The Strength and Design Criteria for a Lift Guiding System Revisited

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Abstract. Lift guides are subjected to variable loading conditions under loading, normal operation / running, and stopping (under the operation of the safety gear). Safety codes demand that under these conditions the guiding system must be designed with adequate strength to withstand bending and buckling and impose limits on the permissible stresses and deflections. Furthermore, maintaining special ride quality requirements of a lift system might impose additional limits on guide deflections. There have been extensive studies carried out to develop models that can provide adequately accurate results for stresses and deflections that must satisfy these conditions. For example, BS EN81-50 / 20:2020 specifications for guide rail bending deflections are based on a three-span beam model. On the other hand, the model for evaluation of the maximum bending moments is a single span beam with one end simply supported and the other end constrained as built-in (fixed). The influence of various boundary conditions and the issue of selecting and providing accurate, practical models for pragmatic strength evaluation of a lift guiding system are discussed and appraised in the paper.

1 INTRODUCTION

The guiding system is the most important interface between the lift installation and the building structure. It is well established that the quality of an elevator installation is founded on the quality of the guiding system. Without an adequate design followed by careful and correct installation, it will, to all intents and purposes, be impossible thereafter to provide a lift system with an adequate or acceptable quality of ride [1].

Under various loading conditions, defined in EN81-20 [2] clause 5.7.2.2 as

- normal operation - running;
- operation - loading and unloading;
- safety device operation,

the guides are subjected to bending / twisting and buckling forces (see Figure 1). The safety code imposes limits on the permissible stresses and deflections. It demands also that the forces must be evaluated in ‘worst case’ conditions. For each loading case the combination which is likely to give rise to the maximum guide rail stress should be considered.

2 BENDING

2.1 Multispan beam model in calculations of deflections

A guide rail can be considered as a multispan beam subjected to lateral loading as shown in Figure 2. The elastic curve formed by the rail with N spans can then be described by the following equation:

$$y(z) = \theta_0 z + \frac{1}{EI_x}\left[\sum_{i=1}^{N-1} R_i \langle z - z_i \rangle^3 - F_y \langle z - z_F \rangle^3\right] \tag{1}$$

where a step function $f(z) = \langle z - z_i \rangle^n$ is introduced [3]. F_y represents a lateral force acting upon the rail at $z = z_F$, R_i is the reaction force at ith support positioned at $z = z_i$, θ_0 represents the slope angle at $z = 0$, and E and I_x are the modulus of elasticity and the 2nd moment of the rail cross-sectional area about the x axis.

