$$

We have 2 equations, (11) and (12), this enables us to calculate B2 and RM.

APP:08/4/b.5:
Just to make sure, calculate the G1 with only 1 brake shoe.
Because of the importance of RM as compared to the other masses, you will find that G1 is about the half of G2.
Since for the comfort of the ride G2 is generally selected close to 1 m/s2, G1 will be around 0.5 m/s2 and looking at HB/Fig 25, you can see that, for the low speed, it is far from the danger zone.

G4 will of course be greater than G2 but, with the counterweight pulling back, there is no problem of rope slipping.
If G4 was so high that the stop becomes rough, you might want to make the calculations again, starting with a lower G2.

APP:08/4/c: Case of the geared machine with a simple 2 speed AC motor
The approach is the same as for a 1 speed AC but using the low speed only.
Be careful about the motor slip at low speed, it might be high.

The braking distance in case of emergency will be quite long because the mechanical brake is applied at the high speed.

APP:08/4/d: Case of the positive drives.
We have, so far, envisaged only the case of the traction machines.
For positive drives, the approach is the same with the following exception: the deceleration (G2) is not limited by any possibility of rope slipping but exclusively by passengers comfort considerations.

In the case of a drum lift, the value of CT is zero.

APP:08/5: Design of the mechanical brake

APP:08/5/a:
So far we have calculated B2 and B1 which are "translation" values.

We must now calculate b2 and b1 related to the rotating drum or disk using the formula (02).

We will select brake linings (or pads) with the highest possible friction factor (generally 0.2 to 0.3) but, above all, a friction factor independent of the rubbing speed and of the temperature.

Dividing b1 by this friction factor will give us the force needed for pressing the lining (or pad) of one of the 2 shoes perpendicularly to the surface of the drum or disk.

The force applying the lining depends on spring(s), exceptionaly on weight(s), and on the geometry of the brake mechanism.

I have no personal experience with disk brakes but I believe that the relation between the spring force and the force applying the lining is straightforward.

As regards the usual drum type brake, a few precautions are necessary to have a brake with predictable and stable performances.

a) The linings should never have too long an arc of contact with the drum.

This is particularly important if the shoe is not articulated. (see Fig 2)

Because of the tolerances, you never know if the lining is bearing on (a) or (b).
The leverage of the spring force could be quite different from what you expect and changes in the course of time.

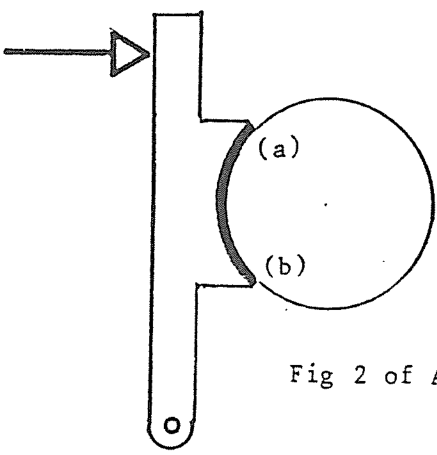


Fig 2 of APP:08/HB

In the case of Fig 3, the design is better, providing of course that the allowable specific pressure is not exceeded.

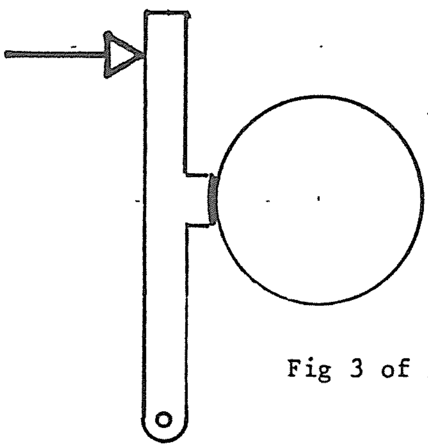


Fig 3 of APP:08/HB

b) For increasing the lining area, articulated shoes are used but some precautions are also recommended.

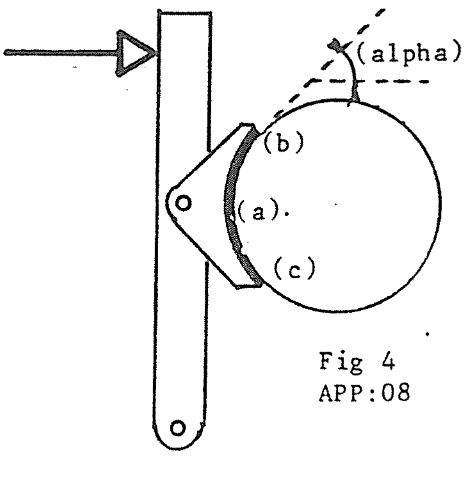


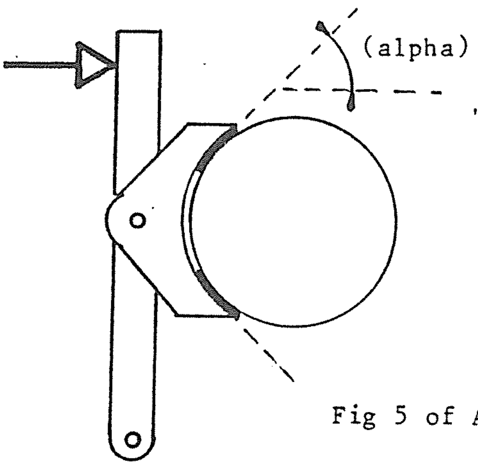
Fig 4
APP:08

With such a long arc of contact, you never know if the lining is bearing on (a) or on (b) and (c).

In the first case the perpendicular force would be (F) whereas in the second it would be: (F*leverage)/Cos(alpha).

In fact, in the first case, (b) or (c), depending on the direction of rotation, would bear at the same time as (a) for equilibrating the torque in the shoe and the calculation of the exact friction force would require taking that into consideration.

c) The Fig 5 shows a design which should be satisfactory



The force applying each individual piece of lining can not be very different from (F*leverage)/Cos(alpha) and, the articulation of the shoe being at the intersection of the tangents to the drum, there will be no torque disturbing the equal repartition of the force between the 2 linings.

Fig 5 of APP:08/HB

APP:08/6 Behaviour of the lift in case of ropes slipping

APP:08/6/a
If the brake has been adjusted harder than the calculated limit, there is a risk of the ropes slipping in the case of an emergency braking.

The risk of slipping exists as soon as:

$$T1/T2 > 0.8 * e^{(C3*0.125*alpha)}$$

The factor 0.8 was introduced, not only to have a creep low enough to avoid rapid wear in normal operation, but also to compensate:

- the fact that, in what we called "constant traction grooves", the value of C3 initial was the same as C3 final but that, in between, it could drop to 90% of this value,
- the fact that the friction factor 0.125 could drop in case of excessive lubrication.

The slipping is practically certain as soon as:

$$T1/T2 \geq e^{(C3*0.125*alpha)}$$

and for analyzing mathematicaly the variations of speeds, decelerations and travels we will consider that slipping occurs exactly at this limit.

APP:08/6/b
At the instant of origin, the deceleration of the car and of the counterweight will be the one to be introduced in the values of T1 and T2 in order to satisfy the equation

$$T1/T2 = e^{(C3*0.125*alpha)}$$

Having found the deceleration Gc of the translating masses (Z1 + Z2), you can calculate the force needed to produce this deceleration.

Substracting this force from the braking force, you have the force remaining for the deceleration of RM (i.e. Gr).

APP:08/6/c
For calculating on a computer, you have to select a time step during which we will assume that the 2 calculated decelerations are constant.
Let us call it Dt.

Let us call the speed of the car "Vc" and the speed of RM "Vr". (as usual, all the calculation are made on values converted to translation).

At the end of Dt, the speed of the car will be: Vc - Gc*Dt
the speed of RM will be : Vr - Gr*Dt

The gliding speed will be: Vg = Vc - Vr

But, as soon as the ropes start gliding in the grooves, the friction factor drops. Its instantaneous value can be expressed as a function of the gliding speed by the formula:

$$\mu = 1/(8 + Vg)$$ (see note next page)

NOTE: Vg is here a number without dimension.
This formula is only an approximation based on a limited number of tests. However, it seems to render correctly the sequence of the facts as witnessed in several gearless installations.

APP:08/6/d
The deceleration of the translating masses have now to be calculated, with this new value of μ, by the formula:

$$T1/T2 = e^{(C3*\mu*alpha)}$$

You make the same calculations as above during a new time step and you do so over and again untill the RM speed is reduced to zero.

You fix then Vr at the zero value and continue the calculations until the Vc speed is also reduced to zero.

APP:08/6/e
This calculation method has been used to draw the curves of Fig 26.

APPENDICES

TO THE EN 81 HANDBOOK

A. LEENDERS

1986

2 APP 09 to APP 16

SAFETY CRITERIA FOR POWER-OPERATED DOORS

The need to examine safety criteria for power-operated doors arises from:

- doubts raised at the 1984 ISO TC 178 meeting about the scientific basis for limitation of power-operated elevator doors' kinetic energy (or the absence of such properly researched grounds for regulation).

- the increasing importance of floor-to-floor time and its influence on the capacity of elevator banks: in other words, the conflict between floor-to-floor time, which consultants want to reduce to a minimum, and the safety of elevator riders.

- unclear wording in most elevator safety codes.

- the ability of new technology to produce door operators that achieve minimum closing times without exceeding the "safety limits" of present codes.

- concern about power-operated door accidents and the great number of liability cases in the United States.

- questions about the assumption that the force necessary to prevent power-operated doors from closing would never exceed the 30 lbf (150 N) now permitted by safety codes.

Limitation for kinetic energy

The ANSI and CEN safety codes as well as most others determine kinetic energy as 7-7.4 lbf'ft (9.5 - 10 J), calculated at an average closing speed.

When the doors are obstructed during the first two-third of their travel, the body parts caught in between are likely to be large and reasonably elastic and the reopening device has enough time to operate. When the doors squeeze a body part during the last third of travel, however, we can assume the member squeezed will be relatively inelastic. The 7-7.4 lbf'ft figure might be justified in the former case but certainly must be questioned in the latter.

According to K. Stutz, convenor for the CEN/TC10/WG1, the limitations for kinetic energy originated in the ANSI Safety Code. Nobody in Europe seems to have been willing to question their correctness even though the members of the CEN working group had been asking different organizations for their opinion. As no one had any idea about the actual effect of these energy figures on the human body, the working group accepted the use of kinetic energy values but decided to convert them into figures representing the accumulated energy of a spring with a spring constant of 142 lbf/in (25000 N/m). The spring constant was determined by measuring the elasticity of the forearm muscles of two persons. The measuring was done at 32 lbf (140 N) - 0.3 in (8 mm), corresponding to a kinetic energy of only 0.3 lbf'ft (0.4 J). It must be noted, however, that calculated kinetic energy is often considerably greater than measured kinetic energy as a result of the elasticity of the door hanger and door coupler. The more elastic this equipment is, the greater the difference.

Force limitations for power door operators

Most safety codes limit the force allowed to prevent the closing of elevator doors to 30-33.7 lbf (130-150 N), but the US safety code states that the force shall be measured on the leading edge of the door with the door at any point between one-third and two-thirds of its travel. It is not specified whether the reopening device shall also function during the first one-third and the last one-third of the travel. It is only specified where it shall be measured. The Canadian safety code does not mention this, but I have heard it is regrettably going to be changed to correspond to the US safety code. CEN mentions clearly that the only time the reopening device need not function is during the first one-third of the door travel.

It is reasonable that the reopening device need not function during the first one-third of the travel because a greater force is needed to accelerate the door to full speed. But if the reopening device does not have to function during the last one-third of the travel, a dangerous situation exists, especially if the force necessary to prevent closing of the doors is not limited. The absence of limitations during the last phase of closing may be motivated by the growing friction which can prevent the doors from closing when there are great fluctuations in air pressure in the elevator shaft and the hall. This may be the case, e.g., in northern climates in mid-winter in skyscrapers where doors from the hall to the street stand open. This problem should, however, be solved in some other way than by risking the safety of elevator passengers.

Risks combined with closing of doors

Safety codes tend to limit the force exerted on the person squeezed, but are they sufficiently effective? It is easy to provide the car door either with an electrical or electromechanical safety edge with such a large radius that it will make the door change direction before it meets an obstacle. Such a provision would make car doors safer. It should, however, be noted that the limitations for reopening devices and kinetic energy are primary safety-technical measures whereas photocells and safety edges are secondary devices whose principal objective is to control the door operation.

The biggest risk lies in getting squeezed between the hoistway doors. The cost and complexity of safety edges generally keep lift purchasers/owners from installing them on hoistway doors. No lift regulations require such precautions, either. Passengers are likely to get squeezed between the hoistway doors because they often try to prevent the doors from closing by putting some part of their body between them. The object getting squeezed may be such a relatively inelastic member as an arm, hand or fist. Another possible object may be the head of a curious child.

With high-quality doors (quick, noiseless movements) it is essential that the door coupler between car and hoistway door and the links between door gear and car door be as rigid as possible to secure smooth door operation during acceleration and deceleration. This means that the force exerted on the obstacle between the doors will grow because the elasticity of the door mechanism will decrease.

Kinetic energy/force/elasticity

The right side of diagram 1 shows the force exerted on the object (part of the body) squeezed, calculated as a function of its own elasticity in the following cases:

1. W = 5 J. Only hoistway doors are considered.

2. W = 10 J. Average total door energy permitted by codes. (The ruled area tries to show actual limit values when the elasticity of the door mechanism and door hangers are taken into consideration.)

3. W = 15 J. Maximum energy in the middle of opening when using sinusoidal operator.

The diagrams show that there is no great risk if the squeezed part of the body is very elastic and if the reopening device adjusted to the permissible force of 30 lbf functions quickly enough. The risk results from the limited elasticity of the squeezed part of the body. In such cases the reopening device will always be so slow that there is insufficient time for the door to change direction before an accident occurs.

Energy tolerance of less elastic parts of the body:

Ref. 1: Saha, S. and Hayes, W.C., "Tensile Impact Properties of Bone", Stanford University, 1973.

Ref. 2: James F. Parker, "Biastronautics Data Book", NTIS, September, 1972.

Ref. 3: E. Egkhor, H. Martinek and B. Wielke, "How to Increase the Stability of External Fixation Units. Mechanical Tests and Theoretical Studies", Arch. Orthop. Traumat. Surg., 1980.

According to the above research reports the strength of various bones varies quite considerably.

	Strength		Elastic modulus	
	$\frac{lbf/in^2}{10^3}$	(N/mm^2)	$\frac{lbf/in^2}{10^6}$	(N/mm^2)
Ref. 1	12...23	(84...157)	1.1...1.6	(7800...11000)
Ref. 2	14	(98)	3.3	(22600)
Ref. 3	10...15	(70...100)	0.6...1	(4000...7000)

On the basis of the values mentioned in these reports it is possible to calculate the tolerance of an arm or a finger to kinetic energy, which can vary from 0 to 10 J for old people and from 0.7 to 100 J for young adults.

According to some physicians the tolerance of the bones of children and old people is only 10% of the tolerance of the bones of a healthy adult. Moreover, the brittleness of the bones of old people considerably reduces their ability to withstand impact type energy. This claim has been supported by several liability cases in the United States concerning old people who have suffered fractured hips when hit by elevator doors. According to Ref. 2 above, the skull of an adult can resist a 45...100 J kinetic energy without breaking. Thus, a 4.5 J kinetic energy may be enough to crack the skull of a child. Severe tissue injuries will certainly occur before that.

From the references we could estimate the following average spring coefficients to be valid for bones of different individuals' fingers and forearms.

	bone surface	length
Finger	0.015...0.062 sq.in	1.2...2.4 in
Forearm	0.023...0.11 sq.in	8...14 in

	spring coefficient in lbf/in		
	Child	Grown up	Old
Finger	7400	37000	74000
Forearm	1700	11000	23000

The left side of diagram 1 shows the compression tolerance of bones with different spring coefficients (elasticity) when different kinetic energy pressure is exerted on them. If we assume, e.g., that the spring coefficient of a forearm is 5000 lbf/in and the door energy is 10 J, we get breaking compression of 0.19 in and (from the right side of the diagram) 950 lbf for corresponding compression force.

Pain threshold

With the assistance of some volunteers the KONE laboratories have measured the energy corresponding to the pain threshold of several parts of the body. The pain threshold was subjectively defined as "painful". The following kinetic energies and spring constants up to the pain threshold were achieved:

Finger	W = 0.2 J	(0.15 lbf·ft)	(400 N/1 mm)
Hand	W = 0.8 J	(0.59 lbf·ft)	(400 N/4 mm)
Head	W = 1 J	(0.72 lbf·ft)	(400 N/5 mm)
Arm	W = 3 J	(2.2 lbf·ft)	(300 N/12mm)

The reliability of the results suffers, however, from the fact that force was only slowly increased during the test. In reality, force will increase much faster, which may at least cause tissue or joint injuries. On the other hand, the elasticity will decrease rather abruptly while force is growing, which will increase the reaction force accordingly.

In the light of these values, one may question the correctness of the present limitations.

Different options for regulating speed

1 The fastest door operator fulfilling the requirements would probably be a linear unit using the highest possible values permitted by the door mechanism for acceleration and deceleration. Energy distribution of such a gear as a function of door movement is presented in diagram 2. It shows that a maximum kinetic energy of 7.4 lbf·ft (10 J) is exerted on all objects wider than 4 in which are squeezed between the doors.

2 Kinetic energy distribution of an operator based on generally used sinusoidal motion is shown in diagram 3. Kinetic energy may grow up to 13 lbf·ft (17 J) with the door still meeting the requirements of the safety code. It can be stated that the squeezed objects, if they are bigger than 8 in, are exposed to a kinetic energy greater than 7.4 lbf·ft (10 J) and energy directed to a hip could be near 11 lbf·ft (15 J).

If the above viewpoints concerning safety risks can be accepted, motion energy distribution of a safer door could look as shown in diagram 4.

The above cases have been calculated by using the following values:

	1	2	3
Max. speed (m/s)	0.53	0.42	0.42
Average (m/s)	0.39	0.39	0.27
Acceleration (m/s^2)	0.65	1.3	1.0
Retardation (m/s^2)	0.65	1.3	0.5
Closing time (s)	2.07	1.85	2.55

Capacity/Safety of elevator

A relatively good rule of thumb for the capacity calculations of elevators is: "Every lost second in the closing time of doors means a 5% lower lift capacity". Consequently, a safer elevator door would mean a 2.5% to 5% decrease in capacity.

Summary

The purpose of this analysis has been to raise questions about the correctness of the limit values for power-operated doors mentioned in the elevator safety codes. These questions are relevant today when most European countries are implementing new EN81 safety norms and the ISO TC178 working group has been urged to clarify the permissible values for international application.

My purpose has, of course, not been to give a scientifically accurate report to be taken as the basis for new permissible limit values.

This analysis should, however, show that any company that literally follows the permissible values of the elevator safety codes could find itself in deep trouble. Our responsibility is to encourage improvement of the wording and substance of the codes so that they can carry out their function of protecting elevator passengers.

