Some Thoughts on Rope Life

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Abstract. This document explores the issues that affect the working lifetime of the ropes used with electric traction lifts and considers how "best practice" has changed over the years by using the modernisation of a common type of lift as an example.

1 INTRODUCTION

In the 1970's engineers were taught the rope life for an electric traction lift depended on traction, groove pressure and rope drag (i.e. fleet angle).

Although modern methods of traction assessment where both the static and dynamic forces are considered are an improvement over the old; groove pressure has fallen out of fashion having been superseded by the mandatory requirements for the minimum rope safety factor in EN81-1 [1] and EN81-20 [2] but the effect of rope drag or fleet angle is mostly overlooked today.

Why does this matter? The fleet angle can have a deleterious impact on rope life; the old engineers considered it to have a worse effect than excessive groove pressure. Experts on ropes acknowledge the part it plays by including a correction factor for fleet angle in rope life calculations; but although recommendations for maximum limits were included in BS5655-6:1990 [3], it is conspicuous by its absence in current standards.

Patrick Ryan's paper, presented at the 2015 lift symposium in Northampton [4] indicated issues with modern machine room less (MRL) lifts having inadequate rope life, despite the fact the requirements of that complex equation in Annex N of EN 81-1having been met.

This paper explores how the requirements of British Standards have developed over the last 40 years and discusses whether looking backwards to past best practice may help resolve this.

2 ROPE SELECTION THROUGH THE AGES

2.1 General Observations

When choosing the correct rope for an application the following factors need to be considered by the designer:

- The number of pulleys in the system and the roping ratio
- The ratio between the rope diameter and sheave diameter
- The material of the sheave and its hardness
- The groove form of the driving sheave
- The construction of the rope
- The minimum safety factor permitted by code
- The usage, i.e. the likely number of trips in a day

All the above factors are considered when checking there will be sufficient traction without excessive groove pressure, but the fleet angle needs to be considered separately. It can be said therefore that rope selection will depend on satisfactory traction, groove pressure, and fleet angle.

2.2 In the 1970s

It must be remembered that at this time the lift industry in the UK was very different to how it is today. Lift manufacturers generally designed, made, installed and serviced their own equipment; the independent sector did not yet exist. It was normal for a building owner to enter into a 25-year comprehensive maintenance agreement with the manufacturer of their lifts, and it was in the interest of the lift manufacturer to ensure their components were designed to give a long life to maximise their long-term profits. If the traction and groove pressure, (which had been calculated in the same way for many years), was close to the limits dictated by experience, the cost of replacement sheaves and additional ropes were factored into the maintenance costs, and so there are no specific requirements in BS 2655 [5] or the code of practice CP 407 [6] regarding traction or groove pressure because it wasn't perceived to be an issue.

As BS 2655 and CP 407 may be unfamiliar to most people under the age of 60, their requirements regarding rope related matters are summarised in Table 1 below.

Table 1 Requirements for Ropes in the 1970s

Item	Description	Source	Remarks
Minimum rope safety factor	10:1 ≤ 2.0 [m/s] rated speed 11:1 ≤ 3.5 [m/s] rated speed 12:1 ≤ 7.0 [m/s] rated speed	BS 2655-1:1970 clause 2.14.2	Note this is speed dependant.
Permitted rope terminations	Spliced or gripped return loops with thimbles or metalled sockets.	BS 2655-1:1970 clause 2.14.2	
Minimum sheave and pulley diameter	d (44 + 3S) with a minimum of 47 for 6 × 19 (9/9/1) construction ropes <u>or</u> d (37 + 3S) with a minimum of 40 for 6 × 19 (12/6 + 6 F/1) or 8 × 19 (9/9/1) construction ropes Where: d = rope diameter S = rope speed = rated speed x roping ratio [m/s]	BS 2655-1:1970 clause 2.14.4.2	Note limited rope types and speed dependant.
Single wrap vs double wrap	A 2:1 roped double wrap system with the rope to sheave ratio increased to 10% above the minimum recommended by BS 2655 will give a similar rope life to a 1:1 single wrap system	CP 407:1972 Clause 2.6.2.4	
Reverse bends	Increase the minimum diameters of the slower speed pulleys by 10% in all cases where the rope speed over such pulleys is more than 0.5 [m/s]	CP 407:1972 Clause 2.6.2.5	
Multiplying pulleys	For 2:1 roped lifts with rated speeds above 1.0 [m/s] only one pulley should be on the car and one on the counterweight	CP 407:1972 Clause 2.6.2.6	
Rope drag	Where the distance between two pulleys or a pulley and a sheave is fixed, the minimum drag ratio should be 100:1. Where the drag is between two points so that the distance between the two points and therefore the drag ratio varies as the car travels then the minimum drag ratio should be 41:1 when the car or counterweight rests on a completely compressed buffer.	CP 407:1972 Clause 2.6.2.8	The rope drag ratios are equivalent to maximum fleet angles of 0.6° between fixed pulleys and 1.4° between a fixed point and a moving point.

