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Comparison of worldwide lift safety standards —

Part 2: Hydraulic lifts (elevators)

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TECHNICAL REPORT

ISO/TR 11071-2

Second edition
2006-04-15

Comparison of worldwide lift safety standards —

Part 2: Hydraulic lifts (elevators)

Comparaison des normes mondiales de sécurité des ascenseurs —

Partie 2: Ascenseurs hydrauliques

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Foreword

ISO (the International Organization for Standardization) is a worldwide federation of national standards bodies (ISO member bodies). The work of preparing International Standards is normally carried out through ISO technical committees. Each member body interested in a subject for which a technical committee has been established has the right to be represented on that committee. International organizations, governmental and non-governmental, in liaison with ISO, also take part in the work. ISO collaborates closely with the International Electrotechnical Commission (IEC) on all matters of electrotechnical standardization.

International Standards are drafted in accordance with the rules given in the ISO/IEC Directives, Part 2.

The main task of technical committees is to prepare International Standards. Draft International Standards adopted by the technical committees are circulated to the member bodies for voting. Publication as an International Standard requires approval by at least 75 % of the member bodies casting a vote.

In exceptional circumstances, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example), it may decide by a simple majority vote of its participating members to publish a Technical Report. A Technical Report is entirely informative in nature and does not have to be reviewed until the data it provides are considered to be no longer valid or useful.

ISO/TR 11071-2 was prepared by Technical Committee ISO/TC 178, *Lifts, escalators and moving walks*.

This second edition cancels and replaces the first edition (ISO/TR 11071-2:1996), and amendment 1 (ISO/TR 11071-2:1996/Amd. 1:1999), which have been technically revised.

ISO/TR 11071 consists of the following parts, under the general title *Comparison of worldwide lift safety standards*:

— *Part 1: Electric lifts (elevators)*

— *Part 2: Hydraulic lifts (elevators)*

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Introduction

Introduction to 1996 edition

At the 1981 plenary meeting of ISO/TC 178, work was started on a comparison of CEN standard EN 81/1 with the American, Canadian, and USSR lift safety standards. In 1983, Working Group 4 was officially formed to carry out the task of preparing cross reference between the relevant sections of these standards and to analyze the differences on selected subjects. The goal at that time was to prepare a technical report which would provide reference information to assist national committees when reviewing and revising individual standards which may initiate a gradual convergence of the technical requirements. In 1984, the study was expanded to include the Council for Mutual Economic Assistance (CMEA) safety standard. That report, ISO/TR 11071-1, *Comparison of worldwide lift safety standards — Part 1: Electric lifts (elevators),* was published 1990-12-01.

In 1989, the charge to WG 4 was expanded to include hydraulic lifts. Since there was no standard for hydraulic lifts in the Russian Federation, and the CMEA standard was being phased out of use, this Part 2 of the comparison is generally limited to the ASME, CEN, and CSA standards. The Japan Elevator Association was invited to add their standards to this comparison, however, no response to this request was received.

This report is intended to aid standards writers in developing their safety requirements, and to help standard users understand the basis for the requirements as they are applied throughout the world.

This report is not intended to replace existing safety standards. Conclusions are arrived at in some cases, but only where there is unanimity amongst the various experts. In other cases, the reasons for the divergent views are expressed.

This report must be read in conjunction with the various safety standards, as it was often necessary to summarize the requirements for the sake of clarifying the comparisons. Further, the information contained in this report does not necessarily represent the opinions of the standards writing organizations responsible for the development of the safety standards which are being compared, and they should be consulted regarding interpretations of their requirements (see Annex B).

Introduction to this edition

After the original publication (1996) of this technical report, including American, Canadian and European data and thereto Supplement 1 (1999-08-01), which added Australian and Japanese data, has been revised or amended. The recommendations in the form of “agreed upon points” stated in the first edition have also affected the revisions of the national standards.

The original report and amendment have been widely used by lift industry and standards writing organizations, including the ISO Technical Committee 178. Users have expressed need for an updated and consolidated version of the document, in particular the comparison tabulations. With the Resolution 208/2002, the ISO/TC 178 requested WG4 to consolidate original publications, including Supplements and “to update comparison tables in ISO/TR 11071 with data from the most recently published standards for lifts and to republish both documents, Part 1 and Part 2 with updated tables and with minimum changes to the narrative sections”.

The narrative sections of the original publication, in particular assumptions, historical backgrounds, observations and suggestions as well as the points agreed upon were the result of extensive work of the ISO/TC 178 Working group 4. ISO/TC 178 is currently working on a new series of ISO documents under the general title *Safety requirements for lifts (elevators)*. In that process the updated comparison tables are being used as reference. Extensive work on complete re-write of the narrative sections was not deemed necessary. However, republication of the text with only minor editorial changes would help readers to understand the background to the safety concerns being addressed in the current national standards. However, because of

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recent (2000) harmonization of ASME and CSA Codes, it was necessary to replace the quoted rule numbers with those in the current Codes In most sentences the ASME and CSA. I some other cases quoted references are updated in a NOTE following the narrative section or sentence.

All quoted requirement referenced in all tables (CEN, ASME/CSA, Japan and SA) are up to date.

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Comparison of worldwide lift safety standards —

Part 2:
Hydraulic lifts (elevators)

1 Scope

This Technical Report consists of a comparison of the requirements of selected topics as covered by the following worldwide safety standards (excluding regional or national deviations):

a) CEN European Standard EN 81-2:1998, *Safety rules for the construction and installation of lifts — Part 2: Hydraulic lifts*;

b) ASME A17.1:2004, Safety Code for Elevators and Escalators and CSA B44:2004, Safety Code for Elevators;

c) Japan - Building Standard Law of Japan (BSLJ);

d) Standards Australia:

— AS 1735-1:2003, *Lifts, Escalators and Moving Walks — Part 1: General Requirements*;

— AS 1735-3:2002, *Lifts, Escalators and Moving Walks — Part 3: Passenger and Goods Lifts — Electro-hydraulic*.

This Technical Report applies to hydraulic lifts only, both of the direct and indirect acting type.

It should be noted that, in addition to the above listed standards, lifts must conform to the requirements of other standards (for example, standards covering mechanical, structural, and electrical equipment; building codes, and environmental regulations). Some of the standards will be referred to in this Technical Report.

2 Terminology

2.1 Lifts and elevators

2.1.1 The CEN term *lift* corresponds to the ASME and CSA term *elevator*. These terms are used inter-changeably in this report.

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2.1.2 For the purposes of this report, unless otherwise specified, the terms *passenger lift* and *freight lift* correspond to the following terms used in other Standards:

Term used in this report	Correspond to terms used in the following standards*	
	CEN	ASME and CSA
Passenger lift	Lift except goods passenger lift	Passenger elevator & freight elevator permitted to carry passengers
Freight lift	Goods passenger lift**	Freight elevator

* See the definitions in the applicable Standards.

** This term is used only to enable comparisons to be made later in this report. It does not indicate recognition of the term "freight lift" by CEN.

2.2 Hydraulic terminology

2.2.1 Difference

There are some notable differences in the standards respecting hydraulic lift terminology as shown in the Table 1, Column A and B.

2.2.2 Agreed-upon points, re: hydraulic terminology

The differences should be eliminated or minimized through proposed changes to ASME and CSA Standards, as shown in Table 1, Column D.

If approved by ASME and CSA Committees, the proposed changes would eliminate major differences between CEN and North American Standards.

Column C gives the description of the equipment that a term (listed in Column A, B, or D) embraces.

In addition to "hydraulic machine", ASME and CSA propose to introduce the term "hydraulic driving machines" hydraulic driving machines". The terms are needed to differentiate between "electric" and "hydraulic" driving machines all covered in one ASME and CSA Standard. This is not necessarily applicable to CEN, as the electric and hydraulic lifts are covered by two separate standards.

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2.2.3 Terminology in this report

In this report, the CEN terminology will be used, with the ASME and CSA terms in brackets if different.

Table 1 — Hydraulic Terminology

Column A	Column B	Column C	Column D
CEN	ASME & CSA Current	Description	Agreed upon points: ASME & CSA proposed changes
Direct acting lift	Direct plunger hydraulic elevator	—	Direct acting hydraulic elevator [a]
Indirect acting lift	Roped hydraulic Elevator	—	No change
Machine	—	Pump, motor, valves	Hydraulic Machine [b]
Jack	Driving machine	Cylinder and ram	Hydraulic jack [c]
Ram	Plunger or piston	—	Plunger (ram) or piston
Base	Head/bottom (Includes plunger end cap as well)	Cylinder end cap	No change
Valves:			
Non-return	Check	—	No change
Pressure relief	Pump relief	—	No change
Direction	Control	—	No change
Rupture	ASME-Safety CSA-Rupture	—	No change

NOTE ASME and CSA adopted terms:

[a] "direct-acting".