COMPRESSION FORCE TO THE BONES BETWEEN LIFT DOOR PANELS

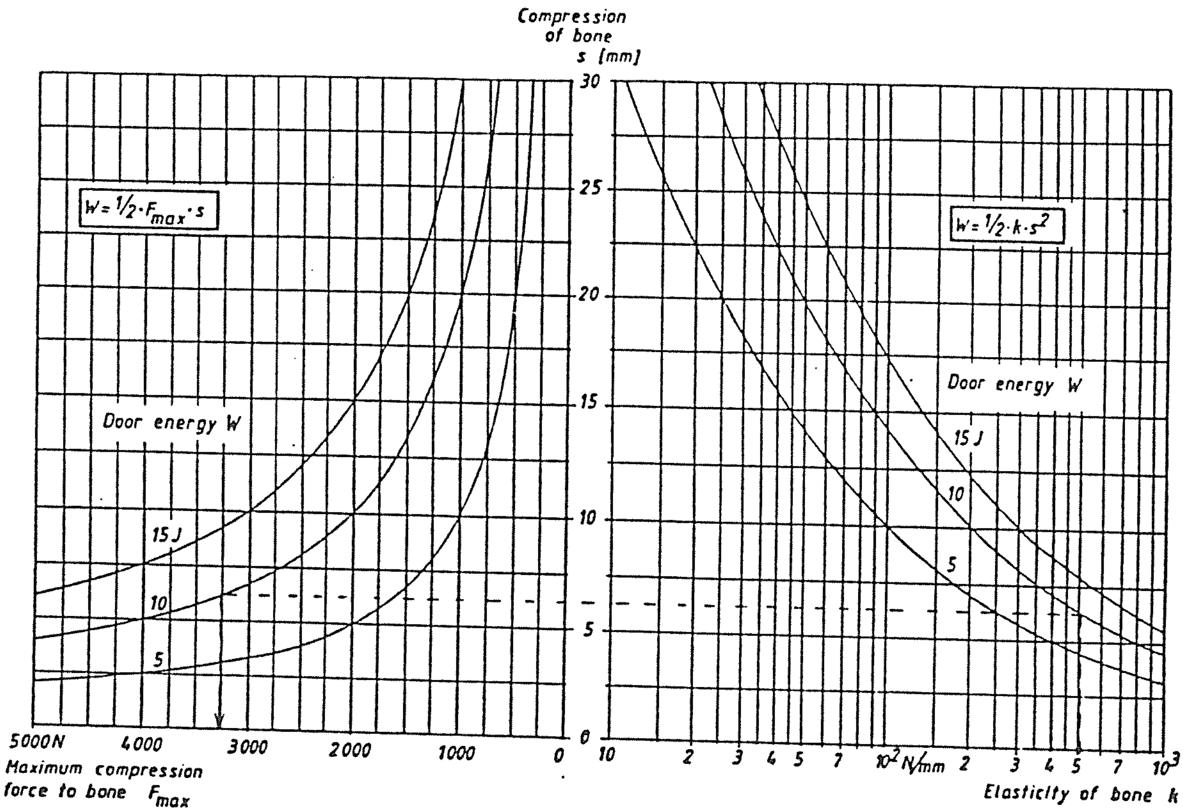
YOUNG´S MODULUS FOR BONES $E = 4000...22000\ N/mm^2$

CHILD GROWN-UP OLD

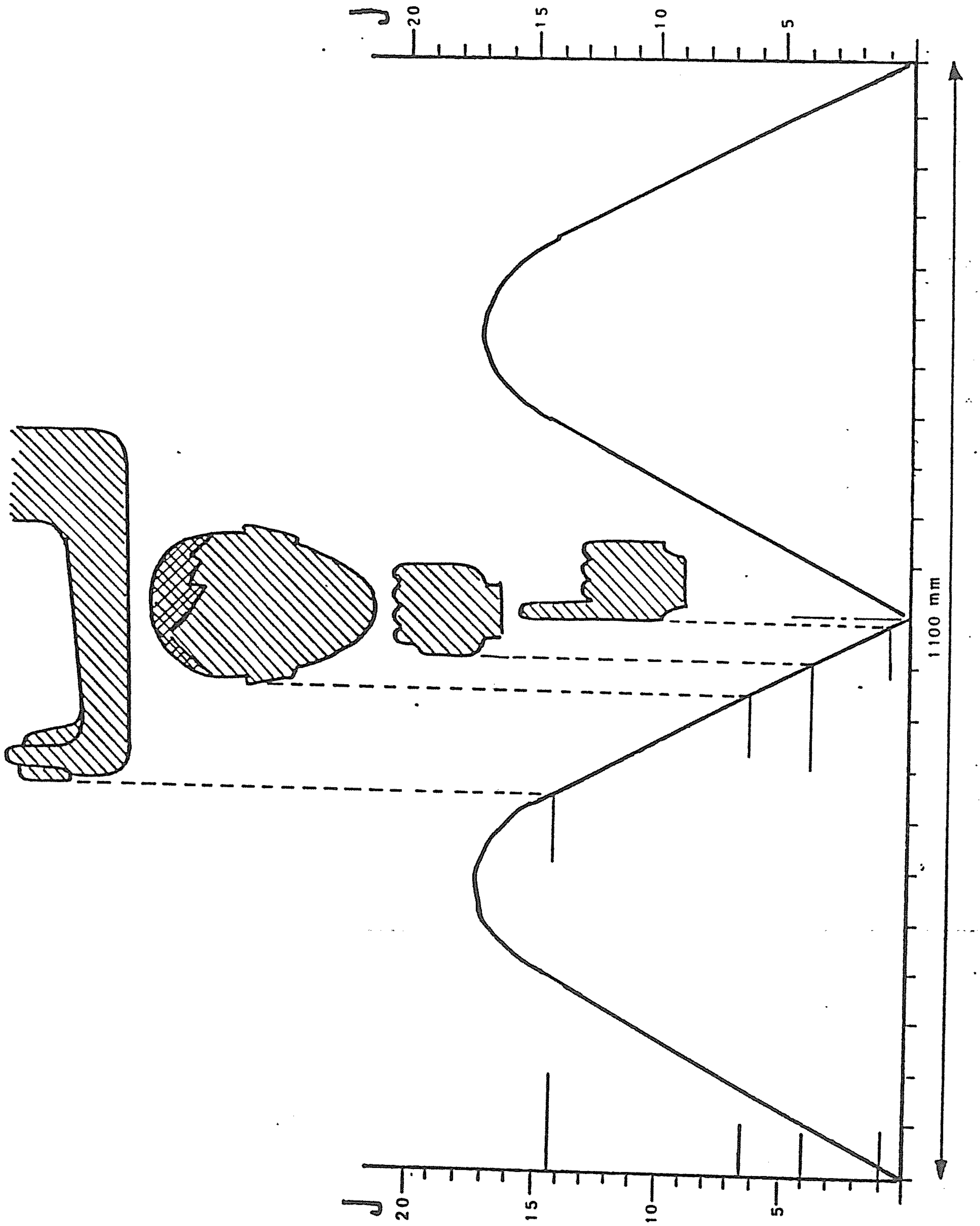
FINGER: $A = 10...40\ mm^2$, $L = 30...60\ mm$, $k = 1300...6500...13000\ N/mm$

FOREARM: $A = 15...70\ mm^2$, $L = 200...350\ mm$, $k = 300...2000...4000\ N/mm$

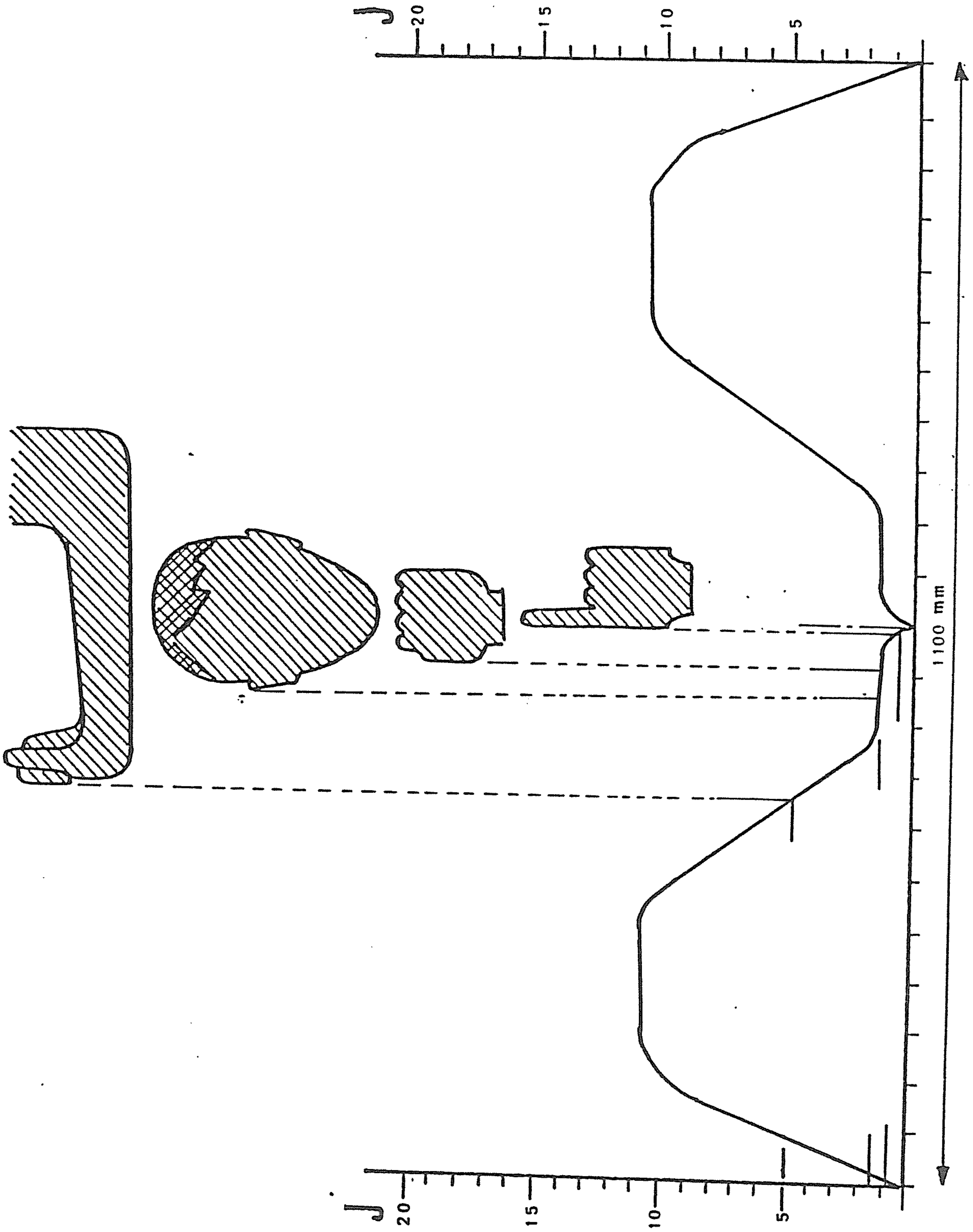
$$k \approx \frac{EA}{L}$$



E.G: ELASTICITY OF THE FOREARM = 500 N/MM. ENERGY OF THE DOOR PANELS = 10 J. COMPRESSION OF THE FOREARM = 7 MM.
MAXIMUM FORCE TO THE FOREARM = 3250 N.


J
20
15
10
5
1100 mm

<u>DIAGRAM N°4</u>



DOSSIER TECHNIQUE d'ASCENSEUR ELECTRIQUE

CONFORME A LA NBN E52-014 DE 79-06

DATE

MODALITÉS D'EMPLOI DU DOCUMENT

Ce document, préparé par le Comité Technique de la Section "Ascenseurs" de Fabrimétal, peut être employé tel quel dans tous les cas d'ascenseurs électriques à adhérence et où la compensation (éventuelle) va de cabine à contrepoids.

Il a été étudié pour que :

1) Les pages 1 et 2 avec l'annexe 1 constituent la "Demande d'autorisation préalable",

2) Les pages 1 à 5 avec les annexes 1, 1a, 1b et 2 constituent la partie "Documentation technique" du "Registre d'Ascenseur Electric

Pour être applicable dans tous les cas possibles, le formulaire devait prévoir diverses possibilités qui ne sont pas utiles dans les ascenseurs courants.

Certains constructeurs voudront sans doute préimprimer des formulaires simplifiés pour leurs ascenseurs normalisés ou programmer l'élaboration du Dossier Technique sur ordinateur.

Il importe cependant, que ceux qui n'utiliseront pas le formulaire d'origine, respectent certaines règles pour que tous ceux qui seront amenés à utiliser nos "Dossier Technique" en retirent tout le bénéfice que nous attendons.

1) La page de garde (page 1 de 5) doit, elle, toujours être utilisée. De toute façon, architectes, organismes de contrôles, ingénieurs conseils, entrepreneurs et clients s'habitueront à l'aspect de notre document.

2) Pour les autres pages, il est recommandé de conserver au minimum l'en-tête et le cadre et de disposer les informations aux mêmes places et avec les mêmes références que le formulaire d'origine pour que tous les intéressés sachent automatiquement où ils peuvent trouver telle ou telle information.
Le cas échéant, Fabrimétal pourra mettre à disposition des intéressés, des jeux de feuilles ne comportant que l'en-tête, le cadre et les notes.
Ces considérations s'appliquent aussi aux annexes à l'exception des annexes 1, 1a et 1b.

.=.=.=.=.=.=.=.=.=.=.=.=.=.=.=.=.=.=.

DOCUMENT PRÉPARÉ PAR LA SECTION ASCENSEURS DE FABRIMÉTAL 82-06

DOSSIER TECHNIQUE d'ASCENSEUR ELECTRIQUE

CONFORME A LA NBN E52-014 DE 79-06

DATE

REFERENCE DE L'ASCENSEUR :

INSTALLATEUR :

ACQUEREUR :

LIEU D'INSTALLATION :

ORGANISME RECEPTIONNAIRE :

RECEPTION accordée le .../.../... selon rapport référence ....
(ci-joint, copie du rapport)

MISE EN SERVICE LE :

PAR FIRME :

SIGNATURE :

NOTES : - LORS DE LA DEMANDE D'AUTORISATION PREALABLE, SEULES SONT A PRESENTER LES PAGES 1 et 2 AINSI QUE L'ANNEXE 1.

- LE DOCUMENT COMPLETE EST A METTRE A DISPOSITION DE L'ACQUEREUR 15 JOURS AU MOINS AVANT LA DATE FIXEE POUR LES EXAMENS & ESSAIS PREALABLES A LA MISE EN SERVICE.

DOCUMENT PRÉPARÉ PAR LA SECTION ASCENSEURS DE FABRIMÉTAL 82-06

REFERENCE DE L'ASCENSEUR :

1. GENERALITES

1.1. TYPE D'ASCENSEUR : PASSAGERS - MONTE-MALADES - DE CHARGE (accessible) (voir NOTE 1)

1.2. CHARGE NOMINALE : .... Q = ____________ Kg

1.3. NOMBRE DE PASSAGERS : .... = ____________

1.4. CATEGORIE D'USAGERS : ORDINAIRES - AUTORISES ET AVERTIS : *VOIR NOTE 1*

1.5. VITESSE NOMINALE : .... v = ____________ m/s

1.6. COURSE : .... = ____________ m

1.7. NOMBRE DE NIVEAUX DESSERVIS : .... = ____________

1.8. NOMBRE D'ACCES : .... = ____________

1.9. MASSE DE LA CABINE + ETRIER : .... P_1 = ____________ Kg : *VOIR NOTE 2*

1.10. MASSE DU PENDENTIF : .... P_2 = ____________ Kg

1.11. EQUILIBRAGE EN % : .... = ____________ %

1.12. MASSE DU CONTREPOIDS $(P_1 + \frac{P_2}{2} + Q \times \%)$: ctps = ____________ Kg

1.13. ACCES AU LOCAL DES MACHINES : *VOIR NOTE 3*

1.14. ACCES AU LOCAL DES POULIES : *VOIR NOTE 3*

1.15. CARACTERISTIQUES DU TABLEAU D'ARRIVEE AVEC PROTECTION SUR CIRCUIT FORCE MOTRICE :

____

1.16. CARACTERISTIQUES DU TABLEAU D'ARRIVEE AVEC PROTECTION SUR CIRCUIT ECLAIRAGE CABINE :

____

1.17. CARACTERISTIQUES DU TABLEAU D'ARRIVEE AVEC PROTECTION DU CIRCUIT D'ECLAIRAGE DE GAINE ET PRISE DE COURANT EN CUVETTE :

____

1.18. DISPOSITION DES POINTS LUMINEUX ET PRISES DE COURANT EN SALLE DES MACHINES ET DE POULIES LE CAS ECHEANT : VOIR *ANNEXE 1*

NOTE 1 : Biffer les mentions inutiles

NOTE 2 : Nous désignerons par :
P1 = la masse de la cabine avec étrier, portes, opérateur, etc ...
P2 = la masse du pendentif
P3 = la masse des câbles (ou chaînes) de compensation (éventuels)
P4 = la masse du compensateur (éventuel)
P5 = la masse des câbles (ou chaînes) de suspension
ctps = la masse du contrepoids

Le présent document est applicable aux ascenseurs avec contrepoids et où la compensation (éventuelle) va de cabine à contrepoids. Pour tout autre cas, un document similaire est à proposer par le constructeur

NOTE 3 : Voir plan d'installation ANNEXE 1

REFERENCE DE L'ASCENSEUR :

2. RENSEIGNEMENTS TECHNIQUES ET PLANS

2.1. PLAN D'INSTALLATION *(VOIR ANNEXE 1)*. PLAN - RÉF. : ____________

2.2. CARACTÉRISTIQUES TECHNIQUES :

2.2.1. SUSPENSION : CÂBLES - CHAÎNES : *VOIR NOTES 1 ET 2*

- RAPPORT DE SUSPENSION : .... 1/1 - 2/1 - 3/1 - 4/1 *VOIR NOTE 1*
- VITESSE DES CABLES (POULIE DE TRACTION) : . v_c = ____________ m/s
- TYPE DE CABLES : .... *VOIR ANNEXE 2*
- NOMBRE DE CABLES : .... n = ____________
- DIAMETRE DES CABLES : .... d = ____________ mm
- MASSE DES CABLES : .... P5 = ____________ kgs
- COEFFICIENT DE SECURITE : .... = ____________ *VOIR ANNEXE 2*

2.2.2 COMPENSATION : CÂBLES - CHAÎNES - SANS : *VOIR NOTE 1*

- NOMBRE DE CABLES-CHAINES : .... = ____________
- DIAMETRE DES CABLES : .... = ____________ mm
- MASSE DES CABLES-CHAINES : .... P3 = ____________ kgs
- MASSE DU TENDEUR (éventuel) : .... P4 = ____________ kgs
- DISPOSITIF ANTI-REBOND : .... OUI - NON

2.2.3. MACHINE DE TRACTION : TYPE TREUIL-GEARLESS : *VOIR NOTE 1*

- MODELE : ____________
- DIAMETRE DE LA POULIE : .... D = ____________ mm
- ANGLE D'ENROULEMENT : .... α = ____________ rad (______°)
- GORGES : SEMI-CIRCULAIRE - EN V ANGLE ... γ = ____________ rad (______°)
  SOUS-TAILLE : .... OUI - NON
  ANGLE ....β = ____________ rad (______°)
- ADHERENCE ET PRESSION SPECIFIQUES : CALCULS A *L'ANNEXE 3*

2.2.4. CÂBLE DU LIMITEUR DE VITESSE CABINE :

- TYPE ET COMPOSITION : ____________
- DIAMETRE : .... D = ____________ mm
- CHARGE DE RUPTURE MINIMALE : .... = ____________ kgs
- COEFFICIENT DE SECURITE : .... = ____________

2.2.5. LIMITEUR DE VITESSE CABINE : TYPE :

- DIAMETRE DE LA POULIE : .... = ____________ mm ≥30 x Ø câbles
- CERTIFICAT VOIR *ANNEXE 10*

NOTE 1 : Biffer les mentions inutiles

NOTE 2 : Ce formulaire est rédigé pour des ascenseurs à câbles. Dans le cas de chaînes, il faut donner des renseignements équivalents tels que : type, composition, pas, charge de rupture, nombre de dents des pignons

2.2.6. PARACHUTE DE CABINE : *VOIR NOTE 1*

- TYPE : INSTANTANNE - COIN - GALET
  INSTANTANNE AVEC EFFET AMORTI
  A PRISE AMORTIE
- CERTIFICAT : VOIR *ANNEXE 8*

2.2.7. PARACHUTE DE CONTREPOIDS : *VOIR NOTE 1*

- TYPE : INSTANTANNE - COIN - GALET
  INSTANTANNE AVEC EFFET AMORTI
  A PRISE AMORTIE
- CERTIFICAT : VOIR *ANNEXE 9*

2.2.8. COMMANDE DU PARACHUTE DE CONTREPOIDS : *VOIR NOTE 1*

- LIMITEUR DE VITESSE
- CABLE DE SECURITE
- RUPTURE DE SUSPENSION

2.2.8.1. CÂBLE :

- TYPE ET COMPOSITION : ____________________
- DIAMETRE : .... = __________ mm
- CHARGE DE RUPTURE MIN. : .. = __________ kgs
- COEFFICIENT DE SECURITE :.. = __________

2.2.8.2. LIMITEUR :

- TYPE : ____________________
- DIAMETRE DE LA POULIE : ... = __________ mm > 30 x ∅ câble
- CERTIFICAT VOIR *ANNEXE 11*

2.2.9.. GUIDES DE CABINE : ÉTIRÉS - USINÉS - RECTIFIÉS : *VOIR NOTE 1*

- DIMENSIONS : .... = ________ / ________ / ________
- PENDUS, POSES, FLOTTANTS
- VOIR CALCULS *ANNEXE 4*

2.2.10. GUIDES DE CONTREPOIDS : ÉTIRÉS - USINÉS - RECTIFIÉS : *VOIR NOTE 1*

- DIMENSIONS : .... = ________ / ________ / ________
- PENDUS, POSES, FLOTTANTS
- VOIR CALCULS *ANNEXE 5*

NOTE 1 : Biffer les mentions inutiles.

