WORLDWIDE STANDARDS FOR GUIDE-RAIL CALCULATIONS

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Abstract:
This paper is a follow up on a preceding paper called "Analysis of Stress in Guide-Rails" and given at ELEVCON 90 in Rome in March 1990. The paper is split into two parts. In the first part the state of the art in the calculation of stress and deflection of guide-rails in compliance with the European Standard EN 81.1, American Safety Code A 17.1, Australian Standard AS 1735.2 and the Czechoslovakian Standard is presented and discussed. The second part of the paper presents the results of calculations for passenger and freight elevators, confirming the correctness of the methods included in the Czechoslovakian Standard.

1. CALCULATION OF STRESS AND DEFLECTION

In calculating the guide-rails three operating conditions should be taken into consideration:

(a) Running conditions with the load unevenly distributed on the car floor
(b) Safety gear operation
(c) Loading or unloading respectively.

1.1 National and International Standards

The calculation of stress in guide-rails is usually carried out for (b), while the calculation of deflection concerns quite different operating conditions, namely (a) and/or (c).

A 17.1

The method of calculation specified in the American Safety Code A 17.1 is not transparent as it is based on graphs for the calculation of guide-rail size without presenting the theoretical background. In other words it states what should be done and not why it should be implemented in a particular way. In the principal graph the maximum suspended weight of the car and its rated load, or the maximum suspended weight of the counterweight, including the weight of compensating and travelling cables, is specified for different sizes of guide-rails and the bracket spacing used, in the case of a single car or cwt safety gear being used. Where two car or cwt safety gears are employed the loads may be increased by multiplying factors depending on the vertical distance between the safety gears. The graphic calculation included in A 17.1 concerns the safety gear operation.

In addition A 17.1 specifies the maximum stress in guide-rail or its reinforcement due to the horizontal forces imposed on the guide-rail during loading, unloading and running, calculated without impact, as 103 Nmm^{-2} based on the class of loading and the maximum permissible deflection as 6.3 mm. Unfortunately, it does not present the method of calculation.

AS 1735.2

The Australian Standard AS 1735.2 seems to be most comprehensive as it specifies the calculation of stress not only for the safety gear operation,
but also for other operating conditions.

(i) Loading, unloading and/or running

For the determination of stresses and deflections the following formulae are presented:

$$\sigma = \frac{F \times l}{6\ W_x} \qquad (Nmm^{-2}) \qquad \text{... (1.1)}$$

where
σ is stress in bending (Nmm^{-2})
W is load on guide-rail (N)
l is vertical distance between the centres of guide-rail brackets (mm)
W_x is modulus in bending of the cross-sectional area of the guide-rail, referring to the gravity axis x - x (mm^3).

$$y = \frac{F \times l^3}{96\ E \times J_{min}} \qquad (mm) \qquad \text{... (1.2)}$$

where
y is deflection of the guide-rail (mm)
E is modulus of elasticity of the material of guide-rails (Young's modulus) (Nmm^{-2})
I_{min} is minimum moment of inertia of the guide-rail cross-section (mm^4).

The formulae are not in compliance with the theory of bending, but seem to have been altered for empirical reasons. Each span of guide-rail is considered a simple beam having ends supported. If the supports of the beam (guide-rail) are pin-pointed, the numbers in denominators of the equations should be 4 for stress and 48 for deflection, while they should be 8 and 192 in the case of the beam being rigidly fixed on both ends. The maximum stress should not exceed 140 Nmm^{-2} based on a yield strength of the material of 228 Nmm^{-2}, while the maximum deflection is specified as 3 mm in the plane of guide-rails (y - y) and 6 mm in the perpendicular plane (x - x) - see Fig.1.

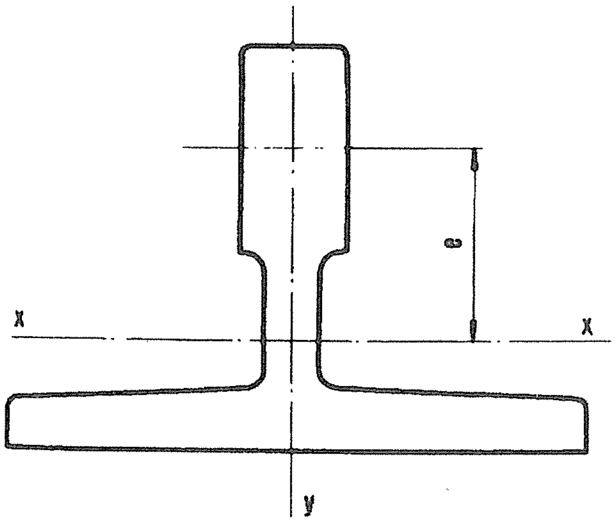

Figure 1: Guide rail

(ii) Safety gear operation

The stresses induced in guide-rails during the safety gear operation must be of such values to satisfy the equation