[b] "hydraulic machine".

[c] "hydraulic jack".

2.3 Working pressure vs full load pressure

ASME and CSA use *working pressure* (WP), which is defined as the pressure at the hydraulic driving machine when lifting the car and its rated load at rated speed, or with class C2 loading, when leveling up with maximum static load.

CEN defines *full load pressure* (FLP) as the static pressure exerted at the piping directly connected to the jack, the car with the rated load being at rest at the highest landing level.

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CEN Annex K, clause K.1.1 recognizes that friction losses as a result of fluid flow are on the order of 15 %; thus a factor of 1,15 is included in their factor of safety determination.

NOTE CEN reference to “Clause 12, NOTE 1” in this clause and through the 1996 edition of this document has been replaced with reference to “Annex K, Clause K1.1” in this edition.

Thus, ASME WP = 1,15 x (CEN FLP).

2.4 Other terms

Additional terminology, where there is a difference between the CEN and the ASME and CSA standards, is shown in Table 2:

NOTE Since ASME and CSA are now harmonized they will be shown through this edition in a column under title “ASME and CSA” or “ASME/CSA”.

Table 2 — Terminology

CEN	ASME and CSA
Docking operation	Truck zone operation
Electric safety device	Electrical protective device
Fixings	Fastenings
Landing door	Hoistway door
Mains	Main power supply
Reeving ratio	Roping ratio
Instantaneous safety gear	Type A safeties (instantaneous safeties)
Progressive safety gear	Type B safeties (progressive safeties)
Pulley	Sheave
Safety gear	Safeties
Well	Hoistway

2.5 Abbreviations

The following abbreviations are used in this report:

FOS = Factor of safety or safety factor.

YP = Yield point.

WP = Working pressure.

UTS = Ultimate tensile strength.

FLP = Full load pressure.

NOTE See also list of abbreviations in item 4.1.2.

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3 Basis for lift safety standards development (basic assumptions)

3.1 Historical background

3.1.1 All lift safety standards assume certain things as being true, without proving them as such, and stipulate safety rules that are based on these assumptions.

3.1.2 No standard, however, clearly spells out the assumptions used. The CEN committee analyzed its standard and summarized in the document CEN/TC10/WG1 N99 (see Annex C) the assumptions that, in the opinion of the committee, were used in the CEN standard.

3.1.3 The CEN assumptions were compared with assumptions implicitly built into other safety standards. It has been indicated that:

a) some assumptions apparently used in the CEN standard were not listed in the document referred to in CEN/TC10/WG1 N99;

b) some assumptions used in other standards differ from those in CEN/TC10/WG1 N99.

3.1.4 Using CEN/TC10/WG1 N99 as a model, the following list of assumptions (see 3.3 through 3.9 in this report) has been developed, which could be used as a basis for future work on safety standards.

The CEN assumptions 5 (related to car speed) and 7 (related to restrictors) as listed in Annex C have not been considered for adoption in this report, since they are deemed to be design parameters.

Further, CEN assumption 2 is adopted in this report as assumption 1 and CEN assumption 6 as assumption 3(c) in order to be consistent with Part 1 of this report.

In summary, CEN assumptions 1, 3, 4, 8, 9, and 10 correspond to assumptions 1, 2, 3, 4, 5, and 6 in this report. Assumption 7 is not covered in the CEN document.

3.2 General

3.2.1 Listed in 3.3 through 3.9 (except as noted) are those things specific to lifts that are assumed as true, although not yet proven or demonstrated as such, including:

a) functioning and reliability of lift components;

b) human behaviour and endurance; and

c) acceptable level of safety and safety margins.

3.2.2 Where the probability of an occurrence is considered highly unlikely, it is considered as not happening.

3.2.3 Where an occurrence proves that an assumption is false, it does not necessarily prove that all other assumptions are false.

3.2.4 The assumptions should be subject to periodic review by standards writing organizations to ensure their continuing validity – considering accident statistics, as well as such things as changes in technologies, public expectations (e.g. product liability), and human behaviour.

3.3 Assumption 1 — safe operation assured to 125 % of rated load

Safe operation of lifts is assured for loads ranging from 0 to 100 % of the rated load. In addition, in the case of *passenger lifts* (see 2.1.2), safe operation is also assured for an overload of 25 %; however, it is not necessary to be able to raise this overload nor to achieve normal operation (rated load performance).

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3.3.1 Rationale for Assumption 1

3.3.1.1 All safety standards limit the car area in relation to its rated capacity (load and/or number of persons) in order to minimize the probability of inadvertent overloading. However, it is recognized that the possibility of an overloading of up to 25 % still exists on *passenger lifts*. To eliminate any hazard for passengers, safe operation must be assured, but not necessarily normal operation.

3.3.1.2 In the case of *freight lifts*, no overloading is anticipated. It is assumed that designated attendants and freight handlers will adhere to instructions posted in cars and will not overload them.

3.3.2 Assumption 1 as applied in current standards

3.3.2.1 Currently CEN does not specifically require a 25 % overload safety margin; however, the design requirements provide for that level of safety.

ASME and CSA requirements 3.16 and 2.16.8 specifically require that safety be assured on *passenger lifts* in the case of 25 % overload.

3.3.2.2 With exceptions given in 3.3.2.5, the ratio of the rated load to the car platform area for passenger lifts is equal (± 5 %) in all standards for the range of 320 to 4 000 kg, and in that respect, universality of the assumption #1 is achieved.

However, the assumed average weight of a passenger differs: 75 kg (CEN) and 72,5 kg (ASME and CSA).

3.3.2.3 Furthermore, the rated load to car platform area ratio is different for *freight lifts*.

CEN (non-commercial vehicle with instructed users)	200 kg/m^2
ASME/CSA (general freight Class A)	244/240 kg/m^2
(motor vehicle Class B)	146/145 kg/m^2
(industrial truck Class C)	244/240 kg/m^2

3.3.2.4 The CEN standard contains two tables showing the ratio between the rated load and the maximum available car area (for *passenger lifts*), see Table 3.

The CEN Table “1.1” corresponding to the requirements for electric lifts is based on the rationale explained in 3.3.1.1 and was taken into consideration when formulating the statement in 3.3.2.2.

3.3.2.5 The CEN Table “1.1 A”, acceptable for Goods passengers lifts, is based on the rationale that where there is a low probability of the car being overloaded with persons, the available area of a hydraulic lift may be increased up to therein specified maximum, provided that additional safety measures are taken to ensure the safe interruption in the lift operation. Such measures include:

a) a pressure switch to prevent a start for a normal journey when the pressure exceeds the full load pressure by more than 20 %;

b) the design of the car, car sling, car-ram connection, suspension means, car safety gear, rupture valve, clamping or pawl device, guide rails, and buffers must be based on a load resulting from CEN Table “1.1”;

c) the design pressure of the jack and the piping shall not be exceeded by more than 1,4.

Starting point for CEN Table “1.1A” was the comparison of safety factors of driving systems on electric traction lifts versus hydraulic lifts. On hydraulic lifts the safety factor for the car suspension means and supporting structure is at least 3 times higher than that of the traction driving systems, when friction between the suspension ropes and the grooves of the drive sheave is taken into account. Consequently, the safety risk of unintended car movement downwards due to the overloading on hydraulic lifts is significantly lower than on electric traction lifts.

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Furthermore, assuming that the car weight is equal to the rated load, in that case an overload of x % on the electric traction lift would correspond to only x/2 % overload for the hydraulic system.

NOTE This is true for machine power only; not for e.g. safety gear operation, guide rails dimensioning, etc.

For car areas up to 5 m^2, the required rated load in CEN Table “1.1 A” for a hydraulic lift may be 1,6 times less than the rated load according to CEN Table 1.1.

NOTE 1.6 is an ISO-standard number R5. This is important in view of the rated loads according to ISO 4190-1 1999, *Lift (US: Elevator) installation — art 1: Class I, II, III and VI lifts,* e.g. a Goods passengers lift with 5 m^2 available car area requires 2 500 kg rated load in the case of an electric lift, and 1 600 kg in the case of a hydraulic lift. For car areas bigger than 5 m^2 there is no mathematical background.

See Table 3 for an abbreviated comparison of the CEN Tables.

Table 3 — CEN Tables

Rated Load	Maximum Car Area		Increase in Car Area "1.1 A" over "1.1"
	CEN Table 1.1	CEN Table 1.1 A for Goods passengers lifts	
kg	m^2	m^2	%
400	1,17	1,68	44
800	2,00	2,96	48
1 200	2,80	4,08	46
1 600	3,56	5,04	42
over 1 600, add	N/A	0,40/100 kg	N/A
2 000	4,20	6,64	58
2 500	5,00	8,84	73
over 2 500, add	0,16/100 kg	0,4/100 kg	250

3.3.2.6 Lift components that are normally designed to withstand, without permanent damage, overloads greater than 25 % (such as ropes, guides, sheaves, buffers, disconnect switches) are not considered in this comparison.

