Lift system calculations in EN 81-50

Nick Mellor

Lift and Escalator Industry Association (LEIA), UK
nick.mellor@leia.co.uk

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Abstract. Lift engineers responsible for the design of lift systems conforming to EN 81-1 should be conversant with the equations for guide rails, rope traction and rope factor of safety in that standard. The new EN 81-20 references EN 81-50 which now includes such equations, with some minor changes, and some additional helpful guidance.

Despite the apparent complexity of these equations, making calculations need not be a daunting prospect. The use of manual or spreadsheet methods are valuable in gaining an engineering appreciation for these calculations. Such an appreciation is important in interpreting the results obtained from software packages and might not be gained simply by "plugging in" numbers. The use of such software packages (which might not be infallible or which might incorporate assumptions not clear to the user) should be subject to verification; one method is comparison with manual calculations.

This paper looks at the main changes in the calculations for guide rails, rope traction and rope factor of safety and through examples provides a means to assess the implications of these changes.

The paper also reflects on some underlying assumptions in these equations and some engineering implications from their use. Implications for conformity with the new standards will be touched-on. Future directions for the development of the standards will be mentioned.

1. INTRODUCTION

For many years EN 81-1 [1] and EN 81-2 [2] have been the standards to which many new traction and hydraulic lifts have been designed. These standards, after a period of co-existence with the new standards EN 81-20 [3] and EN 81-50 [4], will be withdrawn. The normative requirements for both traction and hydraulic lifts are to be found in EN 81-20 while other requirements, including for elements of lift system calculations, are to be found in EN 81-50. The calculations in EN 81-50 are referenced from EN 81-20 so these elements of EN 81-20 are also applicable.

This paper looks at the calculations in EN 81-50 and how these have changed from those in EN 81-1 and EN 81-2. The clauses within EN 81-20 and the calculations in EN 81-50 which they reference are reviewed and compared with their predecessors in EN 81-1 and EN 81-2. For calculations for guide rails, rope traction and rope factor of safety, sample calculations are presented which illustrate the changes made.

The introduction of EN 81-1 and EN 81-2 in 1998 came at a time when much more use was being made of software packages to make lift system calculations[1]. The use of proprietary software packages or the use of spreadsheets allowed rapid calculation which was an aid to more optimal design and selection of components.

However, the use of such packages, especially those whose underlying equations and assumptions are not transparent to the user, raise issues which need to be considered by users:

- Simply taking the lift parameters and "plugging in" these numbers is less likely to promote an appreciation of the fundamentals than would be gained through making manual calculations or even implementing these on a spreadsheet.

[1] Calculations for the system torque are not included within either EN 81-1 or EN 81-20.

- Without such an appreciation, the output from software packages might not be scrutinized so critically and errors or opportunities for improvement might not be identified by the user.
- Most lift designers inevitably now have quality systems and certification to relevant standards such as ISO 9000, possibly supplemented by other requirements such as those imposed by the EC Lifts Directive. The results of any engineering calculations should be checked. One method of verifying the correct operation or "calibration" of software packages is by the comparison of their results with the results of manual calculations.

2. PARAMETERS USED FOR CALCULATION

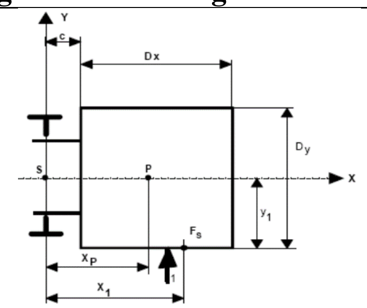
The parameters of two lift configuations which are the subjects of calculation in this paper are tabulated below. Table 1 lists parameters for a conventionally guided situation where it is assumed the line of suspension, centre lines of the guides and centre of the lift car are all coincident. Table 2 lists parameters for a cantilever guided situation suspended from point **s** in the figure in Table 2, reproduced from G.7.4 of EN 81-1 [1]. The parameters listed in Table 2 are those which differ from those in Table 1 owing to the different guidance; parameters for suspension and traction are common to both configurations should be taken from Table 1. All symbols are as used in EN 81-20 [3] and EN 81-50 [4]. There is no compensation included.