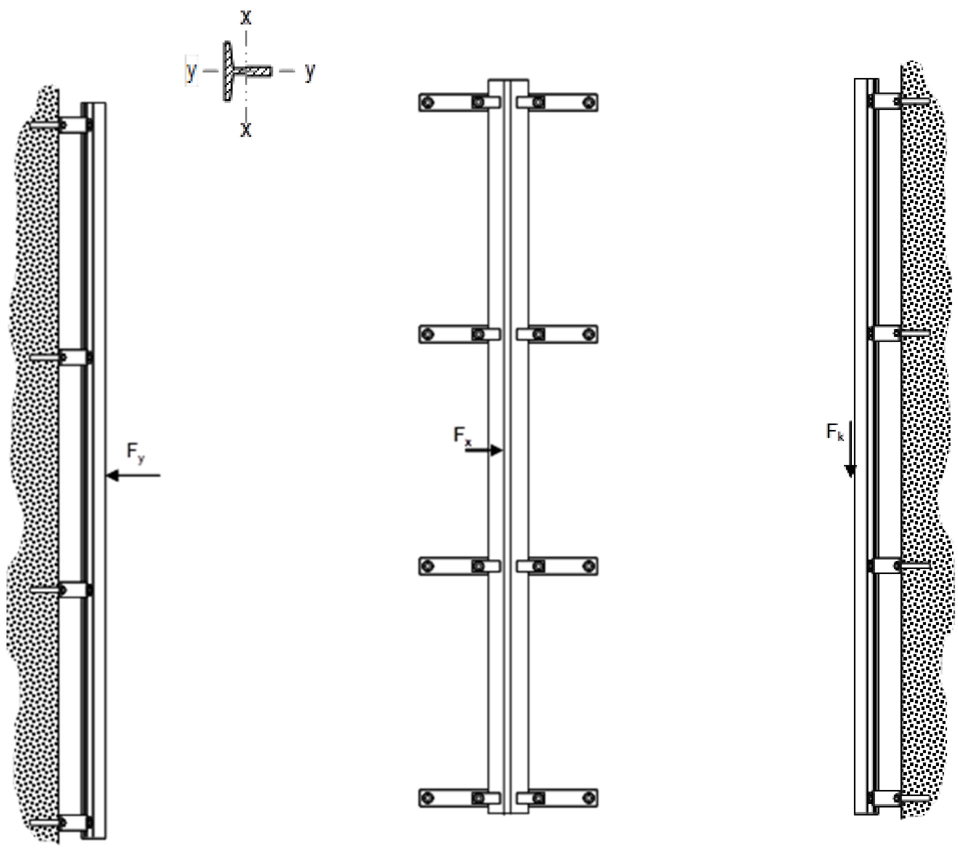

Figure 1 Forces acting on a guide rail: (a) bending, (b) bending and flange bending, (c) buckling [1]

In BS EN81-50:2020 [4] clause 5.10.6 the following formula is given to calculate the deflections of guide rails in the *y* direction

$$\delta_y = 0.7\frac{F_y l^3}{48EI_x} + \delta_{str-y} \tag{2}$$

where I_x is the second moment of area about the neutral *x*-*x* axis of bending, and δ_{str-y} is the deflection of the building structure in the *y* direction. The criterion $\delta_y \le \delta_{perm}$ must then be satisfied, where the permissible deflections δ_{perm} are specified in BS EN81-20:2020 clause 5.7.4.6.

The first term at the right-hand side of (2) is the elastic deflection of the guide rail calculated under the assumption that the shaft walls are perfectly rigid and can be presented as

$$\delta_{rail-y} = \alpha\frac{F_y l^3}{48EI_x} \tag{3}$$

The value and origin of coefficient of $\alpha = 0.7$ in Equation 2 can be explained as follows. Consider a 3-span guide rail with the lateral force F_y applied at $z = z_F = l/2$. Equation 1 (with N = 3) can be applied to determine the bending deflections of the guide rail. The system is statically indeterminate and the most convenient method to calculate the reaction forces at supports is the classical superposition method. The static equilibrium conditions then yield the three reactions $R_0 = 0.4F_y$, $R_1 = 0.725F_y$ and $R_2 = -0.15F_y$ that need to be used in Equation 1 to calculate the deflections. The boundary conditions of zero deflection at the support points yield the slope angle at z = 0 as $\theta_0 = -\frac{11F_y l^2}{240EI_x}$ so that the maximum deflection which occurs at $z = z_F = l/2$ can be determined as

$$y\left(\frac{l}{2}\right) = -\frac{7}{480}\frac{F_y l^3}{EI_x} \cong -0.0146\frac{F_y l^3}{EI_x} \tag{4}$$

By taking the absolute value of $y\left(\frac{l}{2}\right)$ we can re-write Equation 4 as

$$y_{\max} = \left|y\left(\frac{l}{2}\right)\right| = \frac{7}{10}\frac{F_y l^3}{48EI_x} = 0.7\frac{F_y l^3}{48EI_x} \tag{5}$$