NOTE 3.3.2.6 CEN Assumption 2 (see Annex C) is not a new assumption, but rather one of the methods as to how Assumption 1 is applied in the CEN standard.

3.4 Assumption 2 - failure of electric safety devices

The possibility of a failure of an electric safety device complying with the requirement(s) of a lift safety standard is not taken into consideration.

Since national safety rules for lifts may be based on different assumptions (some are listed below), universality of Assumption 2 may be questioned.

3.4.1 Rationale for Assumption 2

Reliability and safety performance of lift components designated as electric safety devices is assured if designed in accordance with rules contained in a given lift safety standard. However, the design rules may be based on different assumptions.

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3.4.2 Assumption 2 as applied in current standards

Most methods of assuring performance reliability of electric safety devices are similar in present standards. There are, however, differences and inconsistencies, as detailed in section 11.

Section 11.1.3 deals in particular with discrepancies in assumptions implied in requirements for design of electric safety devices.

3.5 Assumption 3 - failure of mechanical devices

a) With the exception of items listed below, a mechanical device built and maintained according to good practice and the requirements of a standard comprising safety rules for lifts is assumed not to deteriorate to the point of creating hazards before the failure is detected.

NOTE National practices and safety rules may be different, such as safety factors. See sections 4.1.3 and 4.2.1 of this report;

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

1) rupture of car suspension means;

2) rupture and slackening of any connecting means such as safety related auxiliary ropes, chains and belts where the safety of normal lift operation or the operation of a safety related standby component is dependent on such connections;

NOTE Since 2000, overspeed valve is required by ASME and CSA when flexible hoses are used and when elevator is located in seismic risk zones 2 or greater;

3) small leakage in the hydraulic system (jack included);

c) the possibility of a car or counterweight striking a buffer at a speed higher than the buffer's rating is not taken into consideration;

d) the possibility of a simultaneous failure of a mechanical device listed above and another mechanical device provided to ensure safe operation of a lift, should the first failure occur, is not taken into consideration.

NOTE 1 The Working Group could not agree upon adopting the CEN Assumption 4.3 (see Annex C) requiring that “the possibility of rupture in the hydraulic system (jack excluded) shall be taken into consideration”;

NOTE 2 Presently, this assumption is implemented only in CEN by requiring a rupture valve or similar devices, while CSA assumes the rupture of flexible hoses only and in that case only, the rupture valve is required. In ASME, the overspeed valve (safety valve) is only required in seismic risk zones 2 or greater.

NOTE 3 The CEN rupture valve protects only in the case of rupture of piping, not the cylinder. The USA's experience indicates that most problems arise from the rupture of cylinders rather than piping;

NOTE 4 Refer to section 10 and table 12 in this Report for detailed comparison of requirements for free fall and excessive speed protection.

3.5.1 Rationale for Assumption 3

3.5.1.1 Although recent accident records do not support the assumption in 3.5 (b)(1), most safety standards (including those studied in the preparation of this report) still assume that the risk of suspension means failure, in particular wire ropes and chains, exists.

3.5.1.2 With the assumption in 3.5 (b)(2) it is recognized that the listed components could deteriorate to the point of creating a direct or potential hazard (by making a safety related standby component inoperative) before the deterioration is detected.

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3.5.2 Assumption 3 as applied in current standards

3.5.2.1 CEN (9.5.1) clearly assumes failure of suspension means, while ASME and CSA requirement 3.17 imply that safety gear must be able tostop, or at least slow down, a free falling car.

3.5.2.2 Standards differ significantly in regard to the rupture or slackening of connecting means. Only CEN seems to be consistent in adopting this assumption. Some standards are inconsistent, e.g. ASME and CSA requirement 2.25.2.3.2 anticipate failure of tapes, chains or ropes operating normal terminal stopping devices, but they do not anticipate failure of an overspeed governor rope. Only CEN (9.10.2.10.3) assumes the possibility of governor rope failure.

3.5.2.3 All standards have adopted the assumption that the possibility of a car or counterweight striking buffers at a speed higher than the buffer's rating is not taken into consideration.

3.5.2.4 All standards have adopted the assumption that the possibility of a simultaneous failure of a mechanical device mentioned in Assumption 3 and another mechanical device provided to ensure safe operation of a lift, should the first failure occur, is not taken into consideration.

3.5.2.5 All standards require an anti-creep system based on assumption 3.5 (b)(3).

3.6 Assumption 4 - imprudent act by users

A user may in certain cases make one imprudent act, intentionally made to circumvent the safety function of a lift component without using special tools. However, it is assumed that:

a) two imprudent acts by users will not take place simultaneously; and

b) an imprudent user's act and the failure of the backup component designed to prevent the safety hazard resulting from such imprudent acts will not take place simultaneously (e.g. a user manipulating an interlock and a safety circuit failure).

3.6.1 Assumption 4 as applied in current standards

All three standards are based on this assumption.

3.7 Assumption 5 - neutralization of safety devices during servicing

If a safety device, inaccessible to users, is deliberately neutralized in the course of servicing work, the safe operation of the lift is no longer assured.

3.7.1 Rationale for Assumption 5

If a mechanic, while servicing a lift, neutralizes or circumvents a safety device (e.g. bypassing door interlocks using a jumper cable or readjusting overspeed governor) safe lift operation cannot be assured.

While it is assumed that lifts will be designed to facilitate ease of servicing work and that service mechanics will be equipped with adequate instructions, tools and expertise to safely service lifts, it is recognized that "fail-safe" service work can never be assured solely by the design of a lift.

3.7.2 Assumption 5 as applied in existing standards

3.7.2.1 All three standards are based on this assumption.

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3.7.2.2 The standards, however, differ in requirements for the “tools” that must be provided by the design of a lift in order to facilitate ease and safety of servicing work. All standards require stop switches on the car roof, in the hoistway pit and pulley room, and also means for inspection operation from the car top. The standards differ in the following:

a) CEN (7.7.3.2) requires “emergency unlocking device” to be provided for every landing door, while ASME (111.9 & 111.10) and CSA (2.12.9 & 2.12.10) require such a device only on two landings and permit it on all other landings.

NOTE ASME and CSA Codes (2.12.6) now require unlocking devices on every landing;

b) only CSA (3.12.1.4) requires “bypass switches” to be provided in the machine room, which would bypass interlocks or car-door-contact, disconnect normal operation and enable car-top-inspection operation, in order to facilitate the mechanic's servicing of faulty interlocks or car-door contacts.

NOTE Now ASME/CSA require bypass switches (see 2.26.1.5 in current Codes);

c) only CEN (5.9) requires lighting of the hoistway.

3.8 Assumption 6 - horizontal forces exerted by a person

One person can exert either of the following horizontal forces at a surface perpendicular to the plane at which the person stands:

a) static force - 300 N;

b) force resulting from impact - 1 000 N.

Static forces of short time duration may be exerted by the simultaneous deliberate acts of several people located immediately adjacent to each other at every 300 mm interval along the width of a surface.

3.8.1 Rationale for Assumption 6

It is assumed that a person leaning against a vertical surface will exert these forces at that surface. It is further assumed that more than one person can exert this force on a surface simultaneously. Only by relating a force to the width of a surface on which it can be exerted, can a realistic design requirement be obtained.

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3.8.2 Assumption 6 as applied in current standards

From Table 4 it is obvious that forces assumed in the standards are different.

Table 4 — Assumption 6 (horizontal forces exerted by a person) as applied in current standards

Assumptions	CEN	ASME and CSA	Part 1 SA	Part 3	Japan
1.0 **Static force**					
1.1 Landing doors	300 N (7.2.3)	2 500 N [2.11.11.5.7]	300 N (7.2.3)	1.2 kN (12.2 & AS 1735.2 12.4.1	No spec
1.2 Car enclosure	300 N (8.3.2.1)	330 N [2.14.1.3]	300 N (8.3.2.1)	330 N (23.1 & As1735.2 23.18)	No spec
2.0 **Impact on:**					
2.1 Landing doors	Pendulum shock test when glass is used (7.2.3.3) See Note	5 000 N [2.11.11.8]	Pendulum shock test when glass is used (7.2.3.3) See Note	No spec.	No spec
2.2 Car enclosure	Pendulum shock test when glass is used (8.3.2.2).	No spec.	Pendulum shock test when glass is used (8.3.2.2).	No spec.	No spec.
3.0 **Force distribution**	Evenly distributed over an area of 5 cm^2 (for the 300 N) at any place, and at both sides (for doors) See Note	100 mm x 100 mm [2.11.11.5.7] and 300 mm x 300 mm [2.11.11.8]	Evenly distributed over an area of 5 cm^2 (for the 300 N) See Note	Door - 0.1m^2 Car wall – 50 mm x 50mm	No spec
NOTE The pendulum shock tests – Hard and Soft – required are described in Annex J of EN81-2: 1998.					