$$\frac{f_c}{p_c} + \frac{f_{bc}}{p_{bc}} \leq 1 \qquad \text{... (1.3)}$$

where p_c is maximum permissible compressive stress (Nmm^{-2}); it is quoted in the code in dependence on the slenderness ratio of the guide-rail
f_c is calculated compressive stress (Nmm^{-2})
p_{bc} is permissible bending stress of 173 Nmm^{-2} based on a yield strength of 228 Nmm^{-2}.

$$f_{bc} = \frac{\phi \times F \times e}{4W_x} + \frac{\phi \times K \times l}{6W_x \times h} \quad (Nmm^{-2}) \qquad \text{... (1.4)}$$

where ϕ is decelaration factor depending on the type of safety gear (i.e. the average retardation)
F is half the weight of the empty car plus one quarter of the rated load for passenger elevators; for freight elevators it depends on the class of loading
e is the distance between the point of application of the braking force due to the safety gear operation and the axis of gravity x - x of the guide-rail cross-sectional area (see Fig.1)
K is the largest turning moment caused by eccentric static load in the car (Nm)
h is vertical centre distance between the upper and the lower guide shoes (mm).
l and W_x have been already defined.

Equation (1.3) incorporates:

(1) Stress in buckling as the maximum permissible compressive stress p_c is dependent upon the slenderness ratio of the guide-rail and the minimum radius of gyration is taken into consideration when the slenderness ratio is calculated.
(2) Bending stress due to the eccentric position of the braking force during the safety gear operation (see distance *e* in Fig.1).
(3) Bending stress due to the uneven distribution of the load in the car.

Some aspects of the Australian method of calculation may become a matter of discussion, e.g. guide-rails are never subjected to combined stress in buckling and bending. Furthermore, when calculating the bending stress due to the safety gear operation at mid-span there should be 2 in the denominator of the equation instead of 4 as ϕ x F is the braking force on a single guide-rail.

The resultant stress in guide-rails due to the safety gear operation may be expressed graphically as seen in Fig.2. To satisfy the equation (1.3) the resultant value must lie inside the triangle OAB.