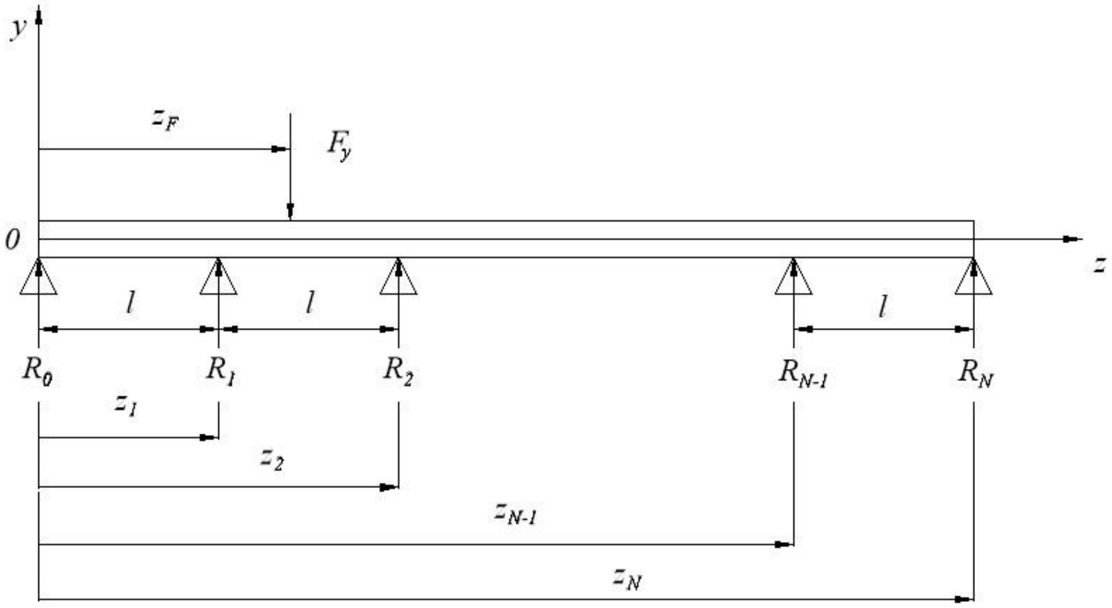

Figure 2 Guide rail beam model in bending

On the other hand, if the load F_y is applied further to the right **from the origin *O***, the maximum deflections are smaller. For instance, in the scenario when the lateral force F_y applied at $z = z_F = 3l/2$ (at the mid-point of the 2nd span) the three reactions forces are $R_0 = -0.075F_y$, $R_1 = R_2 = 0.575F_y$, and the slope angle at $\mathbf{z} = 0$ is $\theta_0 = \frac{F_y l^2}{80EI_x}$. Equation (1) then yields the following result:

$$y\left(\frac{3l}{2}\right) = -\frac{99}{180}\frac{F_y l^3}{48EI_x} \cong -0.0115\frac{F_y l^3}{EI_x} \tag{6}$$

which yields

$$y_{\max} = \left|y\left(\frac{3l}{2}\right)\right| = 0.55\frac{F_y l^3}{48EI_x} \tag{7}$$

Thus, it is evident that the coefficient α of 0.7 that is used in BS EN81-50:2020 (see Equation 2) is based on a 3-span beam model with the lateral load applied at the mid-point of the first span.

As per BS EN81-50:20 2020 the guide rail deflections in the direction x (perpendicular to y) should be calculated as

$$\delta_x = 0.7\frac{F_x l^3}{48EI_y} + \delta_{str-x} \tag{8}$$

where I_y is the second moment of area about the y-y axis, and δ_{str-x} is the deflection of the building structure in the x direction. It should be noted that this is in general valid only when the distance between the centre points of the upper and lower guide shoes/ rollers is greater or equal to 1.5 l [5].

2.2 Evaluation of the bending stresses

Considering the bending stresses, the maximum bending moment with the lateral force F_y acting upon the rail is given as

$$M_{\max} = \beta F_y l \tag{9}$$

where $\beta = 0.2$ when $z_F = l/2$, and $\beta = 0.175$ when $z_F = 3l/2$. The distribution of normal bending stresses is then determined as

$$\sigma_x = \frac{M_{\max} y}{I_x} \tag{10}$$

where y denotes the distance from the neutral x-x axis.

It should be noted that according to BS EN81-50:2020 clause 5.10.2.1 the maximum bending moment should be evaluated by

$$M_{\max} = \frac{3}{16} F_y l \tag{11}$$

where $\beta = 3/16 = 0.1875$ is used which corresponds to the beam model presented in Figure 3. In this model the guide rail a single span beam with one end simply supported and the other end constrained as built-in (fixed) and the lateral load F_y is applied at the midspan ($z_F = l/2$). The maximum bending moment (9) is evaluated at $z = l$ (at the right boundary). In this case the maximum deflection occurs at $z = \frac{l}{\sqrt{5}} \cong 0.447l$ and is given as

$$y_{\max} = \left| y\left(\frac{l}{\sqrt{5}}\right) \right| \cong 0.0093 \frac{F_y l^3}{EI_x} \tag{12}$$

which is a smaller value than for the case of the 3-span simply supported beam model used in the calculations of permissible deflections.