3.9 Assumption 7 – retardation

A person is capable of withstanding an average vertical retardation of 1 g (9,81 m/s^2) and higher transient retardations.

3.9.1 Rationale for assumption 7

The retardation which can be withstood without injury varies from person to person. Historically, the values used in the standards (see table 5) have not been shown to be unsafe for a vast majority of people.

NOTE See 3.9.3 regarding retardation limits on *emergency car stops.*

3.9.2 Assumption 7 as applied in current standards

Table 5 gives a comparison of requirements based on the assumed safe retardation rates. Major differences are noted in relation to rupture valves, plunger stops, and emergency speed limits.

No standard limits retardation in the case of car stops initiated by an electrical safety device.

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Table 5 — Assumption 7 (retardation) as applied in current standards

Assumption	CEN	ASME and CSA	Part 1	SA* Part 3	Japan
Maximum Average Retardation*					
Progressive Safety Gear	1 g (9.8.4) [in free fall and full load]	1 g [2.17.2.2.2] [with counterweight attached]	1 g (9.8.4)	1g (33.1)	1g in vertical direction and 0.5g in horizontal direction (BSL-EO Art 129-10 item 2 parag.1)
Progressive Clamping Device	1 g, when full load (9.9.4)	1 g @ rated load. 2 g peak > 0.04 s [3.17.3.5]	1 g (9.9.4)	N/A	N/A
Oil Buffers	1 g (10.4.3.2) Full Load; Type-Examination	1g [3.22] and [2.22.4.1.1]	1 g (10.4.3.2)	1 g (9.1.5.2)	1g in vertical direction and 0.5g in horizontal direction(BSL-EO Art 129-10 item 2 parag.1)
Rupture valve	1 g (12.5.5.1)	1 g [3.19.4.7.5(b)] 2.5 g peak > 0.04 s [3.19.4.7.5(c)]	1 g (12.5.5.1)	No spec.	N/A
Plunger stops	1 g (12.2.3.3.2)	Rules require ETSR cut speed to 0.25 m/s	1 g (12.2.3.3.2)	1 g (7.2.6)	No spec.
Emergency speed limit	No spec.	1 g [3.25.2.2.2]	No spec.	No spec	No spec.
Emergency car stops	No spec.	1 g for certain EPD stops [3.26.4.2]	No spec.	No spec	No spec.
Maximum retardation					
Safety gear	No spec.	No spec	No spec.	No spec	No spec.
Buffers (*if t = Duration*)	> 2,5 g (10.4.3.2) t =< 0,04 s	> 2,5 g [2.22.4.2] t =< 0,04 s	> 2,5 g (10.4.3.2) t =< 0,04 s	>2,5 g (9.1.5.3) t =< 0,04 s	> 2,5 g t = < 0.04 s (JEAS-517)

* NOTE: Maximum average retardation levels exceeding 1 g can occur with a lightly loaded lift during safety or buffer application.

** NOTE: SA data apply to indirect lifts only.

NOTE 1 g = 9,81 m/s^2.

3.9.3 Agreed-upon points

All Standards should consider retardation limits on emergency stops initiated by an electrical safety device, albeit based on bio-mechanical studies.

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4 Approach to design safety for hydraulic components

4.1 Historical Background

4.1.1 Philosophical differences

This section concentrates on differences between the CEN and ASME requirements for the design of hydraulic components. Reference to the CSA standard is made where it differs from ASME.

NOTE The second sentence is not valid any more because since 1996 there are no difference between ASME and CSA design requirements for hydraulic components any more.

a) Differences in both design philosophy and design formulae lead to different cylinders and rams, valves, pipes, and fittings when designed to CEN and ASME standards. Philosophical differences are as follows:

   1) ASME uses the ultimate tensile strength subject to a minimum percentage elongation of the material as a design criterion;

   2) CEN uses the 0,2 % proof stress yield point as the design criterion. Percentage elongation is not considered;

   3) the working pressure is differently defined in ASME and CEN;

   4) the factors of safety used are also different;

b) the differences are demonstrated by examples as illustrated by the following comparisons:

   1) thickness of cylinder walls of single stage jacks (4.1.4);

   2) thickness of flat cylinder base/head (4.1.5);

   3) thickness of semi-elliptical cylinder head/cambered base (4.1.6);

   4) thickness of ram wall for buckling (4.1.7).

4.1.2 Nomenclature

The following nomenclature is used in the two different standards:

Item	Units*	CEN	ASME
Working pressure	kPa	—	p
Full load pressure	MPa	p	—
Inside diameter of cylinder	mm	D_i	d
Diameter of flat head	mm	—	d
Inside diameter of skirt	mm	D_i	D
Outside dia. of cylinder, pipe	mm	D	D
Wall thickness, cylinder	mm	e_{cy1}	t
Wall thickness, flat bottom	mm	e_1	t
Wall thickness, semi-elliptical	mm	e_2	t
Additional wall thickness	mm	e_0	C
Design or allowable stress	kPa	—	S

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Item	Units*	CEN	ASME
0.2 % proof stress	N/mm^2/kPa	$Rp_{0.2}$	Y.P.
Tensile strength	N/mm^2	R_m	—
Modulus of elasticity	N/mm^2	E	—
Cross-sectional area of plunger	mm^2/m^2	An	A
Slenderness ratio	(dimensionless)	λ	—
Maximum unsupported ram length	mm	1	L
Radius of gyration	mm	—	R
Acceleration of gravity	m/s^2	g_n	—
Reeving (roping) ratio	(dimensionless)	Cm	—
Mass of empty car	kg	P_3	—
Rated load in car	kg	Q	—
Mass of ram	kg	Pr	—
Mass of ram head equipment	kg	Prh	—
Design load on ram	N	F_5	—
Actual load on ram	N	F	—
Second moment of ram area	Mm^4	J_n	—

* If two entries, then the first applies to CEN, the second to ASME.

4.1.3 Factor of Safety Comparison

ASME and CSA clause 8.2.8.5.1 (editions since 2000) requires:

1) for tensile, compressive bending and torsional loading, the plunger, cylinder and connecting couplings shall have a factor of safety not less than 5 based on ultimate tensile strength (UTS);

2) for pressure calculations of the components that are subject to fluid pressure, including the plunger, connecting coupling, control valves, cylinder, and rigid piping shall have a factor of safety (FOS) not less than that calculated from:

$$F = \frac{5{,}04}{E - 2{,}8} + 2{,}7 \qquad \text{(A)}$$

where

F = Minimum FOS based on 0,2 % proof stress yield point. The minimum allowable F shall be 3;

E = Percentage Elongation in 50 mm gauge length as per ASTM Standard E8, expressed as a whole number (eg, 20 % = 20 and 5 % = 5). The minimum allowable E shall be 5.

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The allowable stress to be used for pressure calculations, according to ASME and CSA 8.2.8.5.2 (2000 and later editions), shall be determined as follows:

$$S = \frac{Y.P}{F} \qquad \text{(B)}$$

where

S = Allowable stress (kPa);

Y.P = Yield point based on 0,2 % proof yield stress;

F = FOS per formula (A).

CEN (12.2.1.1.1) requires that rams and cylinders be designed with a FOS of 3,91 (2,3 x 1,7), based on the 0,2 % proof stress (YP) and the *full load pressure* (FLP).

For calculations of tensile, compressive, bending and torsional loads the following relationship between ASME and CEN requirements can be established:

$R_{p0.2}$ = 0,2 % proof stress

ASME WP = 1,15 (CEN FLP)

ASME FOS = 5 (ASME Working Stress)

CEN FOS = 3,91 (CEN working stress at FLP) or

= 3,4 (ASME working stress at WP)

Therefore:

UTS $\geqslant$ 5 ASME working stress

YP $\geqslant$ 3,4 ASME working stress

(0,2 % proof stress = YP)

Nominal equality of ASME and CEN requirements would occur if:

$$\frac{UTS}{YP} = \frac{5}{3,4} = 1,47$$

However, the formulae employed are different in the two codes, so the comparison is more complex.

For comparisons of stresses due to pressure it is necessary to determine the FOS from the formula (A) and the allowable stress from formula (B).

Examples of the differences between CEN and ASME/CSA are presented in the following sections 4.1.4 through 4.1.7.

NOTE 4.1.3 For further observations and suggestions regarding the factor of safety, refer to Section 4.2.1 and 4.2.4.