REFERENCE DE L'ASCENSEUR :

2.2.11. AMORTISSEURS DE CABINE : *VOIR NOTE 1*

- TYPE : A ACCUMULATION D'ENERGIE - A ACCUMULATION D'ENERGIE AVEC RETOUR AMORTI - A DISSIPATION D'ENERGIE
- COURSE : .... = ____________
- VOIR *ANNEXE 12* : - soit les caractéristiques du(des) amortisseurs à accumulation d'énergie avec courbes
  - soit certificat des amortisseurs à dissipation d'énergie

2.2.12. AMORTISSEURS DE CONTREPOIDS : *VOIR NOTE 1*

- TYPE : A ACCUMULATION D'ENERGIE - A ACCUMULATION D'ENERGIE AVEC RETOUR AMORTI - A DISSIPATION D'ENERGIE
- COURSE : .... = ____________
- VOIR *ANNEXE 13* : - soit les caractéristiques du(des) amortisseurs à accumulation d'énergie avec courbes
  - soit certificat des amortisseurs à dissipation d'énergie

3. SCHEMAS ELECTRIQUES

3.1. SCHÉMA DE PRINCIPE DES CIRCUITS DE PUISSANCE ET D'ÉCLAIRAGE.

VOIR ANNEXE 1A - SCHÉMA RÉF. : ____________

3.2. SCHÉMA DE PRINCIPE DES CIRCUITS DE SÉCURITÉ ET D'ALARME.

VOIR ANNEXE 1B - SCHÉMA RÉF. : ____________

4. CERTIFICATS OU NOTES DE CALCULS : *VOIR NOTES 1 ET 2*

	TYPE ET RÉFÉRENCE	ANNEXE
4.1. DISPOSITIFS DE VERROUILLAGE		*N° 6*
4.2. PORTES PALIÈRES		*N° 7*
4.3. PARACHUTE CABINE		*N° 8*
4.4. PARACHUTE CONTREPOIDS		*N° 9*
4.5. LIMITEUR DE VITESSE CABINE		*N° 10*
4.6. LIMITEUR DE VITESSE CONTREPOIDS		*N° 11*
4.7. AMORTISSEURS CÔTÉ CABINE		*N° 12*
4.8. AMORTISSEURS CÔTÉ CONTREPOIDS		*N° 13*

NOTE 1 : Biffer les mentions inutiles.

NOTE 2 : Voir l'Addendum à la NBN E52-014 pour les dispositions provisoires admises en Belgique en ce qui concerne les essais types.

REFERENCE DE L'ASCENSEUR :

L'annexe 1 se compose, en fait, de 3 documents:

1) Plan d'installation.

1a) Schema de principe des circuits de puissance et d'éclairage.

1b) Schema de principe des circuits de sécurité et d'alarme.

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ANNEXE 2

REFERENCE DE L'ASCENSEUR :

CONTROLE DES CABLES DE SUSPENSION.

CARACTÉRISTIQUES : VOIR NOTE 1

- TYPE : SEALE - AUTRE : ____________________
- CONSTRUCTION : 6 x 19 - 8 x 19
- RESISTANCE DES FILS : 1570 ou 1370/1770 ou 1770 N/mm2
- AME : TEXTILE - ACIER - AUTRE A PRECISER : ____________
- CHARGE DE RUPTURE MINIMALE : ____________ kN (ISO/DIS 4344)
- MASSE APPROXIMATIVE : ______________ kg/m

MASSE DES CÂBLES :

P5 = k/m x nombre x rapport mouflage x course
= ___ x ___ x ________ x ___
= ________ kg

COEFFICIENT DE SÉCURITÉ :

$$= \frac{\text{(charge de rupture minimale) x (nombre) x (rapport mouflage)}}{10(P1 + \frac{P4}{2} + P5 + Q)}$$

= ________ > 12 (ou 16 si 2 câbles et adhérence)

RAPPORT DES DIAMÈTRES :

$$\frac{\text{Ø plus petite poulie}}{\text{Ø câble}} = \_\_\_\_\_\_\_\_ > 40$$

NOTE 1 : Biffer les mentions inutiles

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REFERENCE DE L'ASCENSEUR :

CONTROLE DE L'ADHERENCE. (VOIR E52-014 CHAP. 9 NOTE 1)

$$\frac{T1}{T2} \times C_1 \times C2 \leq e^{f\alpha} = \_\_\_\_\_\_$$

Voir les abaques donnant $e^{f\alpha}$:

. en fonction de α et γ pour les gorges en V - VOIR ANNEXE 3A

. en fonction dd α et β pour les gorges rondes ou sous-taillées - VOIR ANNEXE 3B

- CALCUL AVEC CABINE SURCHARGEE EN BAS :

$$\frac{P1 + 1,25Q - 0,5P4 + P5}{ctps + 0,5P4 + P3} \times C1 \times C2 \leq e^{f\alpha}$$

$$\frac{\quad + \quad + \quad}{\quad + \quad} \times \quad \times \quad \leq \_\_\_\_\_\_$$

- CALCUL AVEC CABINE VIDE EN HAUT :

$$\frac{ctps + 0,5P4 + P5}{P1 + P2 + 0,5P4 + P3} \times C1 \times C2 \leq e^{f\alpha}$$

$$\frac{\quad + \quad}{\quad + \quad + \quad} \times \quad \times \quad \leq \_\_\_\_\_\_$$

CONTROLE DE LA PRESSION SPECIFIQUE. (VOIR E52-014 CHAP. 9 NOTE 2)

- LIMITE SUPERIEURE ADMISSIBLE :

$$p \max = \frac{12,5 + 4 v_c}{1 + v_c} = \frac{12.5 + 4}{1 +} = \_\_\_\_\_\_ \text{ N/mm2}$$

- CAS DES GORGES EN V :

$$p = \frac{T}{n\ d\ D} \times \frac{4,5}{\sin(\gamma/2)} = \_\_\_\_\_\_ \times \_\_\_\_\_\_ = \_\_\_\_\_\_ \text{ N/mm2}$$

- CAS DES GORGES RONDES (SOUS-TAILLEES OU NON) :

$$p = \frac{T}{n\ d\ D} \times \frac{8 \cos(\beta/2)}{\pi - \beta - \sin\beta} = \_\_\_\_\_\_ \times \_\_\_\_\_\_ = \_\_\_\_\_\_ \text{ N/mm2}$$

- AVEC $T = \dfrac{10 \cdot (P1 + Q + P5 + 0.5 \cdot P4)}{\text{rapport de suspension}}$

$$T = \frac{10 \cdot (\quad + \quad + \quad + \quad)}{\text{rapport de suspension}}$$

T =.........Newtons

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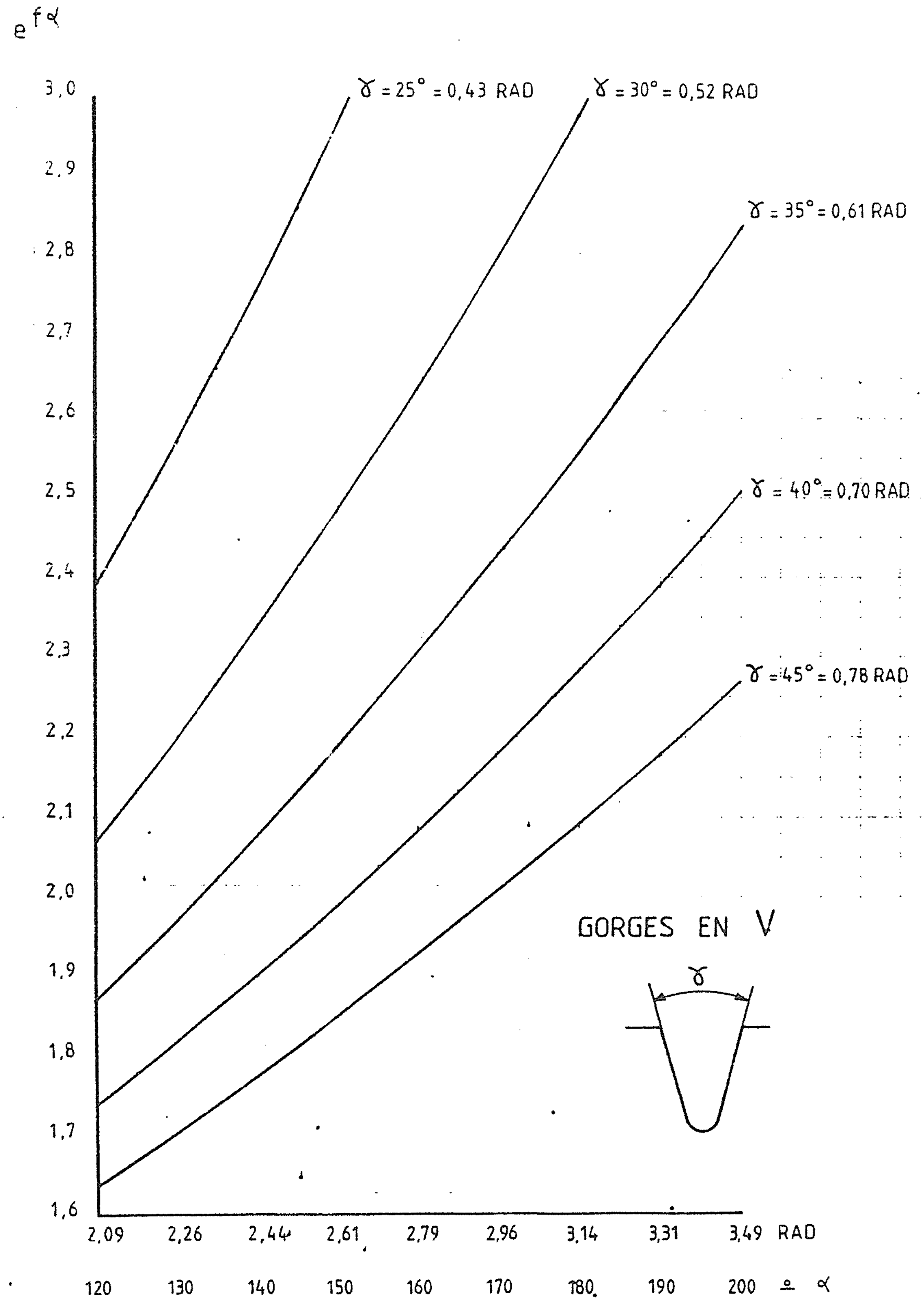
CONFORME A LA NBN E52-014 DE 79-06	DATE ANNEXE 3A
REFERENCE DE L'ASCENSEUR :	



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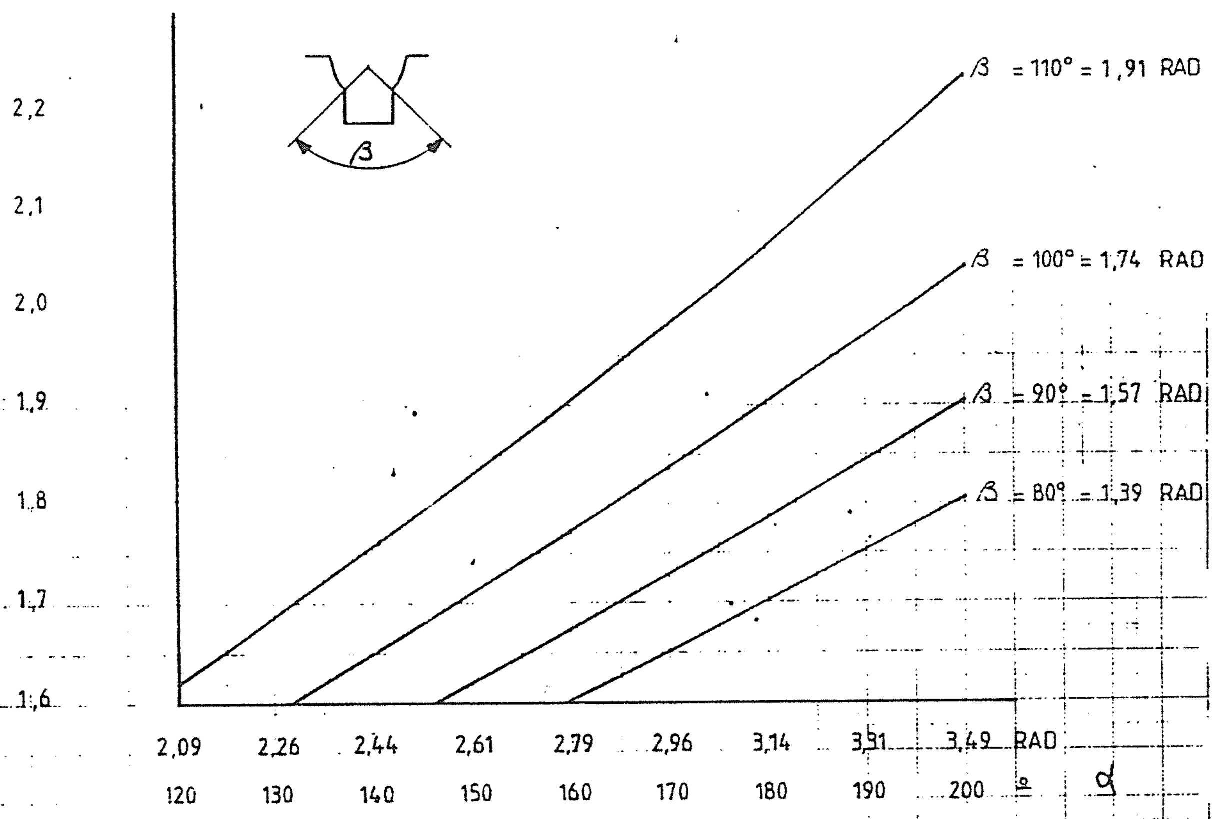
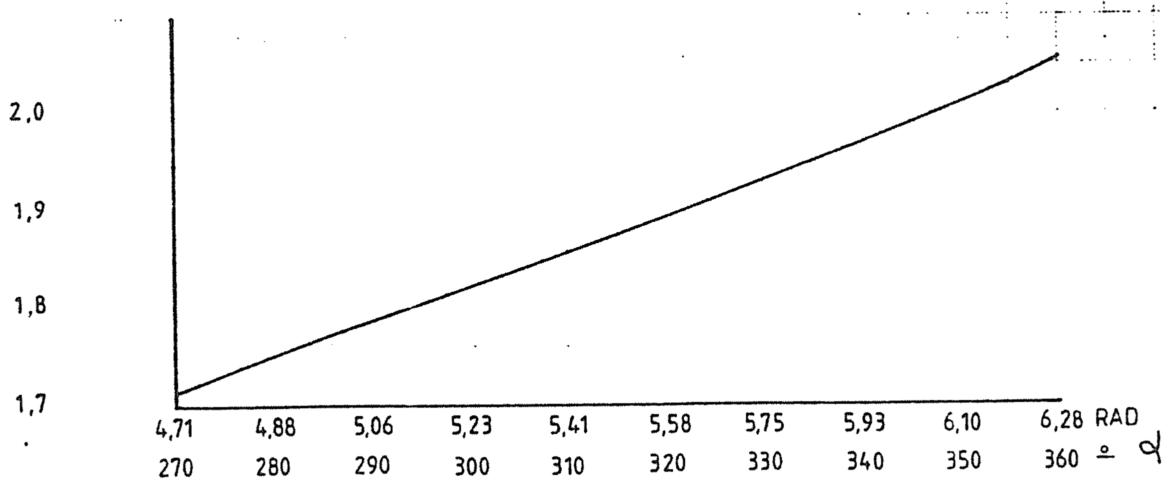
DOSSIER TECHNIQUE d'ASCENSEUR ELECTRIQUE	DATE
CONFORME A LA NBN E52-014 DE 79-06	ANNEXE 3B
REFERENCE DE L'ASCENSEUR :	

$e^{f\alpha}$ GORGES SOUS TAILLEES



$e^{f\alpha}$ GORGES RONDES



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ANNEXE 4

REFERENCE DE L'ASCENSEUR :

CALCUL DES GUIDES DE CABINE : *VOIR NOTE 1*

Les calculs se font en application de la note 1 en fin de chap. 10 de la NBN E52-014.

$$\sigma_k = \frac{(X)(P1 + P2 + P3 + Q) . \omega}{A} \leq \begin{array}{l} 140 \text{ N/mm2 (pour acier à 370 N/mm2)} \\ 210 \text{ N/mm2 (pour acier à 520 N/mm2)} \end{array}$$

avec (X) = 25 pour parachute instantanné à coin
15 " " " à galet
10 " " à prise amortie

Les caractéristiques des guides normalisés sont à prendre au document ISO/ 7465
VOIR ANNEXE 4A

- GUIDE CHOISI : ______________________
- SECTION . . . . A = ________ mm2
- RAYON DE GIRATION : ixx = ________ mm
  iyy = ________ mm
- DISTANCE MAX. ENTRE ATTACHES : L_k = ____________ mm
- COEFFICIENT D'ELANCEMENT : $\lambda = \frac{L_k}{\text{rayon de giration minimum}}$ = ____________
- COEFFICIENT DE MAJORATION DES CHARGES AU FLAMBAGE EN FONCTION DE λ : ω = ____________
  VOIR NBN E52-014 CHAPITRE 10

$$\sigma_k = \frac{(\quad) \times (\quad + \quad + \quad + \quad) \times \quad}{\quad} = \_\_\_\_\_\_\_\_ \leq \begin{array}{l} 140 \text{ N/mm2} \\ \text{ou } 210 \text{ N/mm2} \end{array}$$

NOTE 1 : L'annexe 4 permet la justification du choix des guides posés ou flottants disposés de façon classique par rapport à la cabine.
A fortiori, elle permet aussi la justification des guides pendus.
Si dans ce cas de guides pendus, l'installateur veut utiliser des guides de section plus faible que celle qui résulterait de ce calcul, il lui appartient de proposer le mode de calcul justifiant son choix.
De même, dans tous les cas où la disposition des guides est inhabituelle (ex. : cabine guidée par l'arrière), il appartient à l'installateur de proposer un mode de calcul approprié.

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ANNEXE 4A

REFERENCE DE L'ASCENSEUR :

EXTRAIT DE LA NORME ISO 7465

A = étiré
B = usiné

VOIR NOTE 1

DÉSIGNATION	DIMENSIONS			A cm2	I XX cm	I YY cm
T45A	45	45	5	4,25	1,38	0,95
T50A	50	50	5	4,75	1,54	1,05
T70-1A	70	65	9	9,51	2,09	1,40
T70-2A	70	70	8	10,52	2,12	1,48
T75-1A	75	55	9	7,98	1,76	1,40
T75-2A	75	62	9	10,12	1,92	1,61
T75-3A/B	75	62	10	10,99	1,92	1,55
T82-A/B	82,5	68,25	9	10,90	2,13	1,67
T89-A/B	89	62	16	15,70	1,95	1,83
T90-A/B	90	75	16	17,00	2,44	1,74
T125-A/B	125	82	16	22,9	2,57	2,52
T127-1B	127	89	16	22,50	2,86	2,65
T127-2A/B	127	89	16	28,9	2,63	2,85

SI D'AUTRES TYPES DE GUIDES SONT UTILISES, INDIQUER LEURS CARACTERISTIQUES.

VOIR NOTE 1

DÉSIGNATION	DIMENSIONS			A cm2	I XX cm	I YY cm

NOTE 1 : Coefficient de majoration des charges au flambage ω : voir NBN E52-014 chap. 10 - note 1 tableaux 2 et 3

REFERENCE DE L'ASCENSEUR :

CALCUL DES GUIDES DE CONTREPOIDS : [illegible] NOTES 1 ET 2

Les calculs se font en application de la note 1 en fin de chap. 10 de la NBN E52-014.

$$\sigma_k = \frac{(X)\ (ctps + P3)\ \omega}{A} \leqslant \begin{array}{l} 140\ N/mm2\ \text{(pour acier à 370 N/mm2)} \\ 210\ N/mm2\ \text{(pour acier à 520 N/mm2)} \end{array}$$

avec (X) = 25 pour parachute instantanné à coin
15 " " " à galet
10 " à prise amortie

Les caractéristiques des guides normalisés sont à prendre au document ISO 7465
VOIR ANNEXE [illegible].

- GUIDE CHOISI : ______________________
- SECTION : A = ________ mm2
- RAYON DE GIRATION : ixx = ________ mm
  iyy = ________ mm
- DISTANCE MAX. ENTRE ATTACHES : L_k = ____________ mm
- COEFFICIENT D'ELANCEMENT : $\lambda = \dfrac{L_k}{\text{rayon de giration minimum}}$ =
- COEFFICIENT DE MAJORATION DES CHARGES AU FLAMBAGE EN FONCTION DE λ : ω = ____________
  VOIR NBN E52-014 CHAPITRE 10

σ_k = (____) x (________ + ________) x ________ = ____________ < 140 N/mm2
ou 210 N/mm2

NOTE 1 : [illegible] parachute sur contrepoids.
NOTE 2 : [illegible] justification des guides [illegible].
[illegible], elle permet la justification des guides pendus.
[illegible] guides pendus, l'[illegible] peut utiliser des guides de section plus faible [illegible] résulterait de ce calcul, il lui appartient de proposer le mode de calcul justifiant son choix.