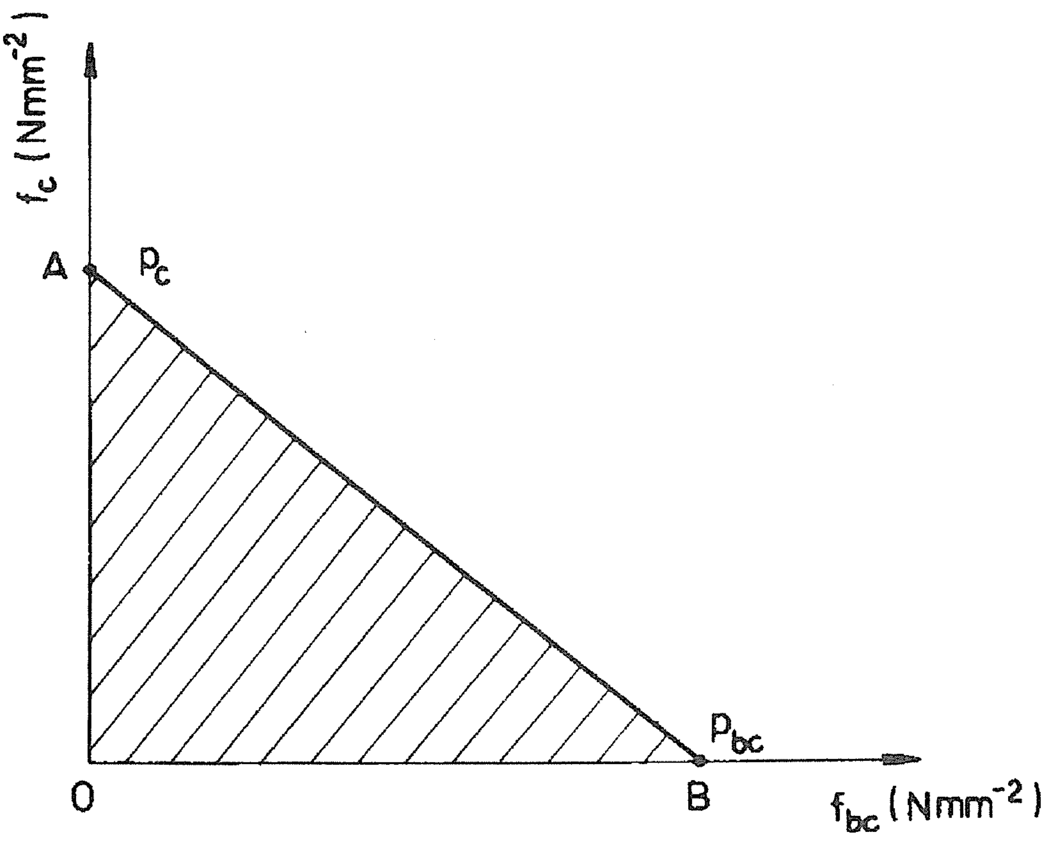

Figure 2: Resultant stress in guide rails

EN 81.1

The stress in guide-rails during the safety gear operation is calculated in case of buckling and is given by the formula

$$\sigma_k = \frac{F_b \times \omega}{S} \quad (Nmm^{-2}) \qquad \text{... (1.5)}$$

where F_b is braking force due to safety gear operation (N)
ω is the buckling factor, quoted in national standards for steels of 370 mm^{-2} grade or 520 Nmm^{-2} grade respectively, as a function of the slenderness ratio
S is cross-sectional area of the guide-rail (mm^2).

The guide-rail is supposed to behave like a simple beam of two pin-pointed supports, subjected to the braking force in its longitudinal axis. Buckling is assumed to take place in the plane of the smallest rigidity in bending of the guide-rail.

The formulae for the evaluation of the braking force F_b are specified in EN 81.1.

The value of σ_k must not exceed:

140 Nmm^{-2} for steel of 370 Nmm^{-2} grade or
210 Nmm^{-2} for steel of 520 Nmm^{-2} grade.

Maximum deflection of guide-rails during loading, unloading or running is specified as 3 mm, but the method of calculation is not mentioned.

CSN 27 4030

(i) Safety gear operation

In conformity with the above mentioned Czechoslovakian standard the calculation of stress in buckling during the safety gear operation is carried out in the same way as in EN 81.1, but only in the case of the slenderness ratio λ being in excess of 105. For $\lambda \leq 105$ the calculation in combined pressure and bending takes place instead.

The formula is as follows:

$$\sigma = F_b \times \left(\frac{1}{S} + \frac{e}{2W_x} \right) \qquad (Nmm^{-2}) \qquad \ldots (1.6)$$

The meaning of all symbols used in this formula has been already explained.

(ii) Running conditions

In the Czechoslovakian standard the method of calculating partial deflections of guide-rails in two perpendicular planes is specified as well as the magnitudes of lateral forces acting upon the guide-rails at right angles and generating these deflections. Details were presented at ELEVCON 90 and are included in ELEVATOR TECHNOLOGY 3.

2. COMPARATIVE CALCULATIONS OF STRESS IN GUIDE-RAILS

The analysis of stress and deflection of guide-rails under all operating conditions carried out some 10 years ago revealed an essential fact: during the safety gear operation the guide-rail was never subjected to the stress in combined buckling and bending, it was always a simultaneous effect of pressure and bending. This knowledge has been reflected in the Czechoslovakian standard.

The fundamentals of the analysis and calculation were as folows: the guide-rail was considered a continuous beam with a variable number of supports. The Theorem of Three Moments was used as a method for solution. All supports were assumed to be at the same level (on one straight line). In spite of this simplification, which need not be correct in the practice, complicated formulae were achieved.

Comparative calculations both in buckling and combined pressure and bending were carried out for a number of passenger and freight elevators and the results are presented in Table 2. The technical parameters of these elevators are quoted in Table 1. As seen in Table 2, up to the slenderness ratio λ of about 105 the calculation in combined pressure and bending is decisive, while the calculation in buckling can be solely carried out for higher values of λ only.

3. CONCLUSION

Having compared the methods of calculation of stress and deflection in guide-rails included in the above mentioned standards the Czechoslovakian standard seems to be reflecting the actual state of guide-rails better than other standards, while remaining quite simple and easy to understand.