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4.1.4 Cylinder wall thickness of single stage jacks

According to ASME and CSA (2000 and later edition, rule 8.2.8.2, the cylinder wall thickness of a single stage jack is calculated with the following formula:

$$t = \frac{pd}{2S} \tag{1}$$

where

d inside diameter;

p working pressure;

S working (or allowable) stress;

t minimum wail thickness.

From CEN Annex K:

$$e_{cyl} \geq \frac{2{,}3 \times 1{,}7}{R_{p0.2}} p \cdot \frac{D}{2} + e_0 \tag{2}$$

where

e_{cyl} wall thickness;

p full load pressure;

D outside diameter;

e_0 1,0 mm.

It was noted that e_0, may be 0,5 in some cases;

however it was agreed to leave it at 1,0 for the sake of simplicity.

For a valid comparison, the two formulae should be written as close, as possible in the same form, using common parameters.

As the full load pressure (p) in Equation (2) is in fact the static pressure of the system (P_s), and, based on Section 2.3 in this report, the working pressure (p) in Equation (1) may be written as p = 1,15 (P_s), consequently Equation (1) may be rewritten as follows:

$$t = \frac{1{,}15}{2} \cdot \frac{P_s \times d}{S} \tag{1a}$$

$$t = 0{,}575 \cdot \frac{P_s \times d}{S}$$

Recognizing that $D = d + 2_{cyl}$, Equation (2) may be rearranged in terms of the inside diameter as follows:

$$e_{cyl} \geqslant \frac{1{,}96 P_s \times d + e_0 R_{p0.2}}{R_{p0.2} - 3{,}91 P_s} \tag{2a}$$

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In order to establish difference in the cylinder wall thickness when calculated per ASME versus CEN formula, the following is assumed:

a) the cylinder is made of material having

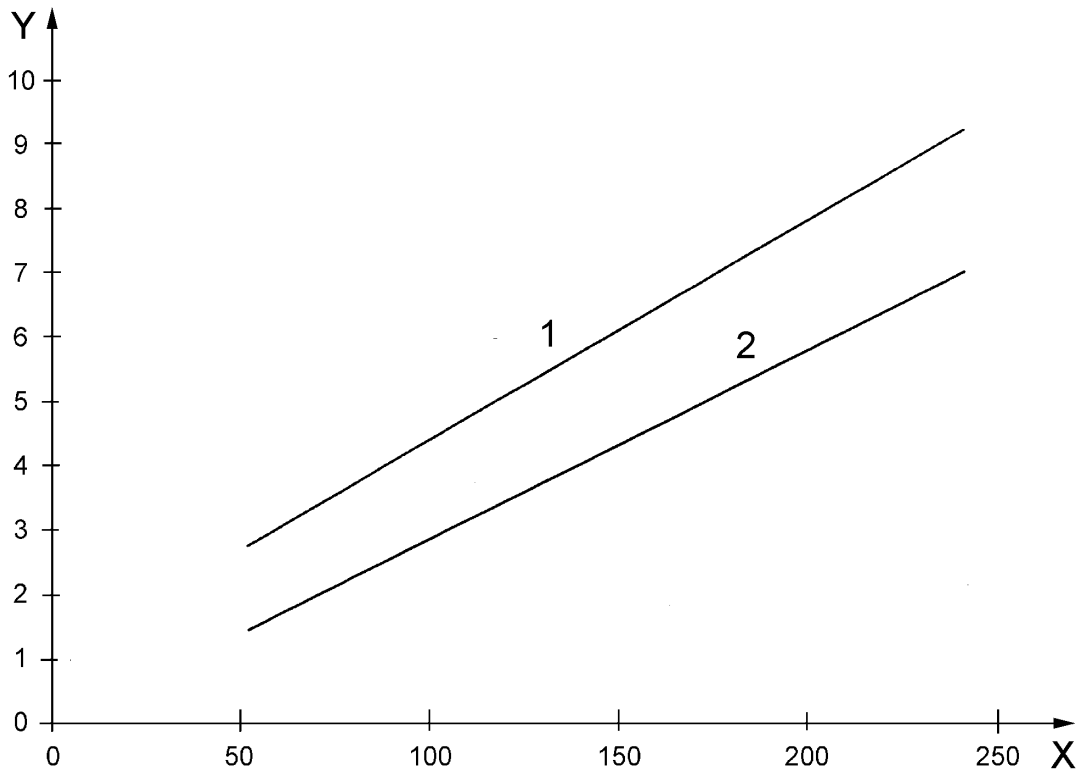
Y.P. = $R_{p0.2}$ = 187,5 Mpa; and

E= 14,8

b) the static pressure of the system is P_s = 3 Mpa From formula (A) F = 3,125, and from formula (B) S = 60 MPa.

The wall thickness is calculated in formulae (1a) and (2a) and plotted against cylinder inner diameter in Figure 1.

The graphs show that for a practical range of cylinder diameters the wall thickness required by CEN is always greater than that by ASME. and CSA.



Key

X cylinder inner diameter (mm) - Working pressure 3 MPa, Yield point = 187,5 MPa

Y minimum wall thickness (mm)

1 CEN

2 ASME

Figure 1 — Variation of Required Wall Thickness with Cylinder Diameter

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4.1.5 Thickness of flat cylinder base/head

ASME and CSA requirement 8.2.8.3(a) requires that the wall thickness of a flat unreinforced head be designed according to the formula:

$$t = d\sqrt{\frac{p}{4S}} \tag{3}$$

For p = 1,15 P_s

$$t = 0{,}536d\sqrt{\frac{p_s}{S}} \tag{3a}$$

CEN Annex K, for flat base with relieving groove, gives the following formula for the cylinder base thickness:

$$e_1 \geq 0{,}4D_i\sqrt{\frac{2{,}3 \times 1{,}7\ p}{R_{p0.2}}} + e_0 \tag{4}$$

or, if simplified

$$e_1 \geq 0{,}791D_i\sqrt{\frac{Ps}{R_{p0.2}}} + e_0 \tag{4a}$$

Where

D_i = inner diameter of the cylinder;

e_1 = cylinder base thickness.

Using the same material and assuming the same static pressure as in section 4.1.4, Equations (3a) and (4a) may be plotted on a common axis, as shown in Figure 2.

It is evident that for cylinder inner diameters greater than about 60 mm, the ASME requires a greater wall thickness than the CEN.

This result is perhaps reasonable in that the stress relieving groove would help to reduce stress concentrations in the CEN case.

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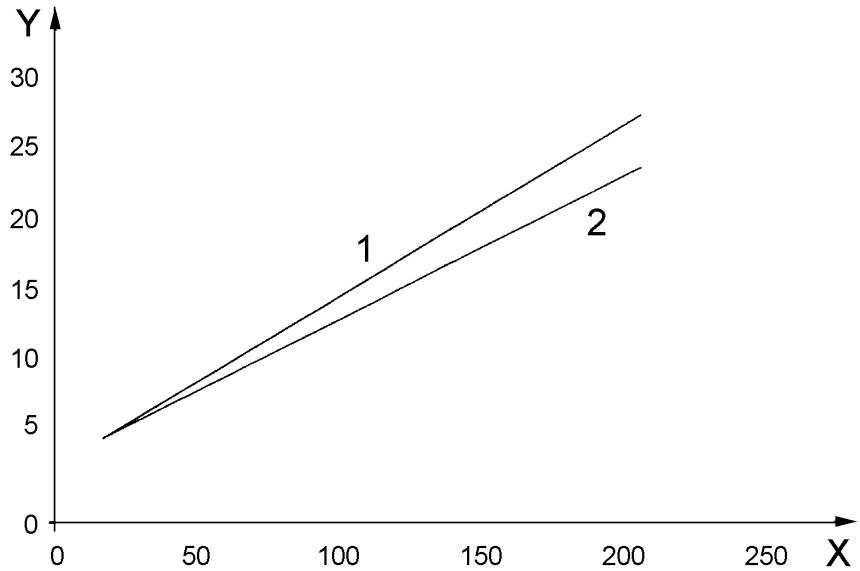

Key

X cylinder inner diameter (mm) - Working pressure 3 MPa, Yield point = 187,5 MPa

Y minimum wall thickness (mm)

1 ASME

2 CEN

Figure 2 — Variation of Required Thickness of Flat Cylinder Base/Head with Cylinder Inside Diameter

4.1.6 Thickness of Semi-Ellipsoidal Cylinder Head/Cambered Base

The case of dished seamless ellipsoidal heads is dealt with in ASME and CSA requirement 8.2.8.3(c).

CEN Annex K, covers the *case* of cambered bases. The cambered shape approximates an ellipsoidal form.

The ASME and CSA requirement for the wall thickness is given by the following formula:

$$t = \frac{5pD}{6S} \tag{5}$$

where D = inside diameter of the skirt.