REFERENCE DE L'ASCENSEUR :

DISPOSITIFS DE VERROUILLAGE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

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ANNEXE 7

REFERENCE DE L'ASCENSEUR :

PORTES PALIERES.

☐ VOIR CI-JOINT CERTIFICAT D'ESSAI SELON NBN 713-020 .

☐ VOIR CI-JOINT CERTIFICAT D'ESSAI SELON E52-014.

☐ LE CERTIFICAT N'EST PAS REQUIS POUR L'IMMEUBLE CONSIDERE.

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ANNEXE 8

REFERENCE DE L'ASCENSEUR :

☐ PARACHUTE DE CABINE A PRISE INSTANTANNEE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

☐ PARACHUTE DE CABINE A PRISE AMORTIE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

NOUS CERTIFIONS QUE LE PARACHUTE A ETE REGLE POUR UNE CHARGE TOTALE DE : ____________ Kgs
(cas des parachutes réglables)

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ANNEXE 9

REFERENCE DE L'ASCENSEUR :

☐ PARACHUTE DE CONTREPOIDS A PRISE INSTANTANNEE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

☐ PARACHUTE DE CONTREPOIDS A PRISE AMORTIE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

NOUS CERTIFIONS QUE LE PARACHUTE A ETE REGLE POUR UNE CHARGE TOTALE DE : __________ Kgs
(cas des parachutes réglables)

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LIMITEUR DE VITESSE CABINE.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

REFERENCE DE L'ASCENSEUR :

LIMITEUR DE VITESSE CONTREPOIDS.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/1 (E52-014 en BELGIQUE).

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ANNEXE 12	

REFERENCE DE L'ASCENSEUR :

AMORTISSEURS DE CABINES A DISSIPATION D'ENERGIE : *VOIR NOTE 1*
A ACCUMULATION D'ENERGIE AVEC RETOUR AMORTI.

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/Partie 1 (E52-014 en Belgique).

NOTE 1 : Biffer les mentions inutiles.

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ANNEXE 12

REFERENCE DE L'ASCENSEUR :

CALCUL DES AMORTISSEURS CABINE A ACCUMULATION D'ENERGIE : *VOIR NOTE 1*

- HAUTEUR DE L'AMORTISSEUR : ___ ____________ mm
- FLECHE MAXIMALE : ______________ mm > ______________ mm

  v étant la vitesse nominale exprimée en m/sec, cette flèche maximale doit être supérieure ou égale à :

  $0,135\ v^2$ = ______________ mm et, en tout cas, égale ou supérieure à 65 mm

- Cette flèche maximale est atteinte sous une charge de : ______________ kg

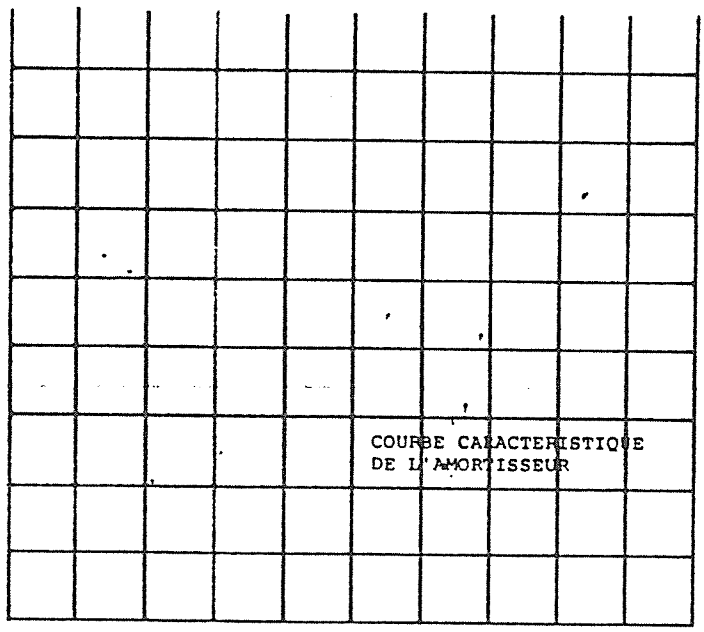
  qui doit être comprise entre $2,5\ \frac{(P1 + Q)}{n}$ soit : ______________ kg

  et

  $4\ \frac{(P1 + Q)}{n}$ soit : ______________ kg

  où n est le nombre d'amortisseurs sous la cabine.

CHARGE



FLECHE

NOTE 1 : Lorsque des amortisseurs à dissipation d'énergie sont employés, voir certificat ou Addendum à la NBN E52-014.

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ANNEXE 13

REFERENCE DE L'ASCENSEUR :

AMORTISSEURS DE CONTREPOIDS A DISSIPATION D'ENERGIE : *VOIR NOTE 1*
A ACCUMULATION D'ENERGIE AVEC RETOUR AMORTI

☐ VOIR CI-JOINT CERTIFICAT SELON EN81/Partie 1 (E52-014 en Belgique).

NOTE 1 : Biffer les mentions inutiles

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REFERENCE DE L'ASCENSEUR :

CALCUL DES AMORTISSEURS CONTREPOIDS A ACCUMULATION D'ENERGIE : *VOIR NOTE 1*

- HAUTEUR DE L'AMORTISSEUR : ________________ mm

- FLECHE MAXIMALE : ________________ mm > ________________ mm

  v étant la vitesse nominale exprimée en m/sec, cette flèche maximale doit être supérieure ou égale à :

  $0{,}135\ v^2$ = ________________ mm et, en tout cas, égale ou supérieure à 65 mm

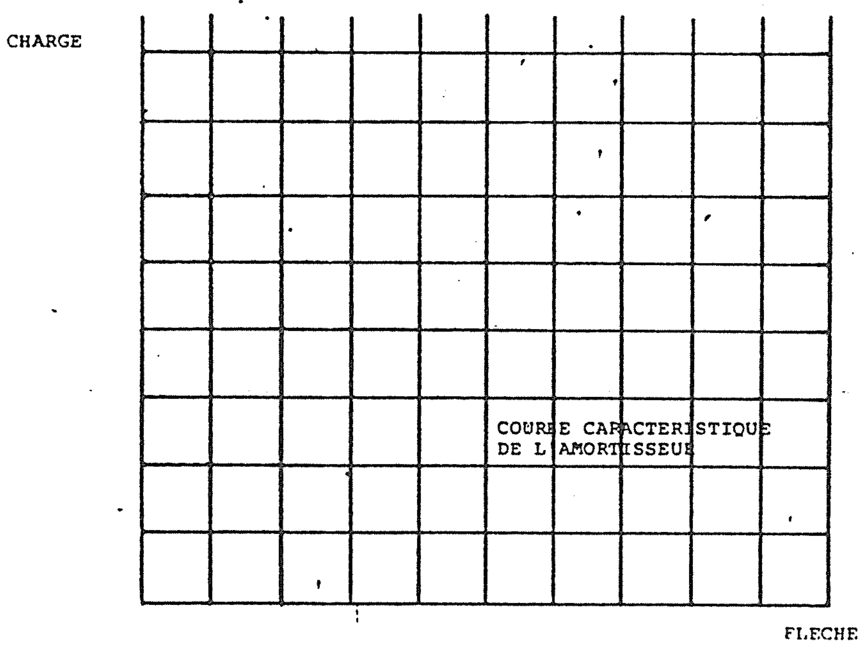
- Cette flèche maximale est atteinte sous une charge de : ________________ kg

  qui doit être comprise entre $2{,}5\ \dfrac{(\text{ctps})}{n}$ soit : ________________ kg

  et

  $4\ \dfrac{(\text{ctps})}{n}$ soit : ________________ kg

  où n est le nombre d'amortisseurs sous le contrepoids.



NOTE 1 : Lorsque des amortisseurs à dissipation d'énergie sont employés, voir certificat ou Addendum à la NBN E52-014.

Stopping the lift with a progressive safety gear

Behaviour in free fall and with counterweight attached

Case of the cast iron brake shoes applied with a constant force
Safety gear adjusted to meet the CEN requirements

APP:11/1
The problem is complex and making too many simplifications might lead to wrong conclusions. We will try to take all the factors into consideration but, nevertheless, a few things must be agreed upon before starting the calculations. Let us first define our goal.

In this APPENDIX, the goal is to find out how a test made with the counterweight attached, should be conducted to give a good enough idea of the behaviour of the safety gear when stopping the car with its rated load in free fall as in the type test.

APP:11/2
As mentioned in the headlines, we will consider only the case of cast iron brake shoes and steel guide rails.
We will also consider that the force applying the shoe remains constant throughout the time of operation of the safety gear. We will then ignore the braking during the building-up of the pressure at the beginning, when the wedge is driven into position.

Experience has shown that the friction factor, in this case, depends on 2 factors: the rubbing speed and the unit pressure in the contact area.

For relating the friction factor UR (standing for μ-rail) to the gliding speed and to the unit pressure, we will use the formula:

$$UR = \frac{Kp*0.4}{1 + 0.3*V}$$

The justification for this formula can be found in the APP:12/HB which follows the present one.

We will use Kp=1 because it has no impact on our conclusions in the present case. Indeed, if Kp had a different value, the force needed to apply the shoe on the rail would be different but the initial braking force would remain the same and its variation in relation with the gliding speed would be exactly the same.

APP:11/3
We will only consider the usual case of the cast iron (or steel) traction sheave and, since the ropes might be slipping, use for the friction factor US (standing for μ-sheave) the formula proposed in APP:05/4/d:

$$US = \frac{0.125}{1 + 0.125*Vg}$$

Where Vg is the gliding speed of the rope in the sheave.

The value US=0.125 when the gliding speed is zero is supported by the experiments of Dr-Ing Mulkow.

The other factors in the formula are based on some other tests.
This formula gives undoubtedly a fair idea of what happens when the ropes are slipping, an idea which is better than if the friction factor value were supposed to remain constant.

APP:11/4

We will assume that the deceleration of the traction sheave is independent of the forces developed by the safety gear and remain constant.

a) We will, at first, calculate, for the free-fall with rated load, the evolution of the acceleration, the speed and the distance during the safety gear operation.

b) Still with the rated load but with the counterweight attached, we will then calculate what would happen if the traction sheave were not pulling, either forwards or backwards.
This is a theoretical case, however it would nearly happen with a gearless machine if the mechanical brake were kept open, the power cut off and the electrical self-braking circuits neutralized.

The only traction needed would be for decelerating the rotating mass at the same rate as the car and counterweight.

Following strictly the wording of the ANSI code as regards the conditions for testing the safety gears, this is how the calculation should be made for evaluating the validity of the ANSI code requirements.

c) We will then make the calculations taking into account the traction in 4 different sets of conditions:

c1) rated load, sheave braking hard (deceleration = -20 m/s2).
This could be the case of the emergency braking of a gearless machine; emergency braking which is always associated with the triggering of the safety gear in normal operation.

To the contrary, it is impossible to have a hard braking with a geared machine having a heavy flywheel, because of the preponderance of the rotating masses. They keep the deceleration of the sheave nearly constant whatever the conditions of the stop.

c2) rated load, sheave kept rotating at constant speed (decel. zero)
This is how OTIS ~~in~~ tests its lifts in the USA.

c3) 25% overload, sheave rotating at constant speed (decel. zero).
Such a test, but at low speed, is required by EN 81/PartI at the acceptance of the lift (see EN/Appendix D2.(j).2).

National regulations may require to make the test at rated speed and in this Handbook, (HB/D2.(j).2), I have explained that such a test:

- is detrimental for the safety gear,
- is meaningless because no sliding distances have been specified.

I indicated however that such a test could be useful as orientation test for the manufacturers (before requiring a type test) providing limits were given for the sliding distances.

Indeed the calculations show that:
- with the traction sheave kept rotating at constant speed,
- with the safety gear adjusted to meet the CEN specifications,

the sliding distance is nearly the same as the one for the free fall test with rated load. This is especially so if a balancing of 40% is considered in the calculations. The distances are slightly reduced if the balancing is 50%.

c4) rated load, sheave braking soft (deceleration = -1m/s2)
This deceleration is more or less the one for normal operation.
This would practically be the case of the emergency braking of a geared machine with a heavy flywheel.

I made the calculations with these parameters to show that the sliding distances were very close to the ones in (c2).

The tabulation (Fig.3 of this APPENDIX) give the sliding distances which should be expected in the 6 cases: a, b, c1,c2,c3,c4.

Since having the traction sheave braking hard in parallel with the safety gear is either impossible (see c4) or questionable, the best thing to do is to keep the traction sheave rotating during the safety gear test.
In this way, the conditions of traction are clearly known, whatever the kind of machine or the kind of braking system.

APP:11/5

We assumed that the roping was 1/1 with double-wrap.
Other traction conditions would not give radically different results, because the value (C3*Alpha) may vary only between relatively narrow limits.

If the calculations were to be made for 2/1 roping, a factor 2 should be introduced into the program to recognize that the rope speed and the sheave speed are twice what they would be with 1/1 roping.

APP:11/6

We considered that there was always a locked-down compensator.
This locked-down compensator is required by the EN 81 code when the rated speed exceeds 3.5 m/s. In this Handbook (ref HB/9.6.1), I recommend using the device for any gearless lift.

For rated speeds up to 3.5 m/s, and if the safety gear is properly adjusted for the CEN specifications, the deceleration will never exceed 1*Gn when testing with the rated load in the car. Consequently, up to 3.5 m/s it does not make any difference in the calculations whether the compensator is locked-down or not, providing of course we take at least the rated load.

APP:11/7

For taking the traction into consideration, we will use the formula proposed in APP:05/HB, but without any safety factor since we reach the slipping limit:

$$\frac{T1}{T2} = e^{(C3 * US * Alpha)}$$

Let us call "Traction factor", TF, the value of $e^{(C3*US*Alpha)}$

Let us call R1 the value of

$$\frac{\text{Traction car side}}{\text{Traction ctw side}} = R1$$

We will then have, depending on the conditions:

R1 = TF   if the rope speed is higher than the sheave speed. (sheave trying to slow down the car)

R1 = 1/TF if the rope speed is lower than the sheave speed. (sheave helping the ctw up)

R1 = 1   if the traction does not interfere with the slow-down.

We will set R1=1 for the calculations in the case (b) of APP:11/4.
But we have also to introduce into the program the condition that R1=1 if the traction in the hoisting ropes becomes negative. This means that the tension is transferred to the compensating ropes. The tension in the hoisting ropes is zero and the traction sheave cannot interfere anymore. With a loaded car, this can happen only if the initial speed is very high; in this case the very low initial deceleration has to be compensated by decelerations exceeding 1*Gn when the speed is nearing zero.

APP:11/8

For writing the equations needed for solving this complex problem we need several new symbols, some of which we already mentioned above.

These symbols are explicited here below:

V = car speed at safety setting
W = tangential sheave speed
G = car acceleration (- if deceleration)
GS = constant acceleration of sheave (- if decel.)
QA = actual load in the car
Q = rated load
CT = mass of ctw
TF = traction factor
XA = factor multiplying US in the traction formula
TR = rope traction on car side
P1 = mass of empty car
P4 = mass of compensator
UR = friction factor on rail
US = friction factor on sheave
T = time
CS = calculating step
GN = 9.81
BP = force applying the brake shoe (as if only one)
NB: mass of ropes are neglected
FC = pull from compensator on car or ctw
D = travel of car
DM = max travel from initial point to buffer
R1 = ratio between rope pull on car side and pull on CTW side
OL = % of overload used for testing with ropes attached
ALPHA = wrapping angle in degres in the data (radians later)
C3 = wedging factor

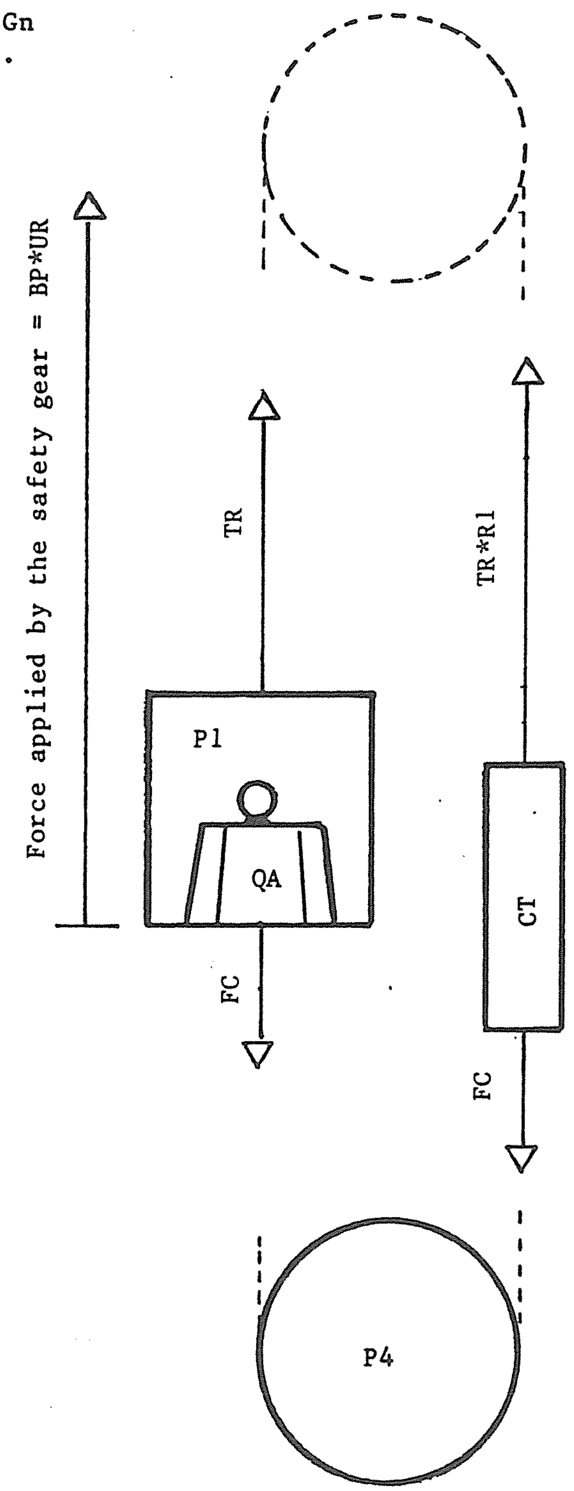
APP:11/9

The Fig 1 (next page) illustrates the elements which will be taken into consideration. Let us already remark that we will write the famous equation F=M*Gamma, separately for the car and for the counterweight.
Then we will introduce the condition that speed and/or acceleration are the same for both of them.

Fig 1 of APP:11/HB

Acceleration of Gravity Gn
Applied to P1, QA and CT.

FC = 0.5 * P4 * Gn



APP:11/10 Calculations sequence

We will review, step by step, the calculations to be made. The computer program was based on the same calculations but included, of course, additional instructions for guiding the operator, for printing and for calculating successively the various conditions listed in APP:11/4.

If not interested in the details of the calculations, you may jump to APP:11/11. You will find an example, a useful tabulation and conclusions.

STEP 1
Select the data on which to base the calculations:

- Initial speed	Vzero	m/s
- Initial acceleration in case of free fall (*) (NB: a negative value means a deceleration)	Gzero	m/s2
- Mass of empty car with accessories	P1	kg
- Mass of rated load	Q	kg
- Mass of counterweight	CT	kg
- Mass of compensator	P4	kg
- Overload to be considered for the case c(3)	OL	%
- Wrapping angle	ALpha	Deg
- Wedging factor	C3	-
- Time step for calculating	CS	sec
- Maximum travel before hitting the buffer	DM	m

(*) In the present case, the safety must be adjusted to meet the EN 81 requirements, i.e that the mean deceleration in free fall with the rated load must be around 6 m/s2.
To achieve this, the initial acceleration in the case of free fall with the rated load in the car may be calculated by the empirical formula:

$$\text{Gzero} = \frac{-\ \text{Gn} * 0.58}{1.4 + 0.2*\text{Vzero}}$$

STEP 2
Some precalculations must be made and some values fixed.