Table 1: Key parameters for conventionally guided configuration

Parameter	Value	Parameter	Value
Distance between guide fixings, l	4000 mm	Traction sheave angle of wrap, α	3.14 rad
Car guide rails (ISO 7465)	T127-1/B	Traction sheave groove	Undercut V
Tensile strength of guide rails, R_m	370 N/mm^2	Groove angle (50°), γ	0.87 rad
Overall height of guide rails	20 m	Undercut angle (105°), β	1.83 rad
Car size: D_x D_y	1400 mm 1600 mm	Rated speed of lift Reeving ratio	1 m/s 2:1
Rated load, Q	1000 kg	Traction sheave diameter, D_t	320 mm
Number of car guides, n	2	Divertor pulley diameters, D_p	320 mm
Empty car weight, P	1250 kg	Reduced mass of a pulley	30 kg
Distance between guide shoes, h	2800 mm	No. pulleys on car side	2
Safety gear impact factor, k_1	3	No. pulleys counterweight side	1
Dimensions x_p, y_p	0	Rope diamter, d_r	8 mm
Acceleration due to gravity, g_n	9.81 m/s^2	Mass of ropes on one fall, M_{SR}	25 kg
π	3.14	Mass of travelling cables, M_{TRAV}	12 kg
Note: guide rail parameters from ISO 7465 [5].		Counterweight balance factor	0.45

Table 2: Key parameters for cantilever guided configuation differing from Table 1

Parameter	Value
Distance between guide fixings, l	2500 mm
Car guide rails (ISO 7465)	T127-1/B
Car size: D_x D_y	1100 mm 2100mm
Distance from guides to car wall, c	200 mm
Distance between guide shoes, h	2800 mm
Dimensions x_p, y_p	500 mm; 0 mm



3. GUIDE RAIL CALCULATIONS

EN 81-1 [1] includes normative requirements for the maximum permissible stresses and for guide rail deflections with a calculation method for forces, stresses and deflection in *informative* Annex G. EN 81-20 [3] has normative requirements generally as EN 81-1 except for the following additions.

- There is a requirement to consider the combination of deflections of guide rails, brackets, play in the guide shoes and straightness of the guide rails which must be taken into account in order to ensure a safe operation of the lift. In cases where previously no allowance was made for deflection of these additional elements, then guide rail selections and maximum fixing intervals may need to be revised.
- Equations for vertical loads include self-weight of guides and push-through force of clips (for longer travels or where building settlement is significant) are included as *normative* requirements. Depending on the travel of the lift and the pull-through force of guide clips, the additional vertical load could be significant and require a review of buckling calculations.
- EN 81-20 requires that guide rails to be calculated according to one of:
  a) EN 81-50, 5.10; or
  b) EN 1993-1-1; or
  c) Finite Element Method (FEM).
  So it is now a *normative* requirement to use one of these methods. While the latter two methods might provide useful alternatives, the discussion here focuses on 5.10 of EN 81-50.

EN 81-50 [4] clause 5.10 has equations generally as EN 81-1 Annex G for calculating bending and buckling stresses, and deflections. It includes an additional equation for evaluating flange bending with sliding shoes. EN 81-50 Annex C is informative and has an example for calculation based on the general case and not including a number of different configurations as in EN 81-1 Annex G. These calculations are to demonstrate the adequacy of a known design solution including guide size, number or guides and fixing interval.

The following are calculations for the two configurations in Tables 1 and 2. In each case, only the worst case situation is calculated for the car guide rails i.e. for the engagement of the safety gears. Depending on the nature of the application, it might be that safety gear operation is not the worst case situation for conformity as the level of permitted stress is lower for normal running. To demonstrate conformity, all loading situations should be calculated and checked to be within the relevant permitted stress for all guide rails used.

3.1 Sample calculations - conventionally guided configuration as Table 1

From EN 81-20, 5.7.2.3.5, the vertical force, F_v, for the car guides, where M_g is the self-weight of the guide rails and F_p is the push through force from guide clips which will be neglected here, is:

$$F_v = \frac{k_1 g_n (P+Q)}{n} + \left(M_g g_n\right) + F_p. \qquad (1)$$

Which can be evaluated as:

$$F_v = 3x9.81x(1250 + 1000)/2 + (17.85x20x9.81) = 36611 \text{ N}.$$

To calculate the buckling stress, σ_k, 5.10.3 of EN 81-50, like EN 81-1, uses the "omega method", ω, (although it does not retain the tables of EN 81-1 Annex G so values need to be calculated). This is based on the slenderness ratio, λ; the ratio of the distance between the guide rail fixings and the lesser of the two radii of gyration of the guide rail:

$$\lambda = \frac{l_k}{i} = 4000/23.61 = 169.4. \qquad (2)$$

From 5.10.3 of EN 81-50, for R_m=370 N/mm^2 and for $115 < \lambda \leq 250$, $\omega = 0.00016887\lambda^2 = 4.85$. (3)

$$\sigma_k = \frac{(F_v + k_3 M_{aux})\omega}{A} \tag{4}$$

M_{aux} and k_3 are the weights of auxiliary equipment and relevant impact factor which will be assumed to be zero here (although in many cases there are loads supported by the guide rails as in the case of many machine room-less designs). Then:

σ_k = (36611x4.85)/2274 = 78 N/mm^2

The calculation of bending loads for safety gear operation is included in C.2 of informative Annex C of EN 81-50 which illustrates the evaluation of worst case bending stress owing to the car load being offset relative to the x-axis (case 1) and y-axis (case 2).

$$F_x = \frac{k_1 g_n(Qx_q + Px_p)}{nh}. \tag{5}$$

$$F_y = \frac{k_1 g_n(Qy_q + Py_p)}{\left(\frac{n}{2}\right)h}. \tag{6}$$

Combining the equations for M_y and σ_y to give the bending stress relative to the y axis:

$$\sigma_y = \frac{3F_x l}{16W_y} \tag{7}$$

Similarly for the bending stress relative to the x axis, σ_x:

$$\sigma_x = \frac{3F_y l}{16W_x} \tag{8}$$

For case 1 relative to the x-axis, $x_q = D_x/8 = 175$ mm and $y_q = 0$ so $F_{x(1)}$ and $F_{y(1)}$ can be evaluated using these and equations (5) and (6):

$F_{x(1)} = \frac{3xg_n(1000x175)}{2x2800}$ = 920 N and $\sigma_{y(1)} = \frac{3x920x4000}{16x23610}$ = 29 N/mm^2

$F_{y(1)} = 0$ and $\sigma_{x(1)} = 0$

For case 2 relative to the y-axis, $x_q = 0$ and $y_q = D_y/8 = 200$ mm so $F_{x(2)}$ and $F_{y(2)}$ can be evaluated in a similar way:

$F_{x(2)} = 0$ and $\sigma_{y(2)} = 0$

$F_{y(2)} = \frac{3xg_n(1000x200)}{2800}$ = 2102 N and $\sigma_{x(2)} = \frac{3x2102x4000}{16x30650}$ = 51 N/mm^2

The combined bending stress, σ_m, is

$$\sigma_m = \sigma_x + \sigma_y. \tag{9}$$

This has its worst case value for case 2 where σ_m = 51 N/mm^2

The worst case combined bending and compressive stress is:

$$\sigma = \sigma_m + \frac{(F_v + k_3 M_{aux})}{A}. \tag{10}$$

Evaluating this: $\sigma = 51 + \frac{(36611)}{2274}$ = 67 N/mm^2

The combined bending and buckling stress is:

$$\sigma = \sigma_k + 0.9\sigma_m. \tag{11}$$

Evaluating this: σ = 82 + 0.9x51 = 128 N/mm^2

None of these combined stresses are close to the permitted stress of 205 N/mm^2 (for steel of R_m =370 and safety factor of 1.8 for safety gear operation). All looks well so far except that the flange bending stress and guide rail deflections have not been calculated.

Equations for guide rail deflections in 5.10.6 of EN 81-50 are:

$$\delta_x = 0.7\frac{F_x l^3}{48EI_y} + \delta_{str-x}. \tag{12}$$

$$\delta_y = 0.7\frac{F_y l^3}{48EI_x} + \delta_{str-y}. \tag{13}$$

These can be evaluated as:

$$\delta_x = 0.7\frac{920x4000^3}{48x207000x1499000} + \delta_{str-x} = 2.8\text{ mm}+\delta_{str-x}$$

$$\delta_y = 0.7\frac{2102x4000^3}{48x207000x1879000} + \delta_{str-y} = 5.0\text{ mm}+\delta_{str-y}$$

So the deflection in the y direction, while it might have been close to being considered in conformity with EN 81-1, is excessive if some deflection of the structure and guide brackets is taken into account. Clearly this needs to be established and the contribution of the guide deflection reduced to keep the overall deflection within 5 mm. Since the geometry of the arrangement is balanced, this would require measures such as reduced distance between guide fixings, larger guide section or a switch to a safety gear with a lower impact factor i.e. to progressive safety gear.

3.2 Sample calculations - cantilever guided configuration as Table 2

The evaluation of forces, stresses and deflection for the cantilever guided arrangement is with the same equations as for the conventionally guided configuration. Only the vertical forces remain the same; the horizontal forces are significantly different and are evaluated as follows.

As noted, the parameters in equations (1) are unchanged so $F_v = 36611$ N.