For p = 1,15 P_s

$$t = 0{,}958\frac{P_s D}{S} \tag{5a}$$

The wall thickness in CEN is given by the following formula:

$$e_2 \geqslant \frac{2{,}3 \times 1{,}7p}{R_{p0.2}}\left(\frac{D}{2} - e_0\right) \tag{6}$$

where D = outside diameter.

Substituting (D_i + 2e_2) for D and simplifying the following relationship results in:

$$e_2 \geqslant \frac{1{,}955pD_i + R_{p0,2}}{R_{p0,2} - 3{,}91p} \tag{6a}$$

where D_i = inside diameter.

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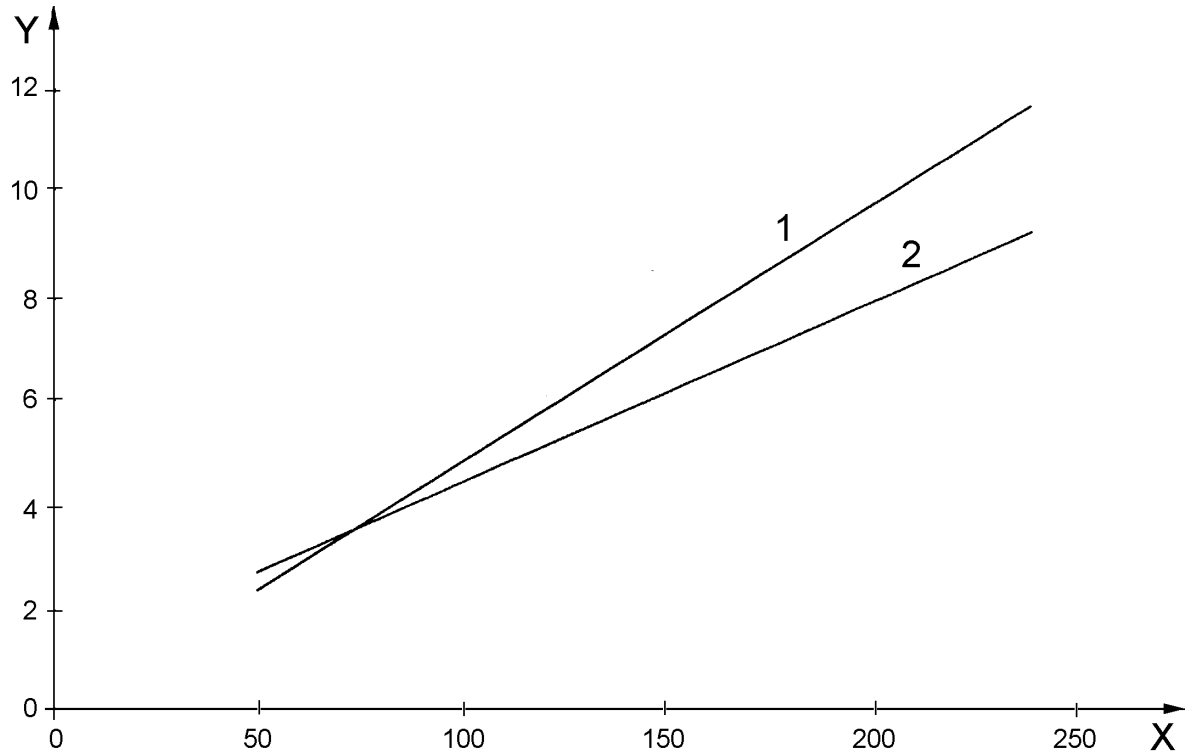
Using the same material and assuming the same static pressure as in Section 4.1.4, the head wall thickness, calculated with formulae (5a) and (6a), may be plotted as shown in Figure 3.

For values of the inside diameter in excess of 75 mm, the wall thickness required by ASME exceeds that required by CEN. This result may in part be due to geometric differences between the two configurations, but is consistent with the findings in section 4.1.5.



Key

X cylinder inner diameter (mm) - Working pressure 3 MPa, Yield point = 187,5 MPa

Y bottom thickness (mm)

1 ASME

2 CEN

Figure 3 — Variation of Elliptical Bottom Thickness with Cylinder Inside Diameter

4.1.7 Thickness of Ram Wall for Buckling

Both the CEN and the ASME standards recognize the elastic stability of rams as a limitation on elevator load. Guidelines are provided to determine the safe strokes for given loading, geometry, and end conditions. The approach to calculation in both standards is essentially on the lines of the conventional Euler analysis, with a factor of safety built into the formulae for long slender rams. For shorter, stockier rams, the limitation of compressive strength of the ram is recognized in the formulae.

The design of rams not subject to eccentric loading is covered by ASME and CSA requirement 8.2.8.1.1.

The relationships between the total load (W), the ram cross sectional area (A), the ram unsupported length (L), and the radius of gyration (R) are expressed as two separate equations, one for a slenderness ratio of L/R < 120, the other for L/R > 120.

For L/R < 120 the following formula applies:

$$\frac{W}{A} = 9{,}773 \times 10^{7} - 3{,}344 \times 10^{2} \left(L/R\right)^{2} \qquad (7)$$

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For L/R > 120, this relationship becomes:

$$\frac{W}{A} = \frac{6{,}552 \times 10^{11}}{(L/R)^2} \tag{8}$$

CEN –Annex K, gives the following formulae for the calculation of rams against buckling (for nomenclature, refer to 4.1.2):

For $\lambda n < 100$:

$$F_5 \leqslant \frac{A_n}{2}\left[R_m - (R_m - 210)\cdot\left[\frac{\lambda n}{100}\right]^2\right] \tag{9}$$

where

$$F_5 = 1{,}4 g_n [C_m (P_3 + Q) + 0{,}64\, P_r + P_{rh}] \tag{10}$$

i.e.: $F_5 = 1{,}4\ F$ (10a)

where

$$F > g_n [C_m(P_3+Q)+0{,}64P_r+P_{rh}]$$

Therefore from (9) + (10a):

$$\frac{F}{A_n} \leqslant \frac{1}{2{,}8}\left[R_m - (R_m - 210)\cdot\left[\frac{\lambda n}{100}\right]^2\right] \tag{9a}$$

For $\lambda n \geq 100$:

$$F_5 = \frac{\pi^2 \times E \times J_n}{2 \times L^2} \tag{11}$$

Therefore

$$\frac{F}{A_n} \leqslant \frac{\pi^2 \times E \times J_n}{2{,}8 \times L^2 \times A_n} \tag{11a}$$

Equation (11a) may be further simplified to resemble (8):

$$\frac{F}{A_n} = \frac{7{,}28 \times 10^{11}}{(L/R)^2} \tag{11b}$$

NOTE F/A_n is the equivalent of W/A, or $F = W$ and $A = A_n$.

For purposes of illustration, the combination of Equation (7) and (8) is compared to the combination of Equations (9a) and (11b), for a steel having ultimate tensile strength of R_m = 331 MPa.

This direct comparison of CEN and ASME is shown on Figure 4.

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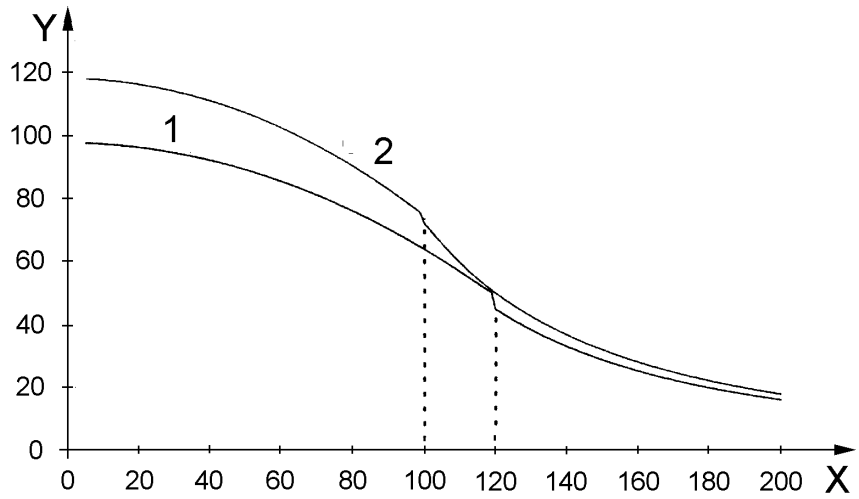
It is noteworthy that both the CEN and the ASME plots show smooth curves of similar shape. The L/R ratio of 120 for ASME and 100 for CEN defines the point of inflection in each curve. For higher values of L/R, the ASME and CEN curves are very close, the ASME curve giving more conservative values of load by about 11 %. The differential between the two is independent of material strength. For lower values of L/R (i.e. L/R < 100), the differential between the ASME and CEN increases as the L/R ratio decreases.