Gn = 9.81 (m/s2)
FC = 0.5 * P4 * Gn (Newtons)

The wrapping angle ALpha must be expressed in Radians for entering in the traction formula:

ALpha = ALpha * 0.0174533 (Radians)

To accelerate the recurrent calculations, let us calculate the following constant value entering the exponant of the traction formula:

XA = C3 * ALpha

Let us now calculate the value of the friction factor at the initial rubbing speed Vzero (assuming Kp=1 as explained in APP:11/2):

$$\text{UR} = \frac{0.4}{1 + 0.3 * \text{Vzero}}$$

Knowing UR and the desired initial deceleration Gzero, we can calculate the force needed for applying the brake shoe (NB: for simplicity, we supposed there is only one shoe, but in reality, this force is to be divided between all the existing shoes):

$$\text{BP} = \frac{(\text{P1}+\text{Q}) * (\text{Gn}-\text{Gzero})}{\text{UR}}$$

This value of BP is the one needed to meet the requirements of the type test. We must keep it throughout all the calculations.

STEP 3: Case (a): Calculations for the free fall conditions

STEP 3/a
To reflect the type test conditions we must set:

QA = Q

At the time zero, we have of course:

V = Vzero
G = Gzero

STEP 3/b
Let us suppose that the acceleration remains constant during the selected time step CS. At the end of the time step CS, we will have:

T = T + CS
D = D + V*CS
V = V + G*CS

STEP 3/c
With the new speed, we can calculate a new value for UR:

$$UR = \frac{0.4}{(1 + 0.3*V)} \qquad \text{(friction factor on rail)}$$

STEP 3/d
BP being known, we can with this UR calculate the new acceleration:

$$G = \frac{(P1+QA)*GN - BP*UR}{(P1+QA)}$$

which is the application of the good old formula: F=M*Gamma

STEP 3/e
With the new values for T, V, D and G, we go back to STEP 3/b and we do the cycle over and again until the speed reaches zero, then we go to STEP 3/f.

Also the distance should not exceed the selected maximum (DM).
This could happen if the initial deceleration were selected too low.
Let us be sure to select a DM high enough for the type of investigation we want to make; reaching this limit would mean that something is wrong.

We should of course make periodically a note of the various values in order to draw curves or to analyse the variations in the course of time.

STEP 3/f
We calculate the mean deceleration based on time

$$Gmean = \frac{Vzero}{Time}$$

It should be in the order of -6 m/s2 (at least with the CEN adjustment).

STEP 3/g
Let us make a note of the distance needed to stop the car in free fall.

STEP 4: Case (b) rated load - ctw attached - no interference from traction.

STEP 4/a
We shall, for this case, reset

QA	= Q	T	= zero
D	= zero	V	= Vzero
W	= V	R1	= 1

As explained in APP:11/7, the value of R1 must be fixed to 1 in this case.

STEP 4/b
Let us calculate the friction factor on the guide rail:

$$UR = \frac{0.4}{1 + 0.3*V}$$

STEP 4/c
Let us calculate the traction in the ropes on the car side.
Since the force applying the shoe (BP) has been calculated in STEP 2, we know all the forces applied to the car with the exception of the tension in the hoisting ropes. We can write the equation:

$$G = \frac{(P1+QA)*GN + FC - BP*UR - TR}{(P1+QA)}$$

For the counterweight, we can write the equation:

$$G = \frac{TR*R1 - FC - CT*GN}{CT}$$

G being of course the same for the car and for the counterweight, we can combine the 2 equation to find TR:

$$TR = \frac{2*CT*(P1+QA)*GN + FC*(P1+QA+CT) - BP*UR*CT}{CT + R1*(P1+QA)}$$

STEP 4/d
Now that we have TR, we can go back to the formula

$$G = \frac{TR*R1 - FC - CT*GN}{CT}$$

The acceleration G applies to the counterweight and to the car as well.

STEP 4/e
Supposing, as in STEP 3/b, that this acceleration remains constant during the selected time step CS, we will have at the end of CS sec:

T = T + CS
D = D + V*CS
V = V + G*CS
W = V

STEP 4/f
With the new values for T, D, V and W, we go back to STEP 4/b and do the cycle over and again until the speed reaches zero.
Then we go to STEP 4/g.

We will also check, at each cycle, whether the distance does not exceed the selected maximum (DM).

We should of course make periodically a note of the various values in order to draw curves or to analyse the variations in the course of time.

STEP 4/g
In this case, we will just make a note of the distance needed for stopping in the selected conditions.

STEP 5: Case (c1) rated load - ctw attached - sheave braking hard.

STEP 5/a:
We shall, for this case, reset:

QA = Q	T = zero
D = zero	V = Vzero
W = Vzero	GS = -20 m/s2

For the cases (c1), (c2), (c3) and (c4), the value of R1 will depend on:
- the traction in the hoisting ropes (positive or negative),
- the differential speed between ropes and traction sheave.

STEP 5/b:
Let us first calculate the friction factor on the guide rail:

$$UR = \frac{0.4}{1 + 0.3*V}$$

STEP 5/c
Let us calculate the friction factor between ropes and traction sheave:

$$US = \frac{0.125}{1 + 0.125*Vslipping}$$

Where: Vslipping = ABSolute value of (V -W)

STEP 5/d
Let us calculate the Traction factor:

$$TF = e^{(XA*US)}$$ (for XA see STEP 2)

STEP 5/e
Now, we have to calculate R1.

If TR = or < Zero then R1 = 1

If TR > Zero with V < W then R1 = TF
with V > W then R1 = 1/TF
with V = W then R1 = 1

For the first cycle of calculation, R1=1 because we still have V=W and the car is still hanging on the ropes.

STEP 5/f
Let us calculate the traction in the ropes on the car side.

We can use the formula developed in STEP 4/c

$$TR = \frac{2*CT*(P1+QA)*GN + FC*(P1+QA+CT) - BP*UR*CT}{CT + R1*(P1+QA)}$$

STEP 5/g
Let us calculate G:

$$G = \frac{TR*R1 - FC - CT*GN}{CT}$$

STEP 5/h
Using the same reasoning as in the previous STEPs, we will have at the end of the end of CS sec:

T = T + CS
D = D + V*CS
V = V + G*CS
W = W + GS*CS

STEP 5/i
With the new values for T, D, V and W, we go back to STEP 5/b and do the cycle over and again until the speed reaches zero.

We will also check, at each cycle, if the distance does not exceed the selected maximum (DM).

We should of course make periodically a note of the various values in order to draw curves or to analyse the variations in the course of time.

STEP 5/j
We will make a note of the distance needed for stopping in the conditions listed in the beginning of the present STEP.

STEP 6: Case (c2) rated load - ctw attached - sheave kept rotating

STEP 6/a
We shall, for this case, reset:

QA = Q	T = zero
D = zero	V = Vzero
W = Vzero	GS = zero

STEP 6/b to STEP 6/j
These STEPs are identical to STEP 5/b to STEP 5/j.

STEP 7: Case (c3) overload - ctw attached - sheave kept rotating

STEP 7/a
We shall, for this case, reset:

QA = Q*(1+ OL/100) (*)	T = zero
D = zero	V = Vzero
W = Vzero	GS = zero

(*) Let us remember that, for the example and for the tabulations developed later we took OL = 25% to be in line with CEN.

STEP 7/b to STEP 7/j
These STEPs are identical to STEP 5/b to STEP 5/j

STEP 8: Case (c4) rated load - ctw attached - sheave braking soft

STEP 8/a

We shall, for this case, reset:

QA = Q	T = zero
D = zero	V = Vzero
W = Vzero	GS = -1 m/s2

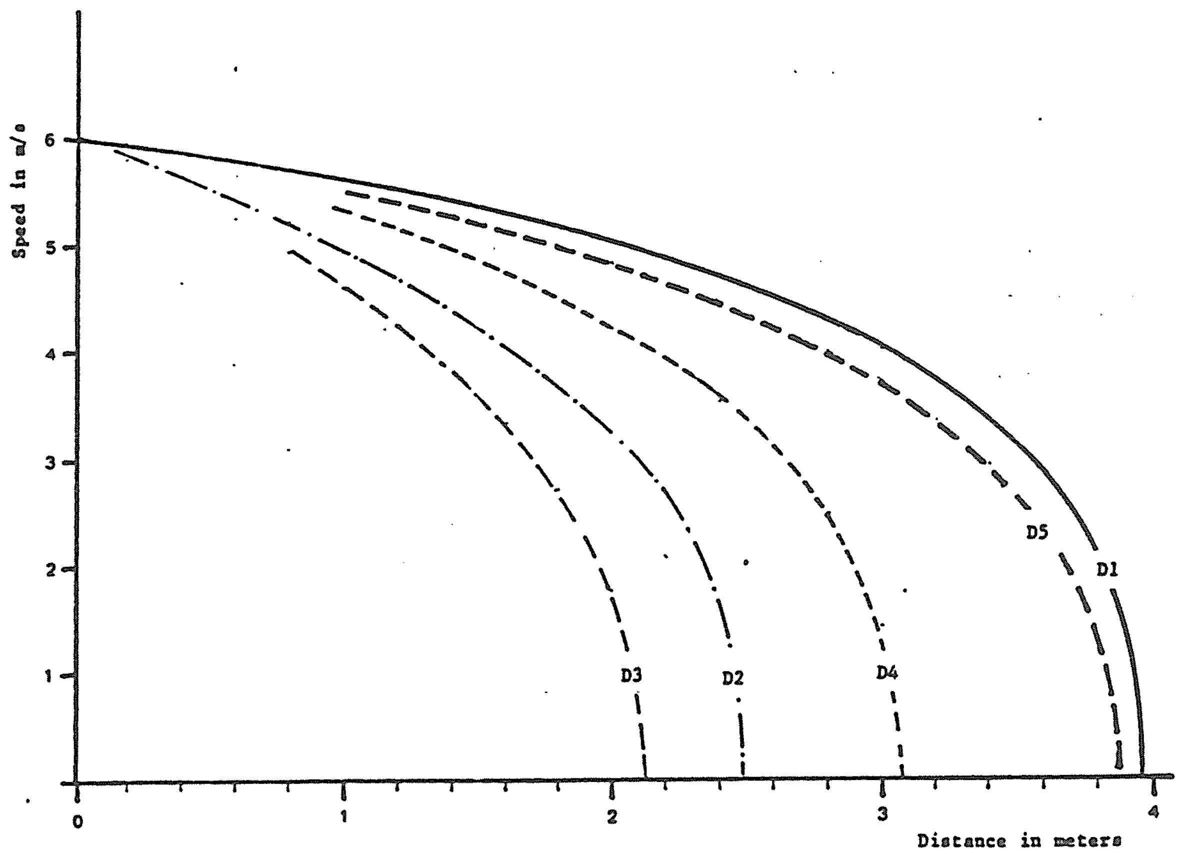
STEP 8/b to STEP 8/j
These STEPs are identical to STEP 5/b to STEP 5/j

APP:11/11: Example with the safety gear setting at 6 m/s

Fig 2 of APP:11/HB

VARIATION OF THE SPEED in relation with the DISTANCE TRAVELLED

D1 : rated load - free fall
D2 : rated load - ctw attached - no interference from traction
D3 : rated load - ctw attached - sheave braking hard
D4 : rated load - ctw attached - sheave kept rotating
D5 : 25% overload - ctw attached - sheave kept rotating



For the example we took of course the typical lift defined in HB/FWD/12 except that the balancing is 40% instead of 50%. We selected 40% because it is quite common for gearless lifts.

The following additional data was selected:

- speed at safety setting = 6 m/s
- wrapping angle Alpha = 315°
- wedging factor C3 = 1.03 (round groves)
- locked-down compensator = 720 kg

The mass of ropes has been neglected.

The Fig 2 illustrates the necessity of knowing if the traction sheave is braking or helping when making a test with the counterweight attached.

It shows also that the test made with 125% of the rated load with the sheave kept rotating at the rated speed gives results comparable to the free fall test. This is true for gears adjusted to meet the CEN requirements, however it is not true anymore for much softer adjustments.

APP:11/12: Sliding distances related to initial speeds

With the same data as was selected for APP:11/11 above but for speeds ranging from 1 m/s to 10 m/s, we calculated the sliding distances under the same conditions as above. In addition we calculated the sliding distance if the machine slows down as in normal operation during the safety operation.

Fig 3 of APP:11/HB

SLIDING DISTANCES for initial speeds ranging from 1 m/s to 10 m/s

D1 : rated load - free fall
D2 : rated load - ctw attached - no sheave interference
D3 : rated load - ctw attached - sheave braking hard
D4 : rated load - ctw attached - sheave kept rotating
D5 : 25% overload - ctw attached - sheave rotating
D6 : rated load - ctw attached - sheave braking soft

Gzero (Go): value selected for the initial deceleration
Gmean : mean deceleration based on time

Data taken into consideration for the calculations

Vinitial =	from 1 to 10	OL =	25	C3 =	1.03
ALPHA =	5.49779 Rad/Deg	P1 =	1300	Q =	1000
QA = 1000	CT = 1700	P4 =	720	CS =	.02

V	Go	Gmean	D1	D2	D3	D4	D5	D6
1.0	-3.6	5.0	0.10	0.06	0.06	0.08	0.10	0.08
2.0	-3.2	5.6	0.41	0.27	0.23	0.33	0.41	0.33
3.0	-2.8	6.0	0.94	0.61	0.52	0.74	0.93	0.74
4.0	-2.6	6.1	1.69	1.09	0.93	1.32	1.66	1.32
5.0	-2.4	6.1	2.69	1.70	1.45	2.07	2.61	2.08
6.0	-2.2	6.1	3.95	2.46	2.10	3.00	3.78	3.01
7.0	-2.0	6.1	5.50	3.35	2.87	4.10	5.18	4.12
8.0	-1.9	6.1	7.34	4.38	3.75	5.38	6.81	5.41
9.0	-1.8	6.0	9.49	5.55	4.76	6.84	8.68	6.89
10.0	-1.7	6.0	11.97	6.87	5.90	8.48	10.77	8.54

The initial deceleration was selected by using the empirical formula mentioned in APP:11/10-STEP 1.

The mean deceleration was calculated to verify that the initial deceleration had been correctly selected. This mean deceleration should ideally be 0.6*Gn.

We assumed that the same gearless arrangment was used for speeds as low as 1 m/s. With other machine arrangments, the results would be slightly different but, considering that:

- the margin of variation for the friction factors,
- the value of XA=ALpha*C3 can vary only within limits,

the need for additional tabulations was not evident.

However, if the best possible approximation is needed, the calculations should be made for each specific set of conditions.

APP:11/13: CONCLUSIONS

APP:11/13/a

The fact that the traction sheave can be either helping the safety gear to slow down the car, or, to the contrary, trying to lift the counterweight during the operation of the safety gear, has a marked influence on the sliding distances.

In some cases, such as a geared machine with a heavy flywheel, the braking of the sheave will be soft even if an emergency braking takes place at the same time as the safety setting. It means that it will try to lift the counterweight during the safety gear operation.

In other cases, it will be difficult to know whether the sheave is braking or driving. It might even do one and then the other.

Consequently, the best way of knowing exactly the conditions of the test is deliberately to keep the machine running at constant speed until after the complete stop of the car.

APP:11/13/b

In order to have an idea of the adequacy of the safety gear by making a test with the counterweight attached, you may:

- either take the rated load and compare the sliding distances with the values in column D4,
- or take 125% of the rated load and compare the sliding distances with the values in column D5.

The advantages of taking 125% of the rated load are that:

- the sliding distances will be very close to the sliding distances to be expected in free fall with the rated load (compare the values in D1 to the values in D5). Consequently the energy dissipated in the substitution test will be the same as the energy to be dissipated in the regular type test.
- the strength of the car structure will be adequately tested.

FRICTION between STEEL and CAST-IRON

Justification of the FORMULA giving the FRICTION FACTOR

APP:12/1

First of all, proper tribute must be paid to Dr Feyrer, now Professor at the University of Stuttgart. As early as 1967, he published in the N°12 of the "Technische Ueberwachung" an article in which he called attention to the variation of friction factor with gliding speed.

Fig 1 of APP:12/HB
Excerpt of Dr Feyrer publication.

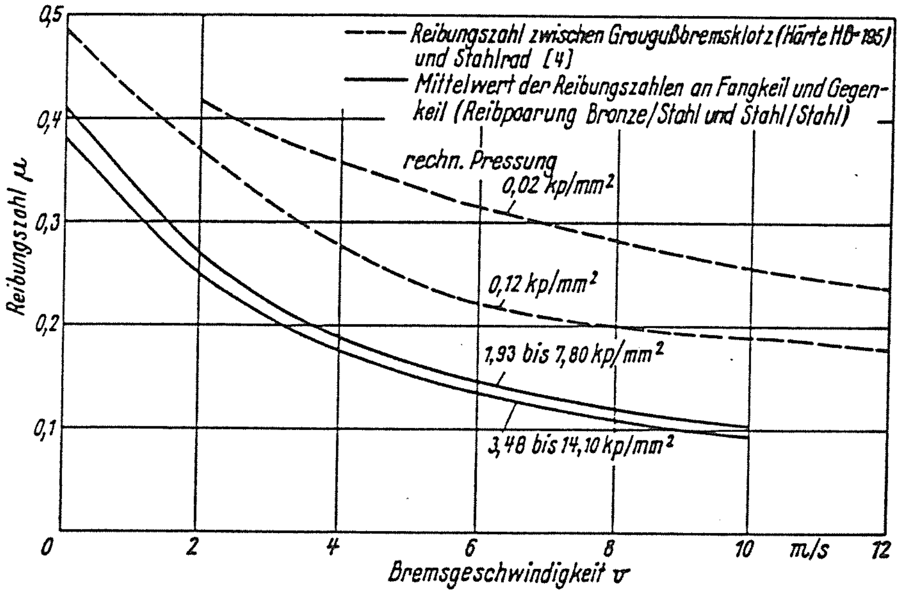


Bild 10. Vergleich der Reibungszahlen zwischen Rad und Bremsklotz aus [4] und Mittelwerten der Reibungszahlen zwischen ungeschmierter Schiene und Fangkeil sowie Gegenkeil

The solid lines represents the average friction factors proposed by Dr Feyrer after numerous tests made with STAHL safety gears.
Each line corresponds to a range of unit pressures.
The dotted lines come from the railroad industry (cast iron brake shoes on steel wheels).

APP:12/2

When the subject came up for discussion in the WG 1 of CEN/CT 10, the majority of the experts preferred to stick to the assumption of the ANSI Code, i.e. that the friction factor can be kept reasonably constant during the operation of the safety gear.

I have to admit that, at that time, I was convinced that the variations reported by Dr Feyrer were due to the specific design of the safety gear used for the experimentation (thin wedges and no prestress in the springs-see APP:14 for details).

However, from then on, I have collected informations on the subject.

APP:12/3

I collected reports on tests made in free fall with safety gears of what I call the "American" type of design, i.e. with sturdy wedges, pre-stressed springs, spring force reasonably centered on the shoe face.

A typical speed/time curve is illustrated in Fig 2. This indicates a strong increase of the braking force when the speed decreases. This can come only from the increase of the friction factor.

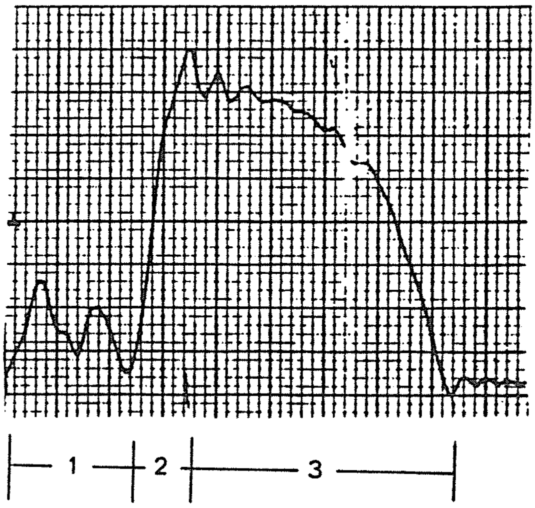


Fig 2 of APP:12/HB
Typical free fall test with an "American" type of safety gear.

1 release of the hook

2 free fall (1 Gn)

3 deceleration during safety gear operation

APP:12/4

I also noted that the TUV Laboratory (in Munich) did allow higher suspended masses when a given safety gear was applied at lower speeds.

Fig 3 of APP:12/HB
Excerpt from a TUV test certificate

2.1 Höchstzulässige Gesamtmasse [kg] in Abhängigkeit von der zulässigen Betriebsgeschwindigkeit und dem Oberflächenzustand der Führungsschienen:

Betriebs-geschwindigkeit	ungeschmierte Schienen (Rollenführungen)	geschmierte Schienen (Gleitführungen)
bis 0,85 m/s	12 000 kg	10 350 kg
über 0,85 bis 1,5 m/s	10 950 kg	10 350 kg
über 1,5 bis 2,5 m/s	8 810 kg	8 810 kg
über 2,5 bis 4,0 m/s	6 790 kg	-
über 4,0 bis 7,0 m/s	5 000 kg	-

There was only one possible justification for this increase of the load: the lower the initial speed, the higher the average friction factor.