Using equation (2) with the smaller distance between guide fixings: $\lambda = \frac{l_k}{i}$ = 2500/23.61 = 106

Similarly to before from 5.10.3 of EN 81-50 but with $85 < \lambda \leq 115$, $\omega = 0.00001711\lambda^{2.35} + 1.04 = 2.02$

Similarly to equation (3), σ_k = (36611x2.02)/2274 = 33 N/mm^2

For the calculation of bending loads, there will be much larger values for F_x than for the conventionally guided configuration (since both the empty car weight and car loads are offset significantly from the guide rails). Using equations (5) to (8) and for the two loading cases, the bending loads and stresses are as folows.

For case 1 relative to the x-axis, $x_q = c+5D_x/8 = 888$ mm, $x_p = 500$ mm; $y_q = 0$ and $y_p = 0$. So $F_{x(1)}$ and $F_{y(1)}$ can be evaluated using these and equations (5) and (6):

$$F_{x(1)} = \frac{3xg_n(1000x888+1250x500)}{2x2800} = 7951\text{ N and } \sigma_{y(1)} = \frac{3x7951x2500}{16x23610} = 158\text{ N/mm}^2$$

$F_{y(1)} = 0$ and $\sigma_{x(1)} = 0$

For case 2 relative to the y-axis, $x_q = c+D_x/2 = 750$ mm, $x_p = 500$ mm; $y_q = D_y/8 = 263$ mm so $F_{x(2)}$ and $F_{y(2)}$ can be evaluated:

$$F_{x(2)} = \frac{3xg_n(1000x750+1250x500)}{2x2800} = 7226\text{ N and } \sigma_{y(2)} = \frac{3x7226x2500}{16x23610} = 143\text{ N/mm}^2$$

$$F_{y(2)} = \frac{3xg_n(1000x263)}{2800} = 2764\text{ N and } \sigma_{x(2)} = \frac{3x2764x2500}{16x30650} = 42\text{ N/mm}^2$$

The combined bending stress, σ_m, has its worst case value for case 2 where σ_m = 185 N/mm^2

The combined bending and compressive stress, from (10) is: $\sigma = 185 + \frac{(36611)}{2274}$ = 201 N/mm^2

The combined bending and buckling stress is: $\sigma = 33 + 0.9x185 = 200$ N/mm^2

All of these combined stresses are close to, but within, the permitted stress of 205 N/mm^2. From a conformity perspective, these are acceptable but might need to be reviewed in an engineering context. For instance, the assumptions made which underly the calculations should be reviewed to make sure they are robust and can be controlled to be within the parameters used. From a practical perspective, any adverse variation in the distance between guide fixings (which can not always be so tightly controlled on site) would be likely to push guide stresses outside the permitted stress levels.

At this point, we should calculate the flange bending stress since the value of F_x is high (again from a strict conformity perspective, this should be done for all cases). EN 81-50, 5.10.5 gives two equations depending on the use of roller guide shoes (concentrated load) or sliding guide shoes:

$$\sigma_F = \frac{1.85F_x}{c^2} \text{ for roller guide shoes} \quad (14)$$

$$\sigma_F = \frac{6F_x(h_1-b-f)}{c^2(l+2(h_1-f))} \text{ for sliding guide shoes} \quad (15)$$

The dimensions introduced here are for the guide rail section: c is the thickness of the neck connecting the blade and the foot (not as dimension c in Table 2); h_1 is the guide rail height and f is foot depth of the where it connects to the blade. For T127-1/B guide rails, these dimensions are 10 mm, 89 mm and 11 mm respectively. b is half the width of the guide shoe lining and l is the length so depend on the type selected – we will assume 19 mm and 140 mm respectively.

Evaluating the flange bending stress using (14) and (15) with the worst case value for F_x:

$$\sigma_F = \frac{1.85x7951}{10^2} = 147 \text{ N/mm}^2 \text{ for roller guide shoes which is less than } 205 \text{ N/mm}^2.$$

$$\sigma_F = \frac{6x7951(89-19-11)}{10^2(140+2(89-11))} = 96 \text{ N/mm}^2 \text{ for sliding guide shoes also less than } 205 \text{ N/mm}^2.$$

Turning to the worst case deflections, these can be evaluated using (12) and (13) as:

$$\delta_x = 0.7\frac{7951x2500^3}{48x207000x1499000} + \delta_{str-x} = 5.8 \text{ mm} + \delta_{str-x}$$

$$\delta_y = 0.7\frac{2764x2500^3}{48x207000x1879000} + \delta_{str-y} = 1.6 \text{ mm} + \delta_{str-y}$$

So the deflection in the x-axis, irrespective of any allowance for the deflection of building structure and guide rails, is clearly excessive. In seeking to reduce this to acceptable limits, we can note that there are three alternatives:

1. Increase the guide section; this is likely to be be expensive and there might be implications for incorporating a larger guide section into the design;
2. Reduce the worst case value of F_x; as we saw above, this could be accomplished with a progressive safety gear reducing the value of k_1 to 2;
3. Reduce the distance between guide fixings, l. We can note that, because the deflection depends on l^3, a modest reduction in this dimension would bring about a significant reduction in deflection.