This reflects the particular value chosen for the material strength. Thus, a lower material strength would bring the curves closer together. The ASME curve assumes a fixed material strength; hence, the constant coefficients in Equation (7).



Key

1 ASME
2 CEN
X L/R UTS = 331 MPa
Y W/A (MPa)
W total load
L free length of ram
A cross sectional area of ram
R radius of gyration of ram

Figure 4 — Comparison of Buckling Criteria

4.1.8 Buckling of Ram - Special Cases

4.1.8.1 Plungers with varying cross sectional area, according to ASME and CSA requirement 8.2.8.1.1(d), must be designed with factors of safety of not less than 3 based on accepted elastic stability analysis. CEN does not specifically address this issue.

4.1.8.2 ASME and CSA requirement 8.2.8.1.2 deals with the specific situation of rams subject to bending as a result of eccentric loading in addition to buckling loads. This situation is not covered by CEN, as clause 12.2.2.1 requires a flexible connection between the car and the ram. (See also 5.1.1 and 5.2.1 of this report.)

4.1.9 Wall thickness of rigid pipes

According to ASME and CSA requirement 8.2.8.4, the minimum wall thickness of rigid pipes is calculated on the basis of allowable stress (see formula (B)), using same formula (1a) as for cylinders, except that the following wall thickness must be added on:

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C = 0 mm, for plain pipes (unreduced wall).

C = 1,25 mm, for threaded pipes up to 9,5 mm inside diameter.

C = actual depth of thread, for threaded pipes in excess of 9,5 mm inside diameter.

C = actual depth of groove, for grooved pipes.

CEN (Annex K) bases calculations of rigid-pipe-wall thickness on proof stress, using the same formula (2a) as for cylinders, in which the added on wail thickness is:

e_o = 1 mm, for pipes between the cylinder and the rupture valve;

e_o = 0,5 mm, for other pipes.

4.1.10 Strength of flexible hoses

Burst strength of flexible hoses is established as

Follows:

ASME and CSA

3.19.3.3.1: 10 × working pressure (WP)

CEN (12.3.3.1): 8 × full load pressure (FLP)

Since WP = 1,15 FLP (see 2.3 in this report), the required factor of safety (FOS), relating to the full load pressure and burst pressure, is 11,5 in ASME and 8,0 in CEN

4.1.11 Capacity of pressure relief valve

The pressure relief valve has to be adjusted to limit the pressure to the following:

ASME and CSA

3.19.4.2.1: 150 % WP

CEN (12.5.3): 140 % FLP, or

170 % FLP in the systems with high internal losses

Considering the relationship WP/ILP established in Section 2.3, one *can* find that ASME requires higher capacity of the relief valve, or:

$$\frac{\text{ASME}}{\text{CEN}} = \frac{150\,\% \cdot 1{,}15}{140\,\%} = 123\,\%$$

However, for the systems with “high internal losses” the requirements in both standards are approximately the same, or:

$$\frac{\text{ASME}}{\text{CEN}} = \frac{150\,\% \cdot 1{,}15}{170\,\%} = 101\,\%$$

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4.2 Observations and suggestions by individual experts

4.2.1 Observations: Factor of Safety and ductility (reference 4.1.3)

4.2.1.1 High ductility materials preferred

One expert suggested that all safety standards should provide an incentive for the use of high ductility materials for hydraulic components in order to reduce the probability of failures. For that reason, all standards should link the factor of safety to the percentage elongation of material, as is the case in ASME and CSA requirement 8.2.8.5.

4.2.1.2 Rationale behind the suggestion in 4.2.1.1

Expressing the failure criteria in safety standards in terms of the proof stress (or yield point) only could tempt the designer to use materials of high strength and low ductility, as these are low cost. However, low ductility materials have a greater scatter in their mechanical strength, and if overloaded could fail without significant permanent deformation.

It is important to have yielding of the material on hydraulic systems so that leakage will precede catastrophic failure and provide visual indication of an impending problem.

The use of high strength materials without consideration of the ductility of the material could mislead the designer and could result in an unsafe design. This has relevance to the CEN Code.

4.2.1.3 Background to the suggestion 4.2.1.1

a) Introduction

It is customary when performing an engineering analysis of a loaded device that the stresses be determined, a failure criteria selected, the material strength determined and factor of safety applied. In the case of ASME, CEN, and CSA standards, differences can occur in all of these parameters; hence very different designs and margins of safety are likely. This study focuses on the failure criteria and the factor of safety to be applied according to the requirements in current standards.

b) Failure criteria - CEN approach

CEN requires the use of the 0,2 % proof stress yield point as the failure criterion. There is no consideration for the ductility of the material, i.e. its deformation beyond the yield point. In general, ductile materials have a greater difference between the yield point and the rupture point than brittle materials.

This is illustrated in Fig 5, which shows the variation of yield point against percentage elongation for materials which may be hardened to different strength values. The figure shows the typical reduction of percentage elongation or permanent deformation before rupture as the yield point is raised. The yield point effectively approaches the ultimate tensile strength until the material ruptures almost immediately after it is overloaded.

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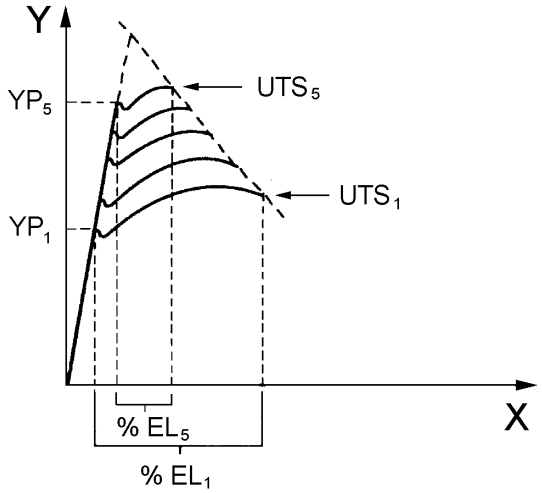


Key

YP yield strength

UTS ultimate tensile strength

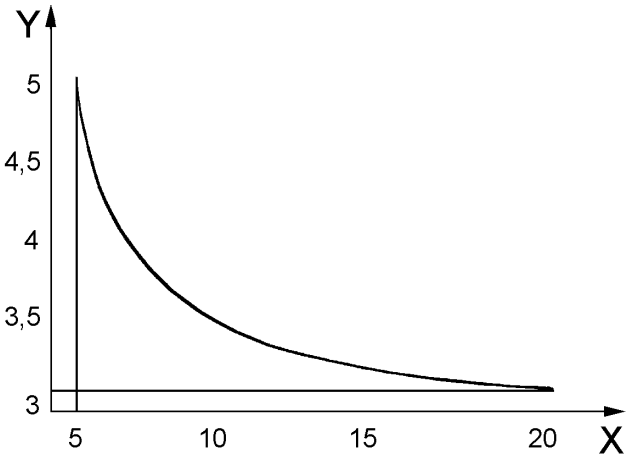
EL elongation

Figure 5 — Effect of increased Strength on Elongation

c) Failure criteria - CSA and ASME approach

ASME and CSA

requirement 3.18.2.1make the factor of safety (F) a function of the percentage elongation as shown in formula (A) in Section 4.1.3 of this report The relationship is illustrated in Fig 6, which shows that, the greater the material ductility, the smaller the factor of safety is permitted.



Key

X elongation percent (E)

Y safety factor on yield point (F)

F 5,04/(E-2,8) + 2,7 valid for $E \geqslant 5$ and $F \geqslant 3$

Figure 6 — Relationship Between Factor of Safety and Percentage Elongation

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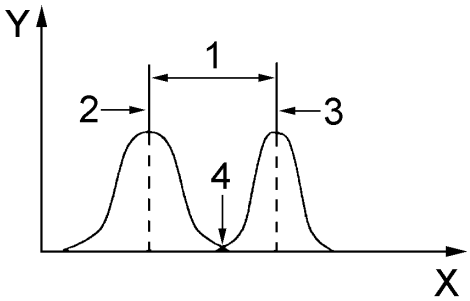
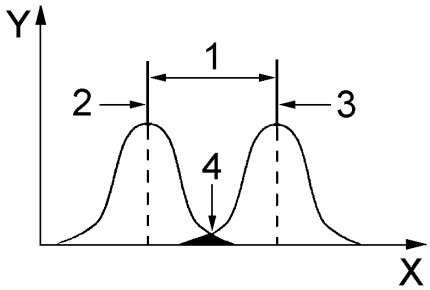
d) Probability of failure

The purpose of applying a factor of safety (FOS) is to reduce the risk of failure to an acceptably low level. Risk may be quantified as the probability of failure under load. Fig. 7 illustrates the probability of failure of two materials, each possessing the same nominal yield point and subject to the same nominal loading. Stress is plotted on the horizontal axis and probability of occurrence on the vertical axis. The shape of the loading distribution reflects the uncertainty in the applied load, while the shape of the material strength distribution reflects the variability from piece to piece in the material.