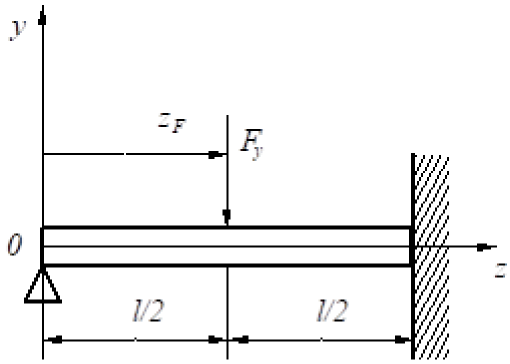

Figure 3 Equivalent beam model for evaluation of the guide bending stress

2.3 Flange bending

Considering flange bending the worst case will occur when the force is applied co-incident with a guide bracket, and that the maximum stress will occur in the web joining the blade of the guide to the foot (see Figure 4).

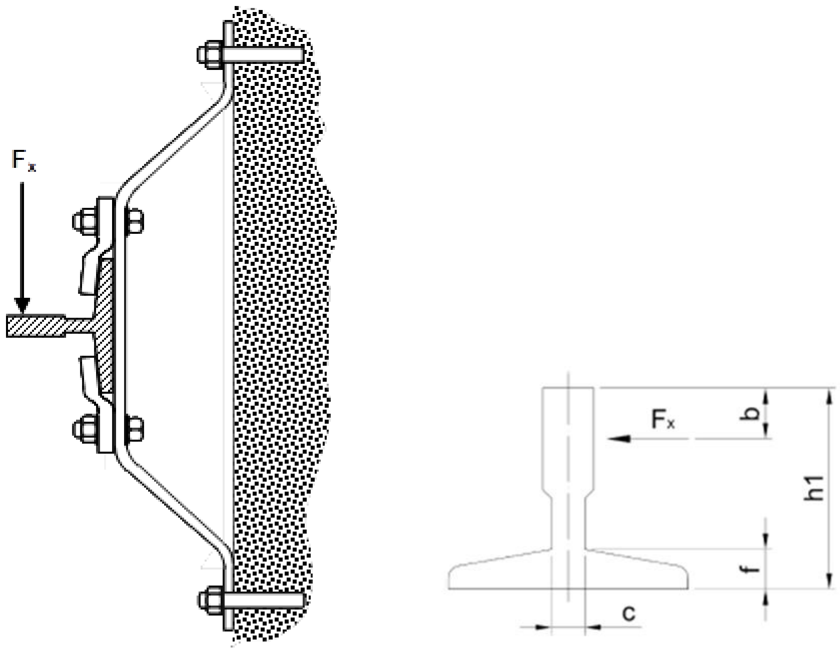

Figure 4 Flange bending [1, 4]

As per BS EN81-50:2020 clause 5.10.5 the stress is to be determined by

$$\sigma_F = \frac{1.85F_x}{c^2} \quad \text{for roller guide shoes}$$
$$\sigma_F = \frac{6F_x\left(h_1 - b - f\right)}{c^2\left[l_s + 2\left(h_1 - f\right)\right]} \quad \text{for sliding guide shoes} \tag{13}$$

where b represents half the width of the guide shoe lining, c denotes the thickness of the web joining the blade to the foot, f is the foot depth at the connection with the blade, h_1 is the height of the guide rail, l_s is the length of the guide shoe lining.

2.4 Seismic and dynamic effects

It should be noted that for lift installations subjected to seismic conditions it is necessary to add seismic forces as short-term loads to determine the guide system design parameters. The limits on the permissible stresses and deflections imposed by BS EN81-77:2022 clause 5.8.2 [6] should then be applied. A comprehensive overview of the dynamic interactions between the guide system and the car / counterweight / suspension system, with a discussion of protection measures, are presented in [7].

3 BUCKLING

A slender structural element / rod in compression might be subjected to buckling (lateral deflections) if the axial loading becomes large enough to exceed its critical value. The critical load P_{cr} is the maximum axial load the member can support when it is on the verge of buckling. For any load which is larger than P_{cr} the structural member will become unstable with its center being displaced laterally by a large amount. In fact, the rod will then be bending as a beam subjected to axial loading.

In a lift guiding system a section of the guide rail between each pair of brackets is subject to buckling due to the braking force F_b arising from safety gear operation (see Figure 5a). The critical load (often referred to as the *Euler critical buckling* force) depends on the end support conditions. For example, for a pinned – pinned conditions the critical load is given as

$$P_{cr} = \frac{\pi^2 EI}{l^2} \tag{14}$$

where *EI* represents the least flexural rigidity of the column and *l* denotes the unsupported length (see Figure 5b). The corresponding critical stress is then defined by

$$\sigma_{cr} = \frac{\pi^2 E}{\lambda^2} \tag{15}$$

where *E* is the modulus of elasticity and λ is the slenderness ratio given as *l/r* with *r* representing the minimum radius of gyration of the cross-sectional area.