3.3 Further comment

A final observation on the two cases examined here is that in both cases, guide rail deflections have determined the design solution used. It is quite straightforward to rearrange (12) and (13) to arrive at equations for the minimum required second moments of area for the guide rail in a given design and hence make at least a first selection of a suitable guide rail for a given distance bewteen fixings.

4. ROPE TRACTION CALCULATIONS

EN 81-20, 5.5.3 [3] has normative requirements generally as those in EN 81-1 with a new possibility, in addition to rope slipping, of using an electric safety device to stop the machine to avoiding raising an empty car or counterweight. A note references calculation examples in 5.11 of EN 81-50; so their use is not a normative requirement of EN 81-20.

EN 81-50 clause 5.11 generally follows Annex M of EN 81-1 except for:

- need to lose traction for stalled car only where machine torque is sufficient to raise the car;
- emergency braking for reduced stroke buffers – acceleration rate to be sufficient to retard car and counterweight to speed for which buffers designed (EN 81-1 had 0.8 m/s/s).
- car and counterweight stalled for empty car at highest and lowest position (EN 81-1 based on worse case).

Equations for calculating applied traction ratios have some changes:

- correctly including successive rope falls after the first;
- correcting treatment of diverter and reeving pulleys;
- split into machine above and machine below;
- guidance including, if minimum friction forces cannot be ensured, deleting those terms.

Informative Annex D provides an example with simplified equations for that case.

4.1 Traction inequalities

The traction inequalities in 5.11.2 of EN 81-50 are as follows where T_1 and T_2 are the rope tensions on either side of the traction sheave and α is the angle of wrap around the traction sheave:

$\frac{T_1}{T_2} \leq e^{f\alpha}$ for car loading and emergency braking. (16)

$\frac{T_1}{T_2} \geq e^{f\alpha}$ for car/ counterweight stalled. (17)

The remainder of these sample calculations concentrate on what is often the worst case; satisfying the first inequality for emergency braking where there is a trade-off between roping and traction. This is not to lessen the importance of the car loading criteria or the second inequality but this can be readily calculated using the higher value for the coefficient of friction from EN 81-50, 5.11.2.2.2.

The first traction inequality has two sides; the first is the calculation of the applied traction ratio which depends on the suspended masses while the second is the calculation of the critical traction ratio which depends on the groove profile.

4.2 Example calculations – critical traction ratio

EN 81-50, 5.11.2.3 provides the equations to determine the friction factor, *f*, for the groove profile details in Table 1:

$f = \mu \frac{4\left(1-sin\frac{\beta}{2}\right)}{\pi-\beta-sin\beta}$ for unhardened undercut-V groove (18)

$f = \mu \frac{1}{sin\frac{\gamma}{2}}$ for hardened V grooves (19)

The coefficient of friction for the emergency braking in EN 81-50 5.11.2.3.2 is dependent on the rope speed, *v*:

$\mu = \frac{0.1}{1+\frac{v}{10}}$. (20)

The value of μ can be readily calculated to be 0.083. Then from (18), *f* can be calculated to be 0.2 for an unhardened undercut-V groove with 105° undercut and from (19) also 0.2 for a hardened V groove with 50° angle. The value of the critical traction ratio can then be calculated from (16) as 1.87 for the emergency braking case. Note that the selection of groove parameters provides a similar critical traction ratio for either an unhardened groove with undercut V or for hardened V grooves.

Although a discussion of the coefficient of friction is outside the scope of this paper, the figures used in EN 81 are intended to be worst case and reflect those measured from oiled rope in traction sheave groves. In normal operation, significantly higher values would be expected so the figures used in EN 81 incorporate some margin of safety.

4.3 Example calculations – applied traction ratio

At first sight the equations in 5.11.3 of EN 81-50 for the applied traction ratio look complex. However, they are for the general case so include for multiple reeving pulleys (m_{Pcar} and m_{Pcwt} are the reduced mass of pulleys on car and counterweight side respectively), all positions of the car in the well, the use of compensation, friction from the guide shoes etc. The following example follows the EN 81-50 Annex D equations for the emergency braking condition.