Anyway, it is necessary to emphasize that formulae and graphs included both in national and international standards are quite simple because the problem itself had been considerably simplified and significant features had been neglected before they were accomplished. Consequently the actual state of guide-rails in operating conditions may

Table 1: Technical parameters

Type of elevator	Passenger				Freight				Hospital bed elevator
Characteristic	320/ 0.7	500/ 0.7	1000/ 2.5	630/ 1.6	1000/ 0.7	2000/ 0.36	3200/ 0.5	5000/ 0.25	1600/ 1.0
Rated load (kg)	320	500	1000	630	1000	2000	3200	5000	1600
Rated speed (ms^{-1})	0.7	0.7	2.5	1.6	0.7	0.36	0.5	0.25	1.0
Mass of the car (kg)	472	703	1500	1100	1180	1575	2745	4310	1815
Mass of the counter-weight (kg)	632	953	1950	1445	1630	2475	4185	6560	2443
Car width (mm)	1300	1100	1600	1100	1620	1162	2220	2520	1400
Car depth (mm)	950	2200	1400	1400	2345	2360	3845	4460	2400
Vertical distance of car guide shoes (mm)	2757	2575	3625	3315	3175	3240	3283	3988	3700
Type of guide rail	60x70	70x90	70x90	70x90	70x90	70x90	2x L 90x60x8	2x L 90x60x8	90x120
Spacing of guide rail brackets (mm)	4000	3000	3300	4000	2600	1900	1600	1250	3300
Safety gear	inst.	inst.	prog.	prog.	inst.	inst.	inst.	inst.	prog.

Table 2: Results of calculations

Type of elevator	Passenger				Freight				Hospital bed elevator
Characteristic	320/ 0.7	500/ 0.7	1000/ 2.5	630/ 1.6	1000/ 0.7	2000/ 0.36	3200/ 0.5	5000/ 0.25	1600/ 1.0
Braking force (N)	19424	29504	24525	16971	53465	87677	145801	228328	33501
Slenderness ratio	289.9	169.5	186.4	226.0	146.9	107.3	71.6	56.0	149.7
Buckling factor for steel of 370 Nmm^{-2} grade	12	4.82	5.86	8.63	3.64	2.05	1.43	1.26	3.78
Buckling factor for steel of 520 Nmm^{-2} grade	18	7.23	8.79	12.94	5.46	2.91	1.61	1.36	5.68
Stress in combined pressure and bending (Nmm^{-2})	36.6	43.4	36.1	25.0	78.7	129.0	113.0	176.9	35.1
Stress in buckling for steel of 370 Nmm^{-2} grade	209.8	83.1	84.0	85.6	113.7	105.1	88.0	121.4	57.6
Stress in buckling for steel of 520 Nmm^{-2} grade	314.7	124.7	126.0	128.4	170.6	149.1	99.1	138.0	86.6
Lateral force on guide rail F_{x1} (N)	84.5	327.4	296.0	203.9	565.3	1116.5	2872.3	4285.6	1371.4
Lateral force on guide rail F_y (N)	185.0	261.9	541.2	256.3	624.9	879.6	2653.5	3874.3	742.4
Partial deflection of guide rail y_x (mm)	0.177	0.115	1.380	1.694	1.290	0.994	0.691	0.492	3.202
Partial deflection of guide rail y_y (mm)	1.80	0.53	1.83	1.55	1.04	0.57	0.59	0.41	1.07
Total deflection y (mm)	1.81	0.55	2.30	2.29	1.65	1.15	0.91	0.64	3.38

be different than calculated. The future evolution should lead to the usage of sophisticated computer programs reflecting actual conditions and taking into consideration all influencing factors.

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CSN 27 4030 Electric Lifts. Calculations of Lift Components.

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Lubomir Janovsky has been employed as a Reader with the Technical University in Prague, Faculty of Mechanical Engineering. He is also an elevator and escalator consultant. He has been a foreign correspondent for ELEVATOR WORLD since 1978, a Vice Chairman of the Executive Board of IAEE and a member of the International Committee of ELEVATORI. He gained his Ing. (Masters level) in Mechanical Engineering in 1957 and his CSc. (doctorate) in Vertical Transportation, both from the Technical University of Prague. He has written numerous books and papers on vertical transportation and materials handling, including ELEVATOR MECHANICAL DESIGN, published by Ellis Horwood Ltd. in 1987.