Let us remember that the TUV Laboratory has a long experience in testing safety gears in free fall.

APP:12/5

I also collected data from:

- A publication by Professor Leloup from the University of Liège
- Publications from the railroad industry
- Schindler and OTIS test reports.

APP:12/6

From the analysis of all the data collected, it became evident that:

- a higher rubbing speed means a lower friction factor,
- the friction factor depends also on the unit pressure.

As regards the first conclusion, the data collected from tests made with lift safety gears was rather consistent and seemed to point to a larger variation than the experiments made in the railroad industry. Consequently, I decided to propose a formula based only on the data coming from tests made with lift safety gears

As regards the second conclusion, most sources indicate an increase of the friction factor when the unit pressure decreases, but the information is not very consistent. Fortunately, it is not very important for our purpose as I will explain later.

APP:12/7

After testing several formulae, I selected the following one:

$$\boxed{\text{Friction factor } \mu = \frac{Kp * 0.4}{1 + 0.3*V}}$$

where: Kp is a factor depending on the unit pressure
V is the figure (without dimension) expressing the rubbing speed when m/s are used for units

APP:12/8

Using Kp=1, I compared the curve built according this formula with:

- values taken from one of Dr Feyrer's curves,
- values derived from the TUV allowed loads,
- values from a Schindler test.

You can see, Fig 4, that the curve fits very well with the test results.

Then, using Kp=0.7, I compared the curve with values from tests made by OTIS with 2 different safety gears.

The Fig 5 shows that this curve also fits well with the tests results.
(NB:Fig 4 and Fig 5 can be found next page)

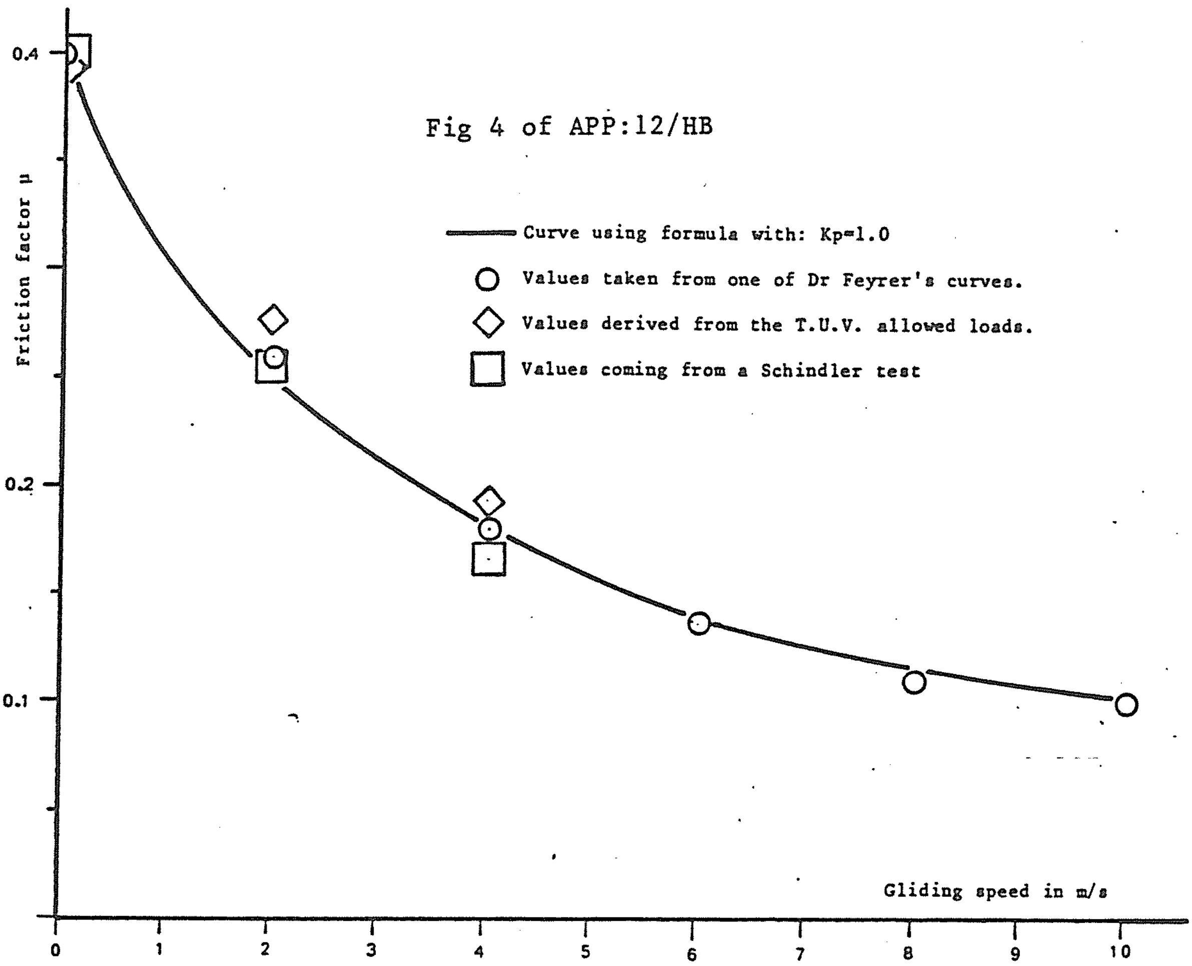


Fig 4 of APP:12/HB
Curve using formula with: Kp=1.0
Values taken from one of Dr Feyrer's curves.
Values derived from the T.U.V. allowed loads.
Values coming from a Schindler test
Friction factor μ
Gliding speed in m/s
0.4
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10

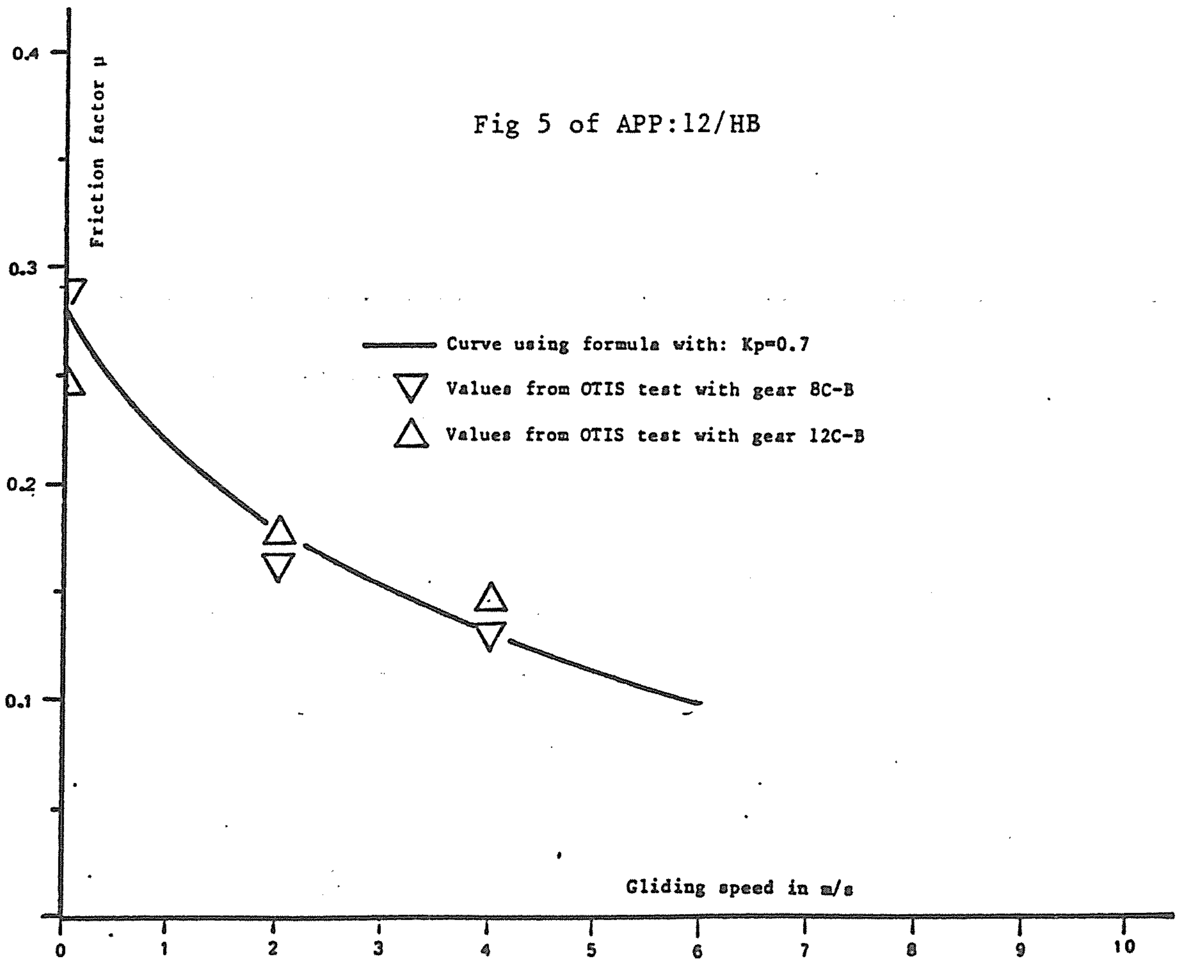


Fig 5 of APP:12/HB
Curve using formula with: Kp=0.7
Values from OTIS test with gear 8C-B
Values from OTIS test with gear 12C-B
Friction factor μ
Gliding speed in m/s
0.4
0.3
0.2
0.1
0
1
2
3
4
5
6
7
8
9
10

APP:12/9

So the formula can be safely introduced into computer's programs.
For most of the programs the factor Kp may be set to 1 because it governs only the value of the force applying the brake shoe and not the braking force itself (the one applied to the car).
The force applying the shoe is calculated by dividing the braking force by the friction factor.

I do not have enough data to indicate values for Kp. You will have to make some tests with your own materials and designs.
However, for a first approach in calculating the spring applying the shoe, I think you can simply use Kp=1 because most of the reports indicate friction factors'values close to 0.4 at zero speed.

The merit of the formula, for the purposes of this Handbook, is that it gives correctly the variation of the braking force as a function of the rubbing speed, once the force needed for applying the shoe (in order to have the necessary initial deceleration) has been calculated.

APP:12/10

Now one must realize that the curves obtained in actual tests are not as neat as the ones calculated with the formula.

The margin might be 10% to 15% but, to be on the safe side, you should reckon with 20% when designing the equipment.

APP:12/11

Let us remark here that the formula applicable to the friction factor of steel ropes on cast iron traction sheave is not the same.

It seems that the formula applicable in this case is:

$$\text{Friction factor for ropes} = \frac{0.125}{1 + 0.125 * V}$$

which can also be written:

$$\text{Friction factor for ropes} = \frac{1}{8 + V}$$

The difference can probably be explained by the creep of the ropes on the traction sheave and by the lateral displacement of the strands when the ropes gets wedged in the groves.

Stopping the lift with a progressive safety gear

Behaviour in free fall and with counterweight attached

Case of the cast iron brake shoes applied with a constant force
Analysis of the ANSI requirements and comparison with CEN

APP:13/1
We shall use the same calculation procedure as in Appendix APP:11/HB. But we will first set the initial deceleration during the free fall test at:

Gzero = zero

This means that the adjustment of the safety gear is so weak that the car tested in free fall keeps gliding at constant speed.
If the adjustment were still weaker, the car would keep accelerating down.

However, we will find that, when tested with the counterweight attached, the car will stop with the same adjustment. The calculation program will give us the sliding distances corresponding to each testing condition.

If, during the test with counterweight attached, these sliding distances were exceeded, it would mean that, if tested in free fall, the car would never stop. To the contrary, it would accelerate faster and faster.

Fig 1 of APP:13/HB

V = triggering speed
Go = initial deceleration at type test (free fall, rated load)
Gmean = mean decelaration based on time (here it is zero)
D1 = stopping distance at the free fall test (here it does not stop)
D2 = stopping distance ctw attached, no interference from sheave
D4 = stopping distance ctw attached, sheave kept rotating
D5 = stopping distance , same as D4 but with 25% overload

Data taken into consideration for the calculations

Vinitial = from 1 to 10 Gzero = 0
C3 = 1.03 ALPHA = 5.50 Rad
P1 = 1300 Q = 1000 OL = 25
CT = 1700 P4 = 720 CS = .02

V	Go	Gmean	D1	D2	D4	D5
1.0	0.0		buffer	0.10	0.16	0.25
2.0	0.0		buffer	0.39	0.61	0.91
3.0	0.0		buffer	0.85	1.29	1.89
4.0	0.0		buffer	1.46	2.19	3.16
5.0	0.0		buffer	2.24	3.30	4.71
6.0	0.0		buffer	3.16	4.60	6.54
7.0	0.0		buffer	4.24	6.11	8.63
8.0	0.0		buffer	5.47	7.81	10.99
9.0	0.0		buffer	6.84	9.71	13.61
10.0	0.0		buffer	8.37	11.79	16.47

It is interesting to note that, under these conditions of deceleration, the test with 25% overload does not give a sliding distance comparable to the free fall test (distance infinite because the deceleration remains zero).

Indeed, it is easy to understand that, to still have the deceleration equal to zero with counterweight attached, the overload in the car should be equal to the mass of the counterweight.

<u>APP:13/2</u>
However, adjusting the safety gear to the limit selected as the base for our calculations in APP:13/1, would be dangerous. We must reckon with a possible margin of error of 10% to 20% on the value of the friction factor as given by the formula justified in APP:12/HB.

To be on the safe side, the ajustment should be such that the initial deceleration during the free fall test with rated load be:

Gzero = -2 m/s2

(it means a margin of 20% on the retarding force exerted by the gear).

Fig 2 of APP:13/HB

The heading of the columns is the same as in Fig 1.

<u>Data taken into consideration for the calculations</u>

Vinitial = from 1 to 10 Gzero = -2
C3 = 1.03 ALPHA = 5.50 Rad
P1 = 1300 Q = 1000 OL = 25
CT = 1700 P4 = 720 CS = .02

V	Go	Gmean	D1	D2	D4	D5
1.0	-2.0	3.3	0.17	0.08	0.10	0.14
2.0	-2.0	4.2	0.59	0.30	0.39	0.51
3.0	-2.0	4.7	1.22	0.67	0.85	1.09
4.0	-2.0	5.1	2.03	1.16	1.45	1.86
5.0	-2.0	5.6	3.02	1.77	2.20	2.80
6.0	-2.0	5.8	4.20	2.50	3.09	3.92
7.0	-2.0	6.0	5.56	3.36	4.12	5.22
8.0	-2.0	6.3	7.09	4.33	5.29	6.68
9.0	-2.0	6.4	8.81	5.43	6.60	8.31
10.0	-2.0	6.6	10.70	6.64	8.05	10.11

<u>APP:13/3</u>
The gliding distances allowed by the ANSI code can be calculated by the following formulae taken from the Section 1306 of A17.1 - 1984:

$$S\ max = \frac{V\ squared}{6.870} + 0.2560 \qquad \text{(SI units)}$$

$$S\ min = \frac{V\ squared}{19.63}$$

Fig 3 of APP:13/HB	V m/s	Min meters	Max meters
Min and Max sliding distances for ANSI	1	0.05	0.40
	2	0.20	0.84
	3	0.46	1.57
	4	0.82	2.58
	5	1.27	3.90
	6	1.83	5.50
	7	2.50	7.39
	8	3.26	9.57
	9	4.13	12.05
	10	5.09	14.81

APP:13/4

Let us now compare the sliding distances with counterweight attached, as calculated above, with the limits proposed by the ANSI code.

The ANSI code, Rule 1003.2d (2), prescribes that "...the safety will retard the car with the minimum assistance from the elevator driving machine...". Following strictly this wording we can select the calculated values in the column D2 of Fig 1 and Fig 2 above (no interference from sheave).

The Fig 4 (page 4) shows how the ANSI minimum and maximum compare with:

- dotted line: the limit corresponding to a zero deceleration in case of free fall. For any sliding distance exceeding this limit, the car would not stop in free fall with rated load.

- solid line: the limit corresponding to an initial deceleration of 2 m/s2 in the case of free fall. Because of possible variations of the friction factor values, this should be considered as the safe maximum for the sliding distance if the car must stop in free fall.

APP:13/6

Now if, instead of following strictly the wording of the ANSI code, we accept the testing procedure proposed by George Gibson in one of his reports to the WG 4 of ISO/TC 178, we should consider that, during the testing with counterweight attached, the machine is kept running at constant speed, trying to lift the counterweight.

Let us remark that, in APP:11, we came to the conclusion that the best way of knowing exactly the conditions of the test is to keep deliberately the machine running at constant speed until after the complete stop of the car. (please read APP:11/13/a)

This means that the values in the column D4 should be selected in the tabulations of Fig 1 and 2 (rated load, sheave rotating).

The Fig 5 (page 5) shows how the ANSI minimum and maximum compare with:

- dotted line: the maximum sliding distance for having a zero deceleration in the case of free fall with rated load.

- solid line: the limit corresponding to an initial deceleration of 2 m/s2 in the case of free fall with rated load.
  Because of the possible variations of the friction factor values, this should be considered as the safe maximum sliding distance.

- line made of small circles: the values corresponding to the CEN adjustment when tested in the same manner (values taken from the column D4 in Fig 3 of APP:11/HB).

Fig 4 of APP:13/HB
Curves based on the strict acceptance of the ANSI wording.
(no interference from traction sheave)