For the car with full load at the lowest landing:

$$\frac{T_1}{T_2} = \frac{(P+Q)(g_n+a)+2M_{SRcar}(g_n+2a)+2m_{Pcar}a-FR_{car}}{(P+BQ)(g_n-a)-m_{Pcwt}a+FR_{cwt}}. \quad (21)$$

For the empty car at the highest landing:

$$\frac{T_1}{T_2} = \frac{(P+BQ)(g_n+a)+2M_{SRcwt}(g_n+2a)+m_{Pcwt}a-FR_{cwt}}{(P+M_{Trav})(g_n-a)-2m_{Pcar}a+FR_{cwt}}.$$
(22)

These can be evaluated with the parameters in Table 1 for the conventionally guided situation. Here, since the suspension is coincident with the centres of gravity of the empty car and load (and assumed also for the counterweight), minimum values for FR_{car} and FR_{cwt} cannot be ensured as required in EN 81-50, 5.11.3 and so these are set at zero.

So for the full car at the lowest landing: $\frac{T_1}{T_2} = \frac{(1250+1000)(9.81+0.5)+2x25x(9.81+1)+2x30x0.5}{(1250+0.45x1000)(9.81-0.5)-30x0.5}$=1.50.

For the empty car at the highest landing: $\frac{T_1}{T_2} = \frac{(1250+450x1000)(9.81+0.5)+2x25x(9.81+1)+30x0.5}{(1250+12)(9.81-0.5)-2x30x0.5}$=1.54.

In this example, the influence of the pulleys is almost negligible and clearly the applied traction ratios are very much within the critical traction ratio calculated.

In the case of the cantilever guided configuration, the parameters used to calculate the applied traction ratios would be identical except that, if sliding guide shoes are used, it is reasonable to argue that, there will always be a frictional force on the car side. In this situation, the worst case is the lowest force so the value of FR_{car} should reflect the minimum steady load on a guide i.e. with empty car:

$$F_{min} = \frac{g_n(1250x500)}{2x2800}. \quad (23)$$

We can evaluate FR_{car} for this situation, taken on 4 guide shoes which have coefficient of sliding friction μ_g:

$$FR_{car} = 4\mu_g F_{min}. \quad (24)$$

Using a worst case (minimum value) for guide shoe coefficient friction of 0.05, this gives:

FR_{car} = 4x0.05x1095 = 219 N

In calculating the likely friction force resisting the normal operation, say for the selection of a machine with sufficient torque, higher values should be used reflecting higher guide forces at full load and reflecting more realistic values of friction normally expected.

Re-evaluating (21) and (22) as above except with this value for FR_{car} leads to:

So for the full car at the lowest landing: $\frac{T_1}{T_2} = \frac{(1250+1000)(9.81+0.5)+2x25x(9.81+1)+2x30x0.5-219}{(1250+0.45x1000)(9.81-0.5)-30x0.5}$=1.49;

and empty car at the highest landing: $\frac{T_1}{T_2} = \frac{(1250+450x1000)(9.81+0.5)+2x25x(9.81+1)+30x0.5-219}{(1250+12)(9.81-0.5)-2x30x0.5}$=1.52.

Clearly all these values for applied traction ratios under emergency braking are comfortably within the critical traction ration calculated earlier. It can be observed that the use of friction forces of this size on a cantilever guided arrangement have not reduced the applied traction ratios significantly.

4.4 Further comment

In this example the applied traction ratios are determined largely by the main lift masses (although this may not be the case in other situations). In such cases, the use of simpler equations might be an alternative to the method in EN 81-50 (recalling that following these is not a normative requirement of EN 81-20). In place of the evaluation of all the various elements, some could be omitted and a suitable margin between applied and critical traction ratios used to take account of the neglected factors (and perhaps also to account for errors in the setting or measurement of parameters such as the empty car weight and counterweight balance). The equations in the Clause 9 notes of the previous EN 81-1: 1985 [6] can be seen to be such a simplification. Here there was a coefficient, C_2, introduced to cope with the wear of V grooves which has been superseded by the more detailed approach to calculating the critical traction ratio of these.

5. ROPE FACTOR OF SAFETY CALCULATIONS

EN 81-20 [1], 5.5.2.2 has requirements similar to those of EN 81-1 with a normative reference to clause 5.12 of EN 81-50. EN 81-50, 5.12 includes the calculation of an additional minimum factor of safety for ropes on traction lifts. This is generally as EN 81-1 Annex N with main changes being:

- Increased values for $N_{equiv(t)}$ for V grooves of 36° to 45°
- New value of $N_{equiv(t)}$ for V grooves of 50°
- The row for $N_{equiv(t)}$ previously for undercut-U and –V grooves is now for undercut-U.
- More definition on what is a reverse bend – rope distance between fixed pulleys less than 200x rope diameter and the bending planes are rotated through more than 120°.