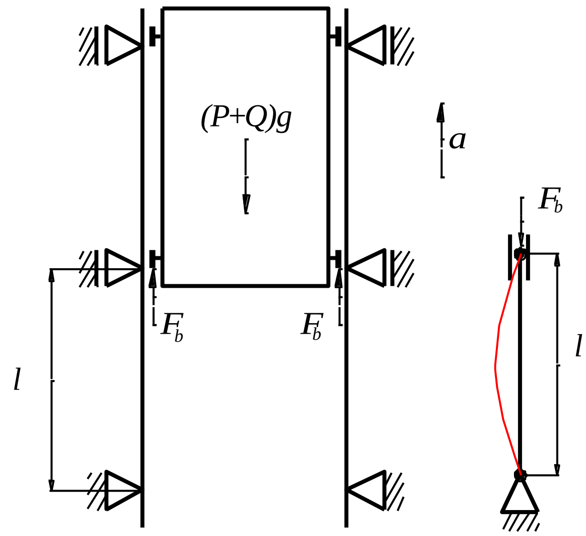

Figure 5 Guide rail buckling

Considering the forces acting on the car as shown in Figure 5a second Newton's law yields the expression to determine the braking force as follows:

$$(P+Q)a = 2F_b - (P+Q)g$$
$$F_b = \frac{P+Q}{2}(g+a) \tag{16}$$

where *a* is the deceleration, *g* denotes the acceleration of gravity, and *P* and *Q* denote the masses of the car and rated load, respectively. Following the code requirements [4] to determine the buckling stress in the guide rail the 'omega' method is used. In this method where the compressive stress due to safety gear operation is calculated and then modified by the factor ω, dependent upon the material of the guide and its slenderness ratio λ as follows:

$$\sigma_b = \frac{F_b \omega}{A} \tag{17}$$

where the coefficient ω values are determined in terms of the guide material tensile strength and the slenderness ratio λ. The stresses calculated from equation (9) should not exceed the permissible value determined as $\sigma_{perm} = \frac{R_m}{FS}$ where R_m denotes the ultimate tensile stress (tensile grade) of the rail material and *FS* is the safety factor.

However, in the case of buckling forces a further consideration must be given to forces due to the weight of any other equipment supported from the guide rails. Therefore, the code requirement is that the buckling stress on the guide is calculated as

$$\sigma_k = \frac{(F_k + k_3 M_{aux})\omega}{A} \tag{18}$$

where F_k is the buckling force due to safety gear operation (i.e. $F_k = F_b$), M_{aux} is the weight of any additional equipment supported on the guide, and k_3 is a coefficient reflecting the nature of equipment supported from the guide (e.g. the lift machine in a machine roomless configuration).

4 CONCLUSIONS

It is evident from the discussion presented above that the safety codes and engineering practice have established some straightforward criteria for a lift guiding system to be designed with adequate strength. However, the evaluation of the actual loading / main forces (such as F_x, F_y and F_k) require careful consideration.

Their magnitude will depend upon the location of the centre of mass of the car and the location of the centre of mass of the load relative to the suspension point and to the guide rails. Annex C of BS EN 81-50:2020 provides an example for calculation of guide rails, which includes some guidance as to how to evaluate non-uniform or off-centre locations for the carload.

Furthermore, some thought must be given to the issue of support of the guide rail through the guide brackets, that provide the interface between the guiding system and the building structure. The key point is that the guide rail must be able to slip vertically through the guide clip, and yet be restrained from horizontal movement in any direction.

The issue of ageing and building settlements and their influence on guide rail systems in long term use should be considered. For example, the building settlement and movement, either in the early stages of the building life, or later on as the building is loading and unloaded by its occupants, may lead to changes in the vertical spacing of the guide brackets. Such changes will lead to distortion of the guide rails, with consequent effects on the ride quality.

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BIOGRAPHICAL DETAILS

Dr Stefan Kaczmarczyk is Professor of Applied Mechanics and Postgraduate Programme Leader for Lift Engineering at the University of Northampton, UK. His expertise is in applied dynamics and vibration, computer modelling and simulation with applications to vertical transportation and material handling systems. He has published over 150 journal and international conference papers in this field. He is a Chartered Engineer, a Fellow of the Institution of Mechanical Engineers and a Fellow of the Higher Education Academy.