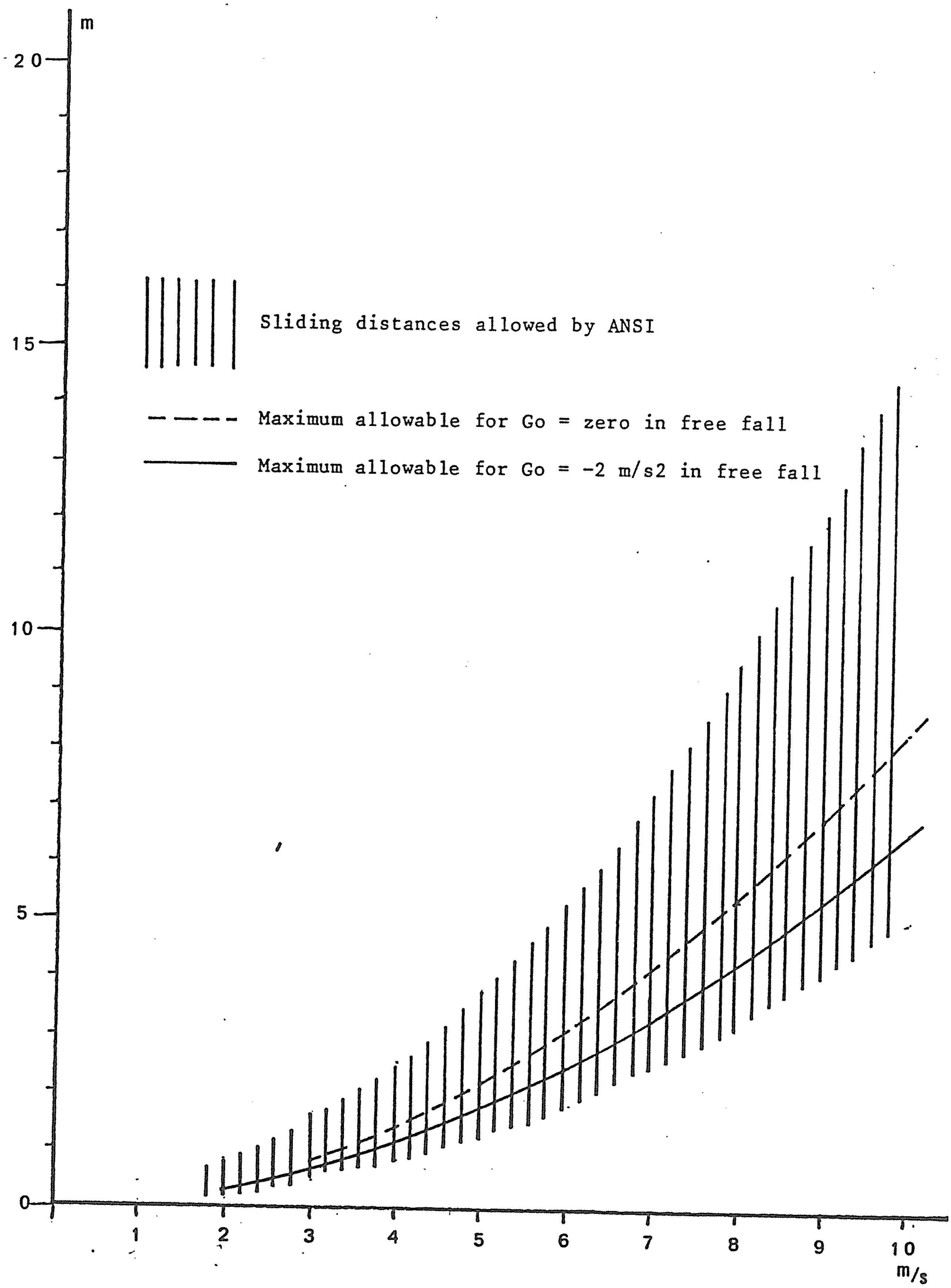
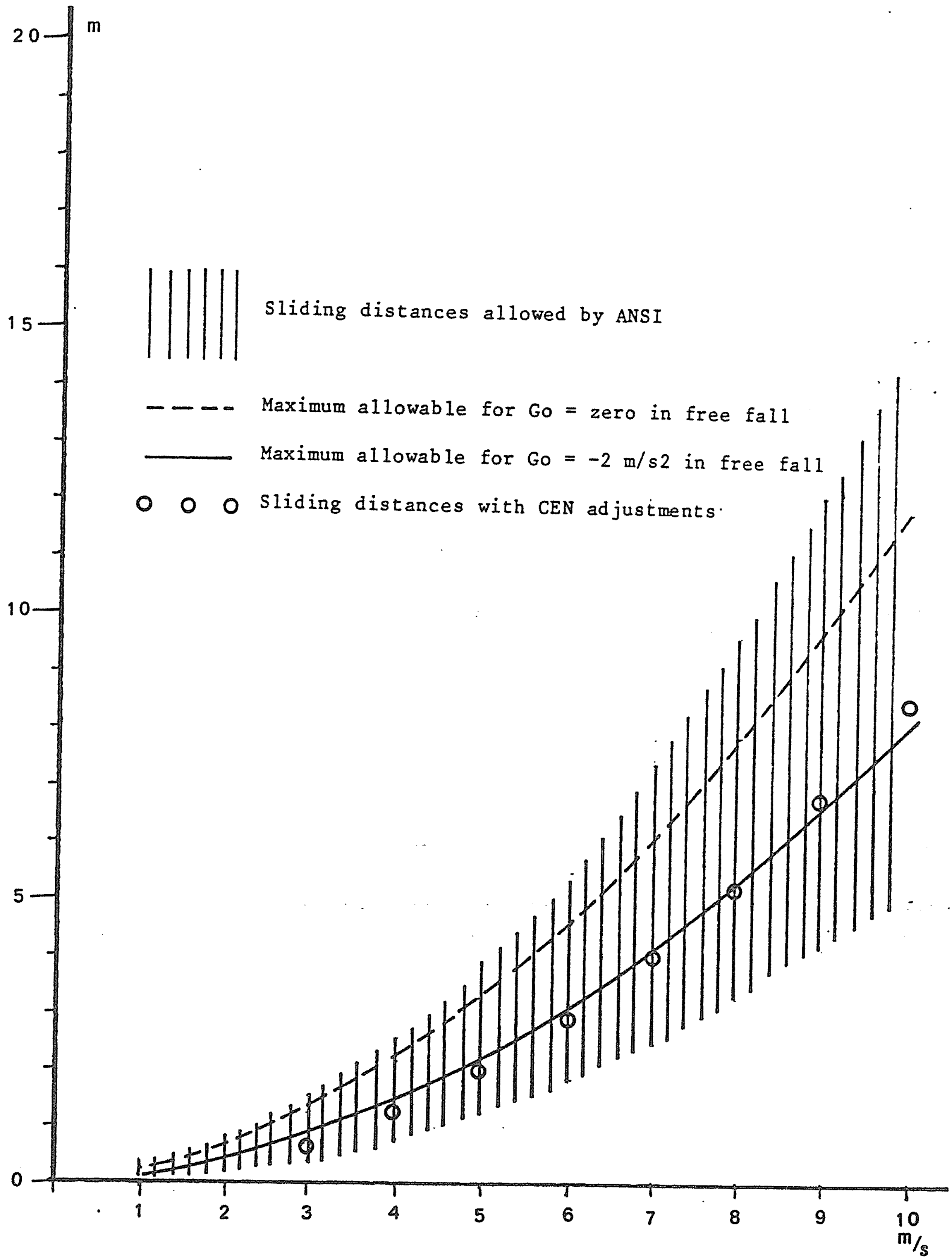


Fig 5 of APP:13/HB
Curves based on George Gibson's interpretation of the ANSI testing procedure (sheave kept rotating at constant speed and trying to lift the counterweight. This is probably the OTIS procedure in the USA)



APP:13/7: CONCLUSIONS

APP:13/7/a
The maximum sliding distances allowed by the ANSI code are much too long. With such an adjustment, the car with its rated load would not stop in the case of free fall (if that case has to be considered at all).

APP:13/7/b
Even if the test were made with 25% overload, the allowed sliding distances would still be a little too long to be really on the safe side.
(see the column D5 in Fig 2)

However, if no safety margin were required on the friction factor value, the allowed sliding distances would be acceptable.
(see the column D5 in Fig 1).

ALTERNATIVE TESTING METHOD for OVERSPEED GOVERNORS

APP:14/1:
This method, based on the principle of the Attwood machine, has advantages over the method proposed by the code:

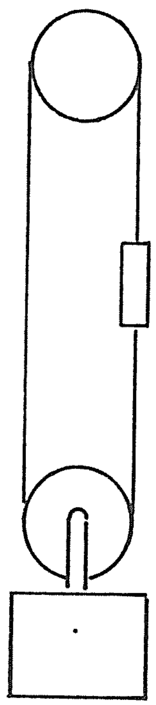
- it gives the values of the braking force with the ropes sliding as in a normal safety setting,
- it gives also an indication on the response time of the governor.

APP:14/2
A selected weight MW should be attached to the governor rope where it is normally fixed to the car.

The governor, the tensioning pulley and the associated ballast (or device) should be installed as normal.

The following masses should be either weighted or calculated:

Fig 1 of APP:14/HB



MW = mass of testing weight
MG = mass of governor rotating parts
MR = mass of governor rope
MP = mass of tensioning pulley
MB = mass of ballast

MT = MG+MR+MP

All masses converted to translation.
(if no better indications, take half the mass of the rotating parts)

MW should be selected so that:
(MW * 9.81) is equal to 2/3 of the expected minimum braking force

However, for the simplified test, (MW*9.81) should be exactly equal to the expected minimum.

APP:14/3
You must, of course, be able to measure the speed.
It should ideally be in relation to the time: set-up (a)
It can be in relation to the distance travelled: set-up (b)

At the limit, you could manage by just measuring the triggering speed and the total distance travelled from the time the weight MW is released to the time it stops: set-up (c)
But then, you need to know precisely the value of each mass.

In the following paragraphs, we will consider only the set-up (a) and (b).

For the set-up (c), the distance of acceleration has to be calculated from the known masses and substracted from the total distance which has been measured: it gives you the deceleration distance and you are back to the case of the set-up (b), but with less accuracy.

APP:14/4

Let us first analyse what happens during the acceleration period, this is the same whatever the type of governor.

When the testing weight MW is released, it accelerates down at:

$$Ga = Gn * \frac{MW}{MT} \qquad \text{(formula 1)}$$

If you have measured the time needed for reaching the triggering speed Vt, you can calculate Ga differently:

$$Ga = \frac{Vt}{T} \qquad \text{(formula 2)}$$

If you have measured the distance, Da, needed for reaching Vt, then

$$Ga = \frac{(Vt)\ \text{squared}}{2 * Da} \qquad \text{(formula 3)}$$

With all these formulae, you can, depending on what you were able to measure, calculate the missing values.

APP:14/5

Deceleration period in the case of the:

- separate "braking system" (see HB/F.4.1.(c)/b),
- special lining material with constant friction factor.

In this case, the braking force will of course be constant and the acceleration (negative) during the deceleration period will be

$$Gdecel = \frac{-\ Vt}{T} \qquad \text{(formula 4)}$$

T being here the time needed from triggering speed to zero speed.

If the distance is measured, the formula to be used is:

$$Gdecel = \frac{-\ (Vt)\ \text{squared}}{2 * DistDecel} \qquad \text{(formula 5)}$$

DistDecel = DD is the distance needed for decelerating.

Let us remark that the 2 results will be the same and also equal to the instantaneous value of Gdecel, since the deceleration is constant.

Having F = MW*Gn - BF
and F = MT*Gdecel

we can calculate the braking force BF:

$$BF = MW*Gn - Gd*MT \qquad \text{(formula 6)}$$

Gd being negative, BF will exceed the weight of the testing mass MW.

Deceleration in the case of the:

- separate "braking system",
- ordinary cast iron braking shoes.

In the present case, depending on what has been measured, we can apply:

- either the formula (5) and have the mean deceleration based on time,
- or the formula (6), then we have the mean deceleration based on distance.

We can, with one or the other, calculate a "mean braking force".

However, we need to know the minimum braking force because it is the one taken into consideration by the code for ensuring the driving of the safety gear wedges. We need also to know the maximum, in order to check the design of the actuating mechanism of the safety gear.

The tabulations (Fig 2 and Fig 3) give, in the last 2 columns, the factors by which to multiply the "mean braking force" for having:

- the minimum braking force: BFmin = BFmean * R1

- the maximum braking force: BFmax = BFmean * R2

Providing the mass of the testing weight is selected to have MW*Gn between 1/2 and 2/3 of the smallest braking force, R1 and R2 do vary with the setting speed, but not much with any other factor (it has been verified).

Of course, the smallest braking force is the one which has to meet the code requirements. It is happening when the speed is at its maximum, i.e. at the initial time of the deceleration.

Trusting that the manufacturer knows his equipment, the testing weight should be selected to be 1/2 or 2/3 of the announced braking capacity. (see later the check-points for the testing procedure)

Because adjusting the minimum braking force requires only a proper selection of the force applying the the brake shoe, the computer program, for preparing the tabulations, has been built up on the assumption that the initial (i.e. also the minimum) braking force was exactly equal to the required minimum.
It means a different adjustment for each speed.
I selected 300 N as minimum available pull, but selecting another value would not change the conclusions.

The tabulations were made primarily for giving the values of R1 and R2. However the other columns give valuable indications. For example, the columns "Dist" and "Time" give an idea of the needs for the test.
The time and distance needed for the period of fall must be added.

Please note that, at high speeds, the initial friction factor being low, a strong force would be needed for applying the brake. It means that the equipment would be bulky and that the brake shoes need to be carefully designed for avoiding deformation of the governor rope. The maximum tensile force is about twice the minimum one and this could complicate the actuating mechanism design. . All this justifies the use of better braking materials for high speed applications.

Fig 2 of APP:14/HB
TABULATION to be used if the deceleration is due to a separate brake fitted with cast iron brake shoes.
Case of the Mean Deceleration based on the TIME.

The values of the initial braking force and of the initial deceleration are the same for all speeds because of the assumption.

MG = 3	MR = 6	MP = 2	MB = 20
MW = 20	VT = var	BF = -300	CS = .01

Acceleration during the fall = 6.33 m/s2
Initial deceleration = -3.35 m/s2
Minimum braking force = -300 Newtons

Vtrigg	Gmean	BFmean	Gmax	BFmax	Dist	Time	R1	R2
1.00	-4.0	-320	-4.5	-336	0.13	0.25	0.94	1.05
2.00	-4.3	-331	-5.8	-375	0.49	0.46	0.91	1.13
3.00	-4.8	-344	-6.9	-411	1.05	0.63	0.87	1.20
4.00	-5.1	-355	-8.1	-446	1.79	0.78	0.84	1.26
5.00	-5.4	-363	-9.4	-487	2.69	0.93	0.83	1.34
6.00	-5.7	-372	-10.5	-522	3.76	1.06	0.81	1.40
7.00	-5.9	-379	-11.8	-561	4.98	1.19	0.79	1.48
8.00	-6.1	-386	-12.9	-596	6.35	1.31	0.78	1.55
9.00	-6.3	-391	-14.1	-634	7.87	1.43	0.77	1.62
10.00	-6.5	-397	-15.1	-663	9.55	1.54	0.75	1.67

Fig 3 of APP:14/HB
TABULATION to be used if the deceleration is due to a separate brake fitted with cast iron brake shoes.
Case of the Mean Deceleration based on the DISTANCE

The values of the initial braking force and of the initial deceleration are the same for all the speeds because of the assumption.

MG = 3	MR = 6	MP = 2	MB = 20
MW = 20	VT = var	BF = -300	CS = .01

Acceleration during the fall = 6.33 m/s2
Initial deceleration = -3.35 m/s2
Minimum braking force = -300 Newtons

Vtrigg	Gmean	BFmean	Gmax	BFmax	Dist	Time	R1	R2
1.00	-3.8	-315	-4.5	-336	0.13	0.25	0.95	1.07
2.00	-4.1	-322	-5.8	-375	0.49	0.46	0.93	1.16
3.00	-4.3	-329	-6.9	-411	1.05	0.63	0.91	1.25
4.00	-4.5	-335	-8.1	-446	1.79	0.78	0.90	1.33
5.00	-4.6	-340	-9.4	-487	2.69	0.93	0.88	1.43
6.00	-4.8	-345	-10.5	-522	3.76	1.06	0.87	1.51
7.00	-4.9	-349	-11.8	-561	4.98	1.19	0.86	1.61
8.00	-5.0	-352	-12.9	-596	6.35	1.31	0.85	1.69
9.00	-5.1	-356	-14.1	-634	7.87	1.43	0.84	1.78
10.00	-5.2	-359	-15.1	-663	9.55	1.54	0.84	1.85

Deceleration in the case of the braking resulting from the wedging of the rope in a narrow V groove made of cast iron

Applications are generally limited to rated speeds below 2 m/s because:

- as soon as the car decelerates, some traction is lost in the governor pulley groove; if the car deceleration is very high, the rope might even become slack over the governor wheel and the rope pull downwards on the actuating mechanism tending to disengage the wedges or rollers (instantaneous safety gears are normally designed in view of that, not always the progressive ones).
- blocking abruptly the governor rotating masses at high speeds can be detrimental. The peak forces are unknown, except if a kind of brake or of buffer is provided; but then it becomes complicated and the use of an external brake is more convenient,
- getting enough traction in the wheel groove when the rope is sliding at high speed is not so easy. Keeping it after some wear in the groove becomes problematic.

Only narrow V grooves are used for this application. Once the angle Gamma of the V has been selected, the adjustment of the traction in the groove can be done only by changing the mass of the tensioning ballast. Contrary to the assumption made above for the separate braking systems, the computer program built up for the present application is based on keeping the same adjustment for all the speeds. However, we made several runs with different ballasts to have an idea of the impact of this factor.

The program is a specific application of the program developed in the Appendix APP:11/HB where all the steps are explained.
The selected example is a governor existing on the market.

The falling mass selected for the present test is MW = 30 kg because the minimum pull considered here is 450 N.
The angle of the V (Gamma) is 30° (wedging factor C3=3.86 for new groove).
The angle of wrap is 180° (3.14 Rad). The mass of the ballast is 30 kg.

As written above, this type of governor is normally not used for triggering speeds exceeding 2 m/s and there is no need to consider speeds up to 10 m/s in the tabulation. However, we calculated the values which could be expected at high triggering speeds to investigate the problems specific to the friction in the groove.

In the Fig 4, which is for an intact new groove, you can see that the minimum braking force falls under the declared value when the speed reaches 2 m/s. At 6 m/s, the mass would not even start decelerating.

But, the deceleration of the test mass has an effect on the calculated traction. The force really available at zero deceleration is higher; we will, in APP:15/HB give the formulae for calculating it on the basis of the test results. In reality, the pull would still be higher in the case of free fall because of the inertia effect due to the acceleration of the car.

The only value which will not be affected, is the limit of 6 m/s because for a gliding speed of just under 6 m/s, the deceleration is zero.

Even after a slight wear, corresponding to a sinking of the rope of 0.1 mm only, the value of the wedging factor C3 drops from 3.86 to 3.2. as you can check in the Appendix APP:05/HB, Fig 15 and 17.

Fig 4 of APP:14/HB
TABULATION of values corresponding to a narrow V groove (Gamma = 30°) in a cast iron sheave.
Mean values based on time.

The values of the final braking force (which is also the maximum because the speed is nearing zero) is the same whatever the speed because nothing is changed in the adjustment from one speed to another.

MG = 3	MR = 6	MP = 2	MB = 30
MW = 30	CS = .01	$\boxed{C3 = 3.86}$	ALpha = 3.14

Acceleration during the fall = 7.18 m/s2
Final deceleration = -8.04 m/s2
Maximum braking force = -584 Newtons

Vtrigg	Gmean	BFmean	Gmin	BFmin	Dist	Time
1.00	-6.2	-519	-5.7	-499	0.08	0.16
2.00	-5.4	-489	-3.9	-435	0.42	0.37
3.00	-4.3	-451	-2.6	-387	1.23	0.69
4.00	-3.3	-414	-1.5	-349	3.04	1.20
5.00	-2.3	-376	-0.7	-318	7.59	2.19
6.00	The governor is not able to decelerate the mass MW					

Fig 5 of APP:14/HB
Tabulation showing what happens to the figures of Fig 4 after only a slight wear in the groove. It justifies the insistence of the TUV for testing the governor with the rope in the regular groove instead of a special test groove with reduced diameter as used in some countries.

MG = 3	MR = 6	MP = 2	MB = 30
MW = 30	CS = .01	$\boxed{C3 = 3.2}$	ALpha = 3.14

Acceleration during the fall = 7.18 m/s2
Final deceleration = -4.48 m/s2
Maximum braking force = -455 Newtons

Vtrigg	Gmean	BFmean	Gmin	BFmin	Dist	Time
1.00	-3.4	-418	-2.7	-393	0.16	0.29
2.00	-2.5	-384	-1.4	-346	0.95	0.80
3.00	-1.5	-348	-0.5	-311	4.06	2.00
4.00	The governor is not able to decelerate the mass MW					

The Fig 5 , shows that the required traction is not reached, even at 1 m/s, and that the mass would keep falling at 4 m/s.
The only purpose of this Fig 5, is to show the impact of a slight wear.

The Fig 6 and Fig 7, next page, will be needed for APP:15/HB.
The Fig 6 gives the same informations as Fig 4 with some intermediate speeds. The mean values are calculated from the time base.

The Fig 7 gives the same informations as Fig 6, but the mean values are calculated from the distance base. Both are limited to a speed of 3 m/s which is more than needed in usual applications.

Fig 6 of APP:14/HB
Tabulation giving the ratios R1 and R2 when the mean values are calculated from the time base. This tabulation is for a new groove.

MG = 3	MR = 6	MP = 2	MB = 30
MW = 30	CS = .01	C3 = 3.86	ALpha = 3.14

Acceleration during the fall = 7.18 m/s2
Final deceleration = -8.04 m/s2
Maximum braking force = -584 Newtons

Vtrigg	Gmean	BFmean	Gmin	BFmin	Dist	Time	R1	R2
1.00	-6.2	-519	-5.7	-499	0.08	0.16	0.96	1.12
1.50	-6.0	-510	-4.7	-465	0.21	0.25	0.91	1.14
2.00	-5.4	-489	-3.9	-435	0.42	0.37	0.89	1.19
2.50	-4.9	-471	-3.2	-410	0.74	0.51	0.87	1.24
3.00	-4.3	-451	-2.6	-387	1.23	0.69	0.86	1.29

Fig 7 of APP:14/HB
Tabulation giving the ratios R1 and R2 when the mean values are calculated from the distance base. This tabulation is for a new groove.

MG = 3	MR = 6	MP = 2	MB = 30
MW = 30	CS = .01	C3 = 3.86	ALpha = 3.14

Acceleration during the fall = 7.18 m/s2
Final deceleration = -8.04 m/s2
Maximum braking force = -584 Newtons

Vtrigg	Gmean	BFmean	Gmin	BFmin	Dist	Time	R1	R2
1.00	-6.2	-507	-5.7	-499	0.08	0.16	0.98	1.15
1.50	-6.0	-488	-4.7	-465	0.21	0.25	0.95	1.20
2.00	-5.4	-467	-3.9	-435	0.42	0.37	0.93	1.25
2.50	-4.9	-446	-3.2	-410	0.74	0.51	0.92	1.31
3.00	-4.3	-426	-2.6	-387	1.23	0.69	0.91	1.37

APP:14/8
I believe that, up to now, the tests for measuring the force which a governor could exert on the governor rope have generally been made statically and the reports ignored the loss of traction due to the sliding of the rope in the groove or in the brake.

However, when these values were calculated (and not measured), they were calculated with a friction factor of 0.09 which, in our formula, corresponds to a sliding speed of about 3 m/s. For the usual applications, this was on the safe side but not good enough for the very high speeds.