5.1 Determining the number of equivalent pulleys

EN 81-50 Annex E has examples to assist with determining the number of equivalent pulleys, N_{equiv}. In this example, the worst case section of ropes will be where the traction sheave and two car pulleys run (there is no section of the ropes over which the traction sheave, car pulleys and counterweight pulley runs). Then the number of equivalent pulleys, N_{equiv}, is:

$$N_{equiv} = N_{equiv(t)} + N_{equiv(p)}. \tag{25}$$

The equivalent number of deflection pulleys, $N_{equiv(p)}$, considers the number of pulleys with simple bends N_{ps}, the number of pulleys with reversed bends N_{pr}, and the ratio between the traction sheave diameter, D_t, and the pulley diameter, D_p:

$$N_{equiv(p)} = \left(\frac{D_t}{D_p}\right)^4 \left(N_{ps} + 4N_{pr}\right). \tag{26}$$

Since there are no reversed bends and two simple bends (car pulleys) then we can evaluate this as:

$$N_{equiv(p)} = \left(\frac{320}{320}\right)^4 (2 + 4x0) = 2$$

The equivalent number of pulleys for the traction sheave, $N_{equiv(t)}$, is found from Table 2 in 5.12.2.1 of EN 81-50. In the example above, this is 5 for a hardened V groove of 50° angle and so $N_{equiv} = 7$.

5.2 Example minimum safety factor calculation

It is now possible to evaluate the minimum value of safety factor from EN 81-50, 5.12.3:

$$S_f = 10^{\left(2.6834 - \frac{\log\left(\frac{695.85x10^6 N_{equiv}}{\left(\frac{D_t}{D_r}\right)^{8.567}}\right)}{\log\left(77.09\left(\frac{D_t}{d_r}\right)^{-2.894}\right)}\right)}. \quad (27)$$

Using $D_t/D_r = 40$ and $N_{equiv} = 7$, the minimum required safety factor, S_f, can be evaluated to be 16.

If 8 mm ropes are selected of 43 kN minimum breaking load, then it is straightforward to determine that 5 ropes attain the necessary minimum safety factor with a safety factor of 19.

5.3 Influence of groove type and parameters, groove pressure

As a comparison, and to illustrate the potential impact of the changes to $N_{equiv(t)}$ for V grooves in Table 2 from those in EN 81-1, we can evaluate S_f for a V groove of 45° (where the value has changed the most) using the values of $N_{equiv(t)}$ from EN 81-1 (4.0) and from EN 81-20 (6.5). Then the minimum safety factor for EN 81-1 would be 15.5 and for EN 81-20 would be 17.6; a relatively modest improvement.

The calculation for S_f allows for the selection of ropes to meet the EN 81-20 standard which as noted earlier is a normative requirement of the standard. This method allows the selection of roping to take account of the nature of traction sheave grooves and pulleys reducing the lifetime of steel wire ropes. Further consideration of the pressure of the ropes in the traction sheave grooves is not a part of the safety standard but would usually be carried out as part of selecting and coordinating the wire rope and traction sheave hardness.

In this case, with hardened V grooves, the pressure in the grooves is of the order of 9.3 N/mm^2. While this would be high for conventional sheave materials, depending on the selection of sheave material and wire rope tensile strength, it could be considered acceptable.

As a comparison, if the groove were to be treated as an unhardened undercut U ($N_{equiv(t)} = 15.2$) then the minimum required safety factor would be 23. This would require at least one more rope and would therefore reduce the groove pressure accordingly.

It was noted above from the evaluation of critical traction ratios that unhardened V grooves with 105° undercut have similar calculated critical traction ratios as hardened 50° V grooves. So grooves of equivalent traction are not equivalent in terms of making rope factor of safety calculations. From an engineering perspective, selecting a hardened V groove, with the smaller $N_{equiv(t)}$, by setting a lower minimum factor of safety, allows fewer ropes than if an unhardened groove were selected.

6. RAMS, CYLINDERS, RIGID PIPES AND FITTINGS CALCULATIONS

EN 81-20 makes normative references to 5.13 of EN 81-50 for calculations for pressure and buckling of the jack, from 5.9.3.2, and for pressure of rigid pipes and fittings, from 5.9.3.3.2. EN 81-50 clause 5.13 is generally as EN 81-2 Annex K with the main changes being:

- Calculation for wall thickness in 5.13.1.1 now correctly uses the internal diameter (so wall thicknesses calculated to EN 81-2 would be slightly thicker).
- Errors corrected in 5.13.1.2.4 for flat bases with welded flange.
- Error corrected to buckling calculation of telescopic jack without guidance yoke.

On the basis that there are no significant changes to these calculations, they are not considered further here.