The nominal factor of safety is the ratio of the mean strength to the mean stress in each combination. The shaded area, i.e. the overlap of the distributions, represents the cases where load stress exceeds material strength and failure will occur. It is evident that the greater the FOS, the smaller the overlap area will be. However, a finite probability of failure always exists.



a Ductile material



b Brittle material

Key

X probability
Y stress
1 margin of safety
2 mean stress
3 mean strength
4 small overlap in case of a) ductile material
large overlap in case of b) brittle material

Figure 7 — Probability of Failure

e) Advantages of Ductile Materials

The ductility of the material affects the shape of the material strength distribution curve. Thus less ductile materials have greater scatter of their yield points and ultimate tensile strengths. Thus the probability of failure is higher with brittle materials than ductile materials with the same mean strength and factor of safety. Moreover, in the case of hydraulic elevators, the use of ductile materials (which allow a substantial degree of deformation before rupture) is particularly desirable. The deformation, in the event of overload, would allow joints to leak with the resultant dissipation of the pressure, rather than a sudden rupture.

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4.2.1.4 Example for FOS: Cylinder Wall Thickness

Calculations were made for four various cylinder materials used in Finland, Russia and U.S.A. (see table 6) in order to illustrate differences in the resulting design of a cylinder, when calculations are based on CEN, ASME or CSA requirements for:

f) cylinder wall thickness, per Section 4.1.4, formulae (1a) and (2); and

g) factor of safety, per section 4.1.3 in this report.

The following conditions were assumed:

— working pressure 3,5 MPa

— inside diameter (ASME & CSA formulae) 125 mm

— outside diameter (CEN formula) 135 mm

As illustrated in table 6, the resulting wall thickness, when calculated in accordance with CEN rules, is 72 to 90% greater than when calculated per CSA or ASME rules.

Table 6 — Illustrative examples of wall thickness for 4 materials

Data for Materials					**Wall Thickness** (mm)				
Source Country	Material	UTS MPa	0,2 % PS MPa	Elong. %	CEN	ASME and CSA	SA		Japan
U.S.A.	ASTM								
	A53-88a	330	205	> 20	5,5	3,7	5,5		No spec
	Grade A								
U.S.A.	ASTM								
	A53-88a	415	240	> 20	4,8	3,1	4,8		No spec
	Grade B								
Finland	DIN 17100	510	353	22	3,6	2,2	3,6		No spec
Russia	35 GOST 1050	520	310	20	4,0	2,4	4,0		No spec

4.2.1.5 Comments on FOS and suggestions in 4.2.1.1

Other experts commented on the suggestions in 4.2.1.1 and the background information in 4.2.1.3 regarding the safety factors and ductility of materials:

a) it was noted that differences between ASME and CEN design approaches are not limited to hydraulic equipment but to all mechanical components in general. With respect to the CEN's not referring to ductility of materials, it was stated that a designer would not be permitted to use any material but only those that have been subjected to tests in accordance with non-elevator rules or Standards.

Another expert stated that more consideration should be given to hydraulic equipment because of the risk of leakage, It was recognized that "codes of practice" (such as DIN) exist in Europe, where appropriate materials, of certain ductility, are suggested, but the expert maintained that a better approach is to interrelate the ductility and strength. This would become a universally acceptable method if introduced in all elevator safety standards.

b) One expert who agreed that failure of ductile materials would result in leakage rather than catastrophic failure (car drop) expressed concern for damage to the environment by the leaking oil.

Another expert suggested that the failure of brittle materials would result in both the unsafe condition and damage to the environment.

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c) It was suggested that CEN does not include *all* rules but rather assumes that all general basic design rules, not specific to lifts, are known to and followed by each elevator design engineer.

Another expert stated that basic general engineering practices may prove insufficient for a particular elevator component and for that reason safety factors have been introduced in ASME and CSA, based on experience.

Another expert reiterated that a fixed safety factor is not a fully satisfactory solution. A better approach is one based on the probability of failure, as in CSA and ASME.

4.2.1.6 Should Factor of Safety be Specified?

a) Several experts suggested that all standards, including CEN, should contain specific rules regarding factors of safety;

b) another expert indicated that CEN does not normally specify factors of safety except for equipment which is specific to the lift industry (e.g., hoisting ropes). The necessary information can be found in the standards relating to structural steel or mechanical construction, such as the standard for the bolts industry;

c) the following ranges of factors of safety are commonly used:

   1) for structures, from 4 if the material has an elongation of 20 % or more to 10 if the material has an elongation of less than 10 %;

   2) for machinery, from 8 if the material has an elongation of 20 % or more to 12 if the material has an elongation of less than 10 %.

4.2.2 Observations: Cylinder Wall Thickness (reference 4.1.4)

4.2.2.1 Background to CEN formula

Regarding the CEN formula (2) in Section 4.1.4, an expert representing CEN explained

a) Factor 2,3 is the product of 1,15 (friction losses) and 2 (pressure peak). The pressure peak factor should take into account the acceleration peaks and stresses created by activation of a rupture valve or when the plunger is stopped at its ring;

b) the denominator $R_{p0.2}/1,7$ corresponds to UTS/5 in ASME, but as they are different, the calculated wall thicknesses are different;

c) "added on" wall thickness e_o, should compensate for tolerances of the material and also the affects of corrosion on buried cylinders;

d) formula (2) was adopted from rules for pressure vessels, where pressure in all directions is present.

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4.2.2.2 Differences in thickness affected by e_o

a) To analyze differences between the ASME and CEN formulae for cylinder wall thickness (see formulae (1a) and (2a) in section 4,1.4), the safe static pressure (P_s), that a cylinder can withstand, was calculated:

For ASME:

$$P_s = \frac{tS}{0{,}575d} \qquad 1(b)$$

For CEN:

If e_{cy1} = t then formula (2a) can be written as follows:

$$P_s \leq \frac{(t - e_o)R_{p0.2}}{1{,}955\ (d + 2t)} \qquad 2(b)$$

According to section 4.1.3 of this report, if the percentage elongation is E = 10, then from formula (A) factor of safety is F= 3,4.

The allowable stress, from formula (B) is:

$$S = \frac{Y.P.}{F} = \frac{Y.P.}{3{,}4} = \frac{R_{p0.2}}{3{,}4}$$

Thus $R_{p0.2}$ = 3.4S. Consequently, formula (2b) may be written as follows:

$$P_s \leq \frac{(t - e_s)S}{0{,}575(d + 2t)} \qquad (2c)$$

b) comparison of formulae (1b) and (2c) shows that the formulas are similar, except that CEN formula has "added on" wall thickness of $e_{,o}$ = 1 mm and is based on the “outside” (d + 2t) instead of the "inside" diameter (d);

c) because of these differences, the static pressure would differ as follows:

For S = 60 Mpa

D = 100 mm

T = 5 mm

e_o = 1 mm

Resulting static pressure is:

Per ASME: P_s = 5,2 Mpa

Per CEN: P_s = 3,8 Mpa

d) Both formulae are based on the thin-walled approximation of Lame's Theorem, which assumes uniform distribution of radial and tangential stresses, provided that the wall thickness is small compared to the cylinder diameter. ASME formula (1b) is identical to the thin-walled cylinder rule.

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4.2.2.3 Tests to verify factors used in formulae

To establish oil pressure and decelerations under various operating conditions, tests were carried out on the following elevator:

Capacity : 8 persons

Speed : 0,45 m/s

Type : indirect acting

Floors : 5

Test Results:

Operating Condition	Pressure (MPa)	
	Up	Down
No load:		
Rupture valve activated	—	4,5
Full load:		
Static	3,8	3,8
Acceleration	4,7	3,5
Rated speed	4,5	3,6
Slowdown	4,3	3,9
Emergency stop	5,0	5,0
Rupture Valve	—	5,7

Full load on rupture valve	Declaration (m/s^2)
Ram declaration	1,6
Car declaration	3,2

Analysis:

The pressure peak of 5,7 MPa is measured when the full load is stopped by the activation of the rupture valve.

For the same operating condition on this elevator, calculated pressure, using CEN formula is 7,5 MPa.

When compared with the measured static full load pressure of 3,8 MPa, then the pressure peak factor is:

Measured peak/static pressure = 5,7/3,8 = 1,5

Calculated peak/static pressure = 7,5/3,8 = 1,96

The tests indicate that the highest pressure peak factor on this elevator is 1,5. The CEN formulae, however, incorporate a pressure peak factor of 2 (see 4.2.2.1a) to provide for the worst cases.

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