APP:14/9
The procedures for testing the governors on this principle are given, step by step, in a separate appendix for easier reference: the Appendix APP:15.

There are several approaches, some very simple, some more sophisticated. I believe that only the simple ones are of practical interest.
(see the conclusions of APP:15)

ALTERNATIVE TESTING METHOD for OVERSPEED GOVERNORS

FORMULAE and CHECK-LIST

APP:15/1:
This method, based on the principle of the Attwood machine, has some advantages over the method proposed by the code:

- it gives the values of the braking force with the ropes sliding as in a normal safety setting,
- it gives also an indication on the response time of the governor.

The behaviour of the various types of governors has been analysed in the preceeding Appendix APP:14/HB. We shall, in the present one, review exactly what has to be done, and in what order, for this testing method.

First of all, the governor should be rigged as indicated in HB/APP:14/1. The possible vertical travel of the mass MW should be at least 3 times the gravity stopping distance. At the lower part of this possible travel, a bumper should be installed to avoid damaging the tensioning device if the mass does not decelerate enough.

Depending on the available measuring equipment and on the information we seek, the test can be conducted in various ways.
We shall successively review several possible procedures, from the simplest one to the most complete one:

Ref	measurements needed
APP:15/2	peak speed, one test mass.
APP:15/3	peak speed, adjustable test mass.
APP:15/4	peak speed, all masses (MW, MG, etc), total travel.
APP:15/5	instantaneous speeds versus distance, all masses (MW, MG, MR, MP, MB).
APP:15/6	instantaneous speeds versus time, all masses (MW, MG, MR, MP, MB).

As explained in APP:15/7, the first 3 ones, being simple to implement, are probably the most interesting from a practical point of view.

APP:15/2 measurements: peak speed, one test mass

APP:15/2/a
The weight of the test mass MW should be selected equal to the minimum required by the code as per EN/9.9.4. Usually, this will be 30 kg.

APP:15/2/b
Release this mass from the highest possible point.

At first, the mass will accelerate until reaching the tripping speed. Make a note of the peak speed reached in the process.

With this set-up, the speed may be measured on the governor wheel.

APP:15/2/c
If, after triggering, the mass starts decelerating or, at least, does not keep accelerating down, the governor braking force is at least equal to the required minimum. If not, the governor does not meet the specifications.

APP:15/2/d
If the braking force was sufficient, repeat the drop test as per APP:15/2/b a number of times (for example 20 times as per EN/F.4.2.2.2).

The range of triggering speeds could be somewhat higher than the range of speeds allowed in EN/9.9.1, because of the "response distance".
See HB/F.4.2.2.2 Fig 38. Remember however that here the acceleration is lower than Gn; the speed increase due to the response distance is smaller.
The spread of the triggering speeds gives an idea of the variable response distance due to the spacing of the teeth in the ratchet.
The spread would be larger if the acceleration were that of gravity (as in free fall of the car), see the HB/Fig 38 mentioned earlier.

APP:15/2/e
This simple testing procedure cannot give an idea of the minimum and maximum values of the actual braking force exerted by the governor.

However, it checks the 2 most important requirements i.e.:

- that the tripping speed is about what it should be,
- that the braking force is at least equal to the required minimum.

APP:15/3 <u>measurements: peak speed, adjustable test mass</u>

APP:15/3/a
First, adjust the mass to the minimum required by the code as per EN/9.9.4, then proceed as per APP:15/2 above.

APP:15/3/b
Then increase gradually the test mass until reaching practically the deceleration zero.

Note the mass at that time. It is the greatest force which can be expected at the time of triggering (with the rope sliding at the tripping speed VT), all acceleration effects excluded.

Safety gears requiring up to this force for being actuated, may be associated with the governor.(the inertia effect of the rope will help pulling in the case of free fall; it comes as a safety margin).

APP:15/3/c
In the case of a governor using a separate "braking system" with special linings having a nearly constant friction factor, the force will remain constant until the complete stop.

APP:15/3/d
In all the cases where the braking is generated by the friction of the rope on cast iron, the force will increase as the rope slows down because of the increase in the value of the friction factor.

We have measured (in APP:15/3/b) the force at the sliding speed VT.

In the case of a governor using a separate braking system, multiply this force by the ratio of the friction factors to have the force at zero speed:

(Static Force) = (Force when slipping) * (1+0.125*VT)

$$\text{Of course, Friction factor} = \frac{0.125}{1 + 0.125*V}$$

APP:15/3/e
In the case of a governor using the wedging of the rope in a V groove for braking, you may proceed as in APP:15/3/d if the governor is to be used for lifts with a rated speed lower than 2 m/s.

In fact, the influence of the friction factor is compounded by the traction formula, but the approximation is acceptable for these speeds.

APP:15/3/f
This still relatively simple procedure gives the following informations:

- it checks that the tripping speeds are about what they should be,
- it gives, with a good approximation, the force which will be exerted by the governor at the moment of triggering.
- it gives, with an acceptable approximation, the maximum pull one has to reckon with, for the selection or the design of the safety gear.

It should be noted that this increase in the braking force is not cumulative with the inertia effect: when the speed is nearing zero, the safety gear is normally braking harder than the governor.

APP:15/4 <u>measurements: peak speed, all masses, total travel</u>

APP:15/4/a
The first thing to do is to request, measure or calculate the mass of the test mass (MW), of the governor rope (MR), of the governor rotating parts (MG) and of the tension pulley (MP).

> NOTE: the actual mass of the governor rotating parts and the mass of the tensioning pulley shall be converted to translation at the rope speed. If you have not enough details on the design, take half their total mass as an approximation (if the governor is not geared).

APP:15/4/b
At first, a small test weight should be selected and about half of the intended tests will be made with that small weight.
The weight should of course be such that the tripping speed can be reached before hitting the bumper at the end of the available travel.

Because of the low acceleration, the average of the tripping speeds corresponds to the adjustment as understood in the EN code.

APP:15/4/c
Then, the test weight (MW*Gn) should be selected to be about 2/3 of the expected braking force.

For convenience we will calculate: $MT = MW + MG + MR + MP$

APP:15/4/d
We can calculate the acceleration before the tripping of the governor:
This acceleration is constant and equal to:

$$Ga = Gn * \frac{MW}{MT}$$

APP:15/4/e
Proceed to the other half of the selected number of drop tests. Note each time the peak speed and the total travel from the point of release of the test mass MW to the place where it stops.

APP:15/4/f
Calculate the average tripping speed and the average travel.
Make a note of the highest and of the lowest tripping speeds.

APP:15/4/g
Using the average tripping speed and the value Ga of the acceleration calculated in APP:15/4/d, calculate the distance, Da, needed for accelerating the complete assembly:

$$Da = \frac{Vsquare}{2 * Ga}$$

APP:15/4/h
Substracting Da from the average travel, you get the distance left for the deceleration: Dd.

The distances are calculated as if there were no friction in the governor assembly. The actual distance left for decelerating is probably shorter than the one calculated; this is to be on the safe side.

APP:15/4/i
Using the deceleration distance Dd, you can calculate the mean deceleration based on the distance, GDmean, by the formula:

$$GDmean = - \frac{Vsquare}{2 * Dd}$$

(NB: GD is negative because it is a deceleration)

APP:15/4/j
The corresponding mean braking force, BFmean, is given by the formula:

$$BFmean = GDmean*MT - MW*Gn$$

(NB:BF is negative)

APP:15/4/k
If the governor is of the separate "braking system" type, using a special lining, the braking force remains equal to BFmean from beginning to end.

In all other cases, you must calculate:

Minimum Brake Force = BFmean * R1

Maximum Brake Force = BFmean * R2

In the case of a separate braking system usin cast iron shoes, R1 and R2 must be taken from the Fig 3 of APP:14/HB.

If the governor is of the type using a grooved sheave, the value of R1 and R2 must be taken from the Fig 7 of APP:14/HB.

APP:15/4/1
The procedure APP:15/4 gives the same information as the procedure APP:15/3. The advantage of the present method is that you do not have to let the test mass slide to the bumper at high speed and the results are probably a little more accurate.

APP:15/5 <u>Measurements: speed versus distance, all masses</u>

APP:15/5/a
The measuring device must be a continuous recording system giving the instantaneous speed of the test mass at constant distance intervals.

The distance "tops" can, for example, be given by a perforated tape along the travel. The speed may not be taken from the governor sheave if the sheave is blocked when the governor is tripping.

APP:15/5/b
In this system you can read directly the distance needed for accelerating and the distance needed for decelerating from the recordings and make all kinds of cross-checks.

Calculate the average of all the recordings.
Having the distance needed for the deceleration, you follow the procedure APP:15/4 from APP:15/4/i to APP:15/4/1

APP:15/5/c
The advantage of the system over the previous one, is that the distances are measured separately, hence they are more accurate.

The disadvantage is that the measuring equipment is more expensive and more delicate. It is not convenient in the field.

APP:15/6 <u>Measurements: speed versus time, all masses</u>

Proceed as in APP:15/5 except that the mean deceleration can be simply calculated by dividing the average tripping speed by the the time needed for decelerating. This time can be read directly from the recordings.

As regards the values of R1 and R2, they should of course be taken from:
- the Fig 2 if the governor uses a separate brake,
- the Fog 6 if the braking is due to the traction in a V groove.

All kinds of cross-checks are possible for greater accuracy.

Note: this translation of the Belgian Standard E 52-017 (ed. March 1984) is not an official one.
All references to other specific local standards have been replaced by general indications.

DIVIDING-UP of the WORK

to be foreseen to meet the requirements of EN 81/Part I

Standards to be taken into consideration:

- EN 81/Part I
- all other relevant local standards.

TABLE of CONTENTS

1: INTRODUCTION - SCOPE

The present standard gives the dividing up of the work to be foreseen to meet the requirements of the lift code EN 81/Part I (electric lifts).

Most of the requirements of the lift code relate either to the material to be supplied by the lift contractor or to the way of installing it.
Some of the requirements clearly involve other trades (builders work).
But some supplies or construction work which could eventually be made by the lift contractor, are, in the current practice, requested from other specialised suppliers or contractors unless the tender or the contract clearly specifies which of these supplies and construction work will be part of the lift contract itself.

In order to avoid misunderstandings and disputes, the present standard defines what supplies and prestations should not be expected from the lift contractor unless clearly spelled in his contract.
The owner, or the general contractor, must of course place orders with other trades for what the lift contractor will not supply.

2: GENERAL PROVISIONS

The orders to be placed with other trades must take into account the indications given by the lift contractor in the Technical Dossier. (see Appendix C to the EN 81/Part I).

Consequently, it is recommended to request, in the lift contract, that the documents submitted when applying for the preliminary authorization do include the following:

- the general layout of the lifts,
- any other technical information usually required locally within the frame of the Appendix C of EN 81, but at least, the main characteristics of the distribution panel(s) to be installed in the machine rooms at the locations indicated on the general layout:
  - Electrical power.........A........V 3-phase (see EN/3.4.1)
  - Car lighting.............A........V per lift (see EN/3.4.2)
  - Lighting of the well (see below and EN/3.4.3)
- location and characteristics of the junction boxes (*) to be installed in the machine room.

(*) Note: junction box(es) where the circuits for the lighting of the well and of the car will be connected.

3: SUPPLIES & PRESTATIONS NOT TO BE MADE BY THE LIFT CONTRACTOR

unless clearly otherwise specified in the contract.

3.1: Administrative forms and applications

3.1.1 : Obtaining the necessary authorizations and permits for installing the lift in the building.

3.1.2 : Obtaining the authorization of the Fire Department.

3.1.3 : Requesting the electrical power supply.

3.2: Masonry and concrete work

3.2.1 : Building

All calculations related to the building.

3.2.2 : Machine and pulley room (if any)

3.2.2.1: Support points for the hoisting machine and for the pullies (forces given in the Technical Dossier)

3.2.2.2: Fixing of beams and hooks for handling the heavy equipment as indicated in the Technical Dossier (see EN/6.3.1).

3.2.2.3: Building of walls, floors and ceilings meeting the requirements of EN/6.3.1.1 (material not favouring the creation of dust) and EN/6.3.1.2 (floors made of non-slip material).

3.2.2.4: Construction of accesses in conformity to EN/6.2

3.2.2.5: Noise protection according to local requirements.

3.2.3 : <u>Well shafts</u>

3.2.3.1: Construction of walls, floor and ceiling (EN/5.3):
- using materials not assisting the creation of dust,
- conforming to the requirements of the Fire Department,
- able to support the loads applied by the guide rails, the doors, etc..at the places indicated in the Technical Dossier. The Technical Dossier also gives the values of the forces.

The layout drawings, in the Technical Dossier, give the minimum dimensions (with tolerances) which are needed for installing the lift in a well perfectly plumb.

3.2.3.2: Installing, at the locations required by the Technical Dossier, of the dividing beams beween adjacent wells, if these beams are made of concrete.

NOTE: To the contrary, if steel beams have been selected, they should be supplied and installed by the lift contractor. The customer must specify his choice in the request for bids.

3.2.3.3: Construction of masonry or concrete dividing walls (if any) between adjacent lift wells.

3.2.3.4: Providing, at the locations indicated in the Technical Dossier, bays for the installation, by the lift contractor, of the landing doors, emergency and inspection doors and traps.

NOTE: As regards the fire resistance of the door assemblies, the limit of what may be assimilated to the landing doors should be defined by the National specifications by analogy with EN/F.2.3.2 and related note at the end of EN/F.2 in the EN 81.
Anything exceeding these dimensions shall be assimilated to the walls as regards the fire resistance.

3.2.3.5: Fixing and marking durably the level of the finished floor at each landing as well as the axis of the doors.

NOTE: Let us remark that the axis of all the landing doors of a given lift must be exactly on the same plumb line and that, on the other hand, the lift must fit in the minimum space mentioned in 3.2.3.1. Fixing these axis is a delicate matter, specially if the wall decoration is modular. This type of decoration should be avoided for the walls around all lift doors.

3.2.3.6: Providing the holes and conduits required for the ventilation. (see EN/5.2.3)

3.2.3.7: Providing the holes and conduits for evacuating fumes and gases if required by the local rules for fire protection.

3.2.3.8: Providing the orifices required by the Technical Dossier between the lift well and the machine and/or pulley room.

3.2.3.9: Masonry or concrete work for filling up the walls after installing the landing doors and any other lift door or trap.

3.2.4 : Pits

3.2.4.1: Providing a pit with the measurements indicated in the Technical Dossier. The walls and the bottom shall be able to resist the forces indicated in the Technical Dossier for the reaction of guide rails, buffer(s), compensators, etc..

3.2.4.2: If there is an accessible space below the pit, providing a floor meeting the specifications of EN/5.5.2 and providing a solid pier extending down to solid ground if and where required in the Technical Dossier.

3.2.4.3: Protection of the pit against water infiltration, taking into account the requirements mentioned in 3.2.4.1 and the type of anchoring to be used by the lift contractor. (see Technical Dossier).

NOTE: the same applies to any part of the lift well which might be exposed to water infiltration.

3.2.4.4: Providing the means for draining the water if needed.

3.3: Steel works

3.3.1 : Supply of the hooks and beams mentioned in 3.2.2.2.

3.3.2 : Supply and installation of the doors and/or trap doors to the machine room and to the pulley room (if any) with adequate locking devices, all meeting the requirements of EN/6.3.3.

3.3.3 : Supplying and installing the enclosure and the access door in the case where the clause EN/6.1.2.2 is applicable.

NOTE to 3.3 as a whole:
The lift contractor shall himself supply and install the emergency and inspection doors for the well and the pit with adequate locks and contacts as required by EN/5.2.2. He shall aso supply and install the partitions required by EN/5.6 as the case may be.

3.4: Supply and installation of electrical material

3.4.1 : As regards the power supply to the motor, supply and installation:
- of the watt*hours meter (if separate recording),
- of the protection devices for the main feeder as required by the local rules,
- of the main feeder coming from the building busbar,
- of the distribution panel at the arrival in the machine room including,for each lift, adequate protection of the feeder(s) going to each control cabinet (against short-circuits),

The data in the Technical Dossier shall be taken into consideration for selecting the proper equipment.

NOTE: the requirements of the EN code do apply to the main switches and dependent circuits (see EN/13.1.1.1).

3.4.2 : As regards the supply to the lighting devices and socket outlets, supply and installation of:

- the feeder (if separate) coming from the building bus-bar, including the measuring devices (if required) and the adequate protections against overload and short-circuits (see EN/13.6.3.2),
- the distribution panel in the machine room including:

  a) a switch controlling the supply to the circuit of the machine room, well and pit (see EN/13.6.3.2),
  b) per lift, a switch controlling the supply to the circuit of the car (see EN/13.6.3.1).

Each circuit shall have an adequate electrical protection in line with the data of the Technical Dossier.

NOTE: the requirements of the EN code do apply to the circuits (b) to the car whereas the requirements for the local power distribution do apply to the circuit (a).
(see 13.1.1.1 and 13.1.1.2 of EN 81/Part I)

3.4.3 : Supply and installation, in the machine and/or pulley romm, of the circuit(s) controlled by the switch(es) mentioned in 3.4.2 (a) above as well as of the related lighting fixtures and socket outlets and of a junction box located as indicated in the Technical Dossier.

The lighting fixtures shall be selected to have the illumination required in EN/6.3.6.

NOTE: The lift contractor himself takes care of:
a) the circuit which, from the above junction box, feeds the lighting fixtures and socket outlets located in the well and in the pit as well as these fixtures and outlets.
b) the circuit which, from the switch mentioned in 3.4.2 (b), feeds the lighting and socket outlets of the car.

3.4.4 : Supply and installation of the emergency power sources, which might be required by the owner or by the local authorities, for the power circuits.

NOTE: These emergency power sources are different from the ones required by the EN 81 code in the paragraphs 8.17.3 and 14.2.3.2 which are to be supplied by the lift contractor.

3.4.5 : Supply and installation of:
- the devices for commuting from the normal power to the emergency power circuit,
- the adequately protected feeder for the emergency power up to and including the panel in the machine room with a switch and the protections for the subsequent circuitry.

NOTE: The devices for the sequence starting-up of the lifts according to a preselected program are to be supplied and installed by the lift contractor. But the lift contractor must give all necessary indications when requesting bids.

3.4.6 : Earthing conforming to the local requirements and circuitry up to and including and collector panel in the machine room.

3.4.7 : Supply and installation of the adequate lighting fixtures on the landings (see requirements of EN 81 parag 7.6.1).

3.5: Signaling and alarm devices

3.5.1 : Wiring and connection of the alarm device from 2 terminals located either at the base of the well or in the machine room (to be agreed before hand), to the location selected for the bell, the intercommunication device or equivalent.

The lift contractor supplies the device meeting the requirements of EN 81/14.2.3 and installs the liaison between the lift car and the 2 terminals mentioned above.

3.5.2 : It will be the same for any special signal devices which might be included in the request for bids (for carrying the information to a central general alarm station for example).

3.6: Ventilation, heating, air conditioning

3.6.1 : Supplying and installing the equipment necessary for maintaining the temperature of the machine room between + 5 and + 40C, taking into account the heat dissipated by the machine(s) (to be indicated by the lift contractor) and the requirements relating to the ventilation and the smoke evacuation (see EN 81/6.3.5 and 6.4.6).

3.6.2 : Supplying and installing the equipment which might be necessary for the ventilation (see also 3.2.3.6 above) and for the heating of the well (see EN 81/5.8).

3.7: Fire protection

PRELIMINARY NOTE:
It has been said earlier (in 3.1.2) that the owner has to apply for the authorization from the Fire Protection Authorities (if required) and make sure that they are available in due time.

The owner shall specify, when requesting for bids, the fire protection rules which apply in this specific building and ask the lift contractor to include in his bid the corresponding control devices (amongst the ones described in G.3 and G.4 of the App G to EN 81)

3.7.1 : Supply and installation of the temperature rise detectors mentioned in G.2 of EN 81 as well as the wiring and conducts carrying the information up to 2 terminals in the machine room.

3.7.2 : Supply and installation of the signal emitting device mentioned in G.3.5.2 of EN 81 as well as the conducts and wiring carrying the signal up to 2 terminals in the machine room.

3.7.3 : Supply and installation of the signals mentioned in the second paragraph of G.4 in EN 81.

NOTE: To the contrary, the Fireman's switch mentioned in the 3d paragraph of G.4 in EN 81 shall be supplied and installed by the lift contractor.

3.7.4 : Supply and installation of the automatic fire protection devices (if any). See G.5 in EN 81.

3.8: Examinations and tests before going into service

Selection and remuneration of the Inspector or Organization for making the examinations and tests required in EN 81 (Appendix D). The selection shall be made according to the local rules.