7. UNDERLYING ASSUMPTIONS – USUAL ENGINEERING PRACTICE

There are many assumptions made in the writing of a standard; in the case of EN 81-20, many are stated explicitly in clause 0.4. Included in the assumptions at 0.4.3 and at 0.4.12 are:

> "Components are…designed in accordance with *usual engineering practice* and calculation codes taking into account all failure modes;" and that:
>
> "a mechanical device built according to *good practice* and the requirements of the standard….will not deteriorate to a point of creating hazard without the possibility of detection provided that all of the instructions given by the manufacturer have been duly applied…".

Further guidance on the importance of assumptions and some of the concepts included in the assumptions is available in CEN/TR 81-12 [7]. Although the scope includes: "This Technical Report gives guidance to users, specifically outside Europe….", it is of more general interest. The technical report (it is not a standard and does not contain normative requirements) provides some helpful guidance on:

- the use of words such as "shall", "should", "may" and "can" in standards;
- guidance on notes and annexes including the difference between "informative" and "normative";
- more guidance on the assumptions and how these could be applied in different territories;
- references to EN standards; and
- specific national requirements.

In particular at 5.7, it discusses good engineering practice and elaborates important roles for the designer. Included in these is the use of calculations where CEN/TR 81-12 makes some important points in the context of the example calculations made above.

- For every calculation, all probable load cases need to be defined. It may be the case that a factor is not included in the equations and method in EN 81-50 and that additional factors need to be included. Clearly the designer should take account of these.
- When using calculation methods, consideration should be given to the inclusion of inherent simplifications and error factors. In the context of using simplified equations for calculation applied traction ratios, this would imply the use of a factor to take account of these simplifications such as was discussed in section 4.4.

The final point made is that good engineering practice entails subsequent design review by peer(s) or expert(s) in the appropriate discipline. This discussion in CEN/TR 81-12 therefore very neatly frames the context for the issues discussed in this paper.

8. CONCLUSION

The example calculations have been made using a set of lift parameters which are quite unremarkable. For this example design, car guide calculations were made for both conventionally guided and cantilever guided configurations, critical and applied traction ratios and rope minimum safety factor. These helped to highlight some changes between the approaches taken in EN 81-1 and EN 81-50.

For guide rail calculations, EN 81-20 has a normative requirement to evaluate the vertical loads on guide rails and a new requirement to consider deflection of building structure. For the calculations in EN 81-50, which are one method to satisfy the normative requirements of EN 81-20, the most significant change is the inclusion of the deflection in the building structure and guide brackets into the deflection calculations.

Example calculations illustrated some important differences between conventionally guided and cantilever guided configurations. The worst case selected was for safety gear operation showed both stress levels approaching the permitted stress level and deflections close to or greater the permitted levels.

The engineering implications of reducing guide rail deflections was considered where it was noted that reducing the distance between fixings is very much more effective in reducing deflections than either increasing guide rail size or reducing loads on the guides.

For traction calculations in EN 81-50, which were seen to be referenced informatively from EN 81-20, the calculation of critical traction ratios was unchanged from the EN 81-1. Two groove parameters were selected reflecting similar critical tractions ratios.

The emergency braking situation was calculated. On the simple lift model considered, the influence of omitting or including elements such as pulley masses and guide rail friction was considered. The conclusion was that simplified calculation methods might be used on simplified designs if these included a suitable factor or margin to take account of the parameters neglected or not included.

For the minimum safety factor for ropes on traction lifts in EN 81-50, which is normatively referenced from EN 81-20, the model lift allowed a simplified evaluation of the number of equivalent pulleys and a calculation of the minimum safety factor. This was calculated for both groove profiles to illustrate significant differences in these for different groove profiles of equivalent traction.

Calculations were also made to illustrate the relatively modest influence on safety factor from the changes in the table for V grooves from EN 81-1 to EN 81-50.

The discussion closed with a brief review of some useful guidance in CEN/TR 81-12 on good engineering practice and, to close the loop, on some guidance on making calculations to support the design.

Closing remarks: the calculations presented necessarily are not comprehensive or exhaustive. Where calculations are being made to demonstrate conformity with EN 81-20 then these should be comprehensive i.e. all cases calculated and all relevant factors taken in consideration.

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BIOGRAPHICAL DETAILS

Nick Mellor has worked for the UK's Lift and Escalator Industry Association (LEIA) as Technical Director since January 2012 and has been in the industry for 22 years. Nick was in the inaugural cohort of the MSc in Lift Engineering at Northampton. More recently, as an Associate Lecturer, he has done some tutoring on the MSc. The idea for this paper came from a chance remark from the technical manager of a UK lift company earlier this year. Hopefully the paper is of some interest to both those in the industry working with EN 81-50 and those studying lift engineering.