Other points to note:

1. Sheaves were generally made from cast iron and had a Brinell hardness in the region of 200-250, i.e. grooves were not hardened.
2. Rope anchorage plates were designed to keep the distances between the anchorages to a minimum to increase the rope drag ratio (i.e. minimise the fleet angle). “Long and short” eyebolts especially with the 2:1 roping anchorages were common and allowed an even more compact arrangement.

2.3 In the 1980s and early 1990s

In 1979 EN 81-1 was published in the UK as BS 5655-1 and included several national variations. The standard went through several amendments in the early 1980s, the “definitive” version which will be considered by this paper was published in 1986 [7].

The old code of practice CP 407 was replaced by BS 5655-6:1985.

Following some controversial remarks about the state of modern architecture by the Prince of Wales, and the planning authorities tightening up on interruptions to the skyline, designers wanted to avoid placing lift machine rooms on the top of their buildings, leading to the rising popularity of underslung lifts with the machine room located in a basement if you were lucky or at the top floor at the rear or side of the lift well if you were unlucky. As result of the experience gained by the industry in the UK during the 1980s, BS 5655-6:1990 [3] included a clause intended to reduce the permitted fleet angle between fixed pulleys to 0.4° (equivalent to 143:1 rope drag). But, due to an unfortunate typographical error, a figure of 4° was stated in the standard which has been adopted into common lift culture despite the clause being omitted from later issues of the standards.

The requirements of these standards are summarised in Table 2 below.

Table 2 Requirements for Ropes in the 1980s and early 1990s

Item	Description	Source	Remarks
Rope Specification	8 mm minimum diameter, wires to have a minimum tensile strength, characteristics to be as specified in international standards.	BS 5655-1:1986 clause 9.1.2	New requirement
Minimum rope safety factor	12 for systems with 3 or more ropes, 16 for those with two ropes A very high factor is not recommended since insufficient loading on a rope may reduce rope life.	BS 5655-1:1986 clause 9.9.2 BS 5655-6:1990 clause 4.4.1.2	No longer speed dependent. Note the comment on high safety factors!
Permitted rope terminations	Must withstand at least 80% of the breaking load of the rope. Spliced return loops with thimbles, gripped return loops with thimbles and at least 3 grips, metalled or resin sockets, self-tightening wedge sockets, ferrules or any system with equivalent safety.	BS 5655-1:1986 clause 9.2.3 clause 9.2.3.1	New strength requirement, more types of terminations now permitted
Minimum sheave and pulley diameter	40 x the rope diameter In some cases, it may be advantageous to increase this ratio to extend rope life.	BS 5655-1:1986 clause 9.2.1 BS 5655-6:1990 clause 4.4.1.3	No longer dependant on the rope construction or speed
Traction and groove pressure	Formulae are given for traction, limits specified for groove pressure.	BS 5655-1:1986 clause 9.3	New requirement
Rope tensioning devices	Must be fitted at one end at least, if a spring it must be in compression, slack rope switches to be fitted on systems with only two ropes.	BS 5655-1:1986 clause 9.5	New requirement

<table>
<tr><th>Item</th><th>Description</th><th>Source</th><th>Remarks</th></tr>
<tr><td>Single wrap vs double wrap</td><td></td><td></td><td>Withdrawn</td></tr>
<tr><td>Reverse bends</td><td>The minimum diameter of the pulleys should be increased by at least 10 % when the rope speed is greater than 0.5 [m/s].</td><td>BS 5655-6:1990 clause 4.4.1.6</td><td>Like CP 407:1972 clause 2.6.2.5</td></tr>
<tr><td>Multiplying pulleys</td><td>The more pulleys introduced into a roping system, the greater will be the rope wear.</td><td>BS 5655-6:1990 clause 4.4.1.4</td><td>No longer limits on the numbers of pulleys</td></tr>
<tr><td>Rope drag</td><td>Where the distance between two pulleys/sheaves is fixed, the fleet angle of the ropes in relation to the grooves should not exceed 0.4° (4° sic) either side of the groove axis. Where the distance between the two points varies as the car travels, the fleet angle should not exceed 1.4° when the car or counterweight is on a compressed buffer.</td><td>BS 5655-6:1990 clause 4.4.1.7</td><td>Conversion table
<table>
<tr><th>Fleet Angle</th><th>Rope Drag Ratio</th><th></th></tr>
<tr><td>0.6°</td><td>100:1</td><td>CP 407</td></tr>
<tr><td>4°</td><td>14:1</td><td>Typo!</td></tr>
<tr><td>0.4°</td><td>143:1</td><td>Correct</td></tr>
<tr><td>1.4°</td><td>41:1</td><td>Same</td></tr>
</table>
Increasing the fleet angle from 0.6° to 4° makes no sense!</td></tr>
<tr><td>Machine layouts</td><td>Machine above arrangements preferred, others require more pulleys and lead to greater rope wear.</td><td>BS 5655-6:1990 clause 4.4.1.8</td><td></td></tr>
</table>

Other points to note:

1. The independent sector started to take off in the 1980s; the harmonisation of British Standards with European Standards allowed for the importation of components and package lifts from suppliers in other parts of Europe.
2. It was discovered through bitter experience that hardened grooves on sheaves may require a different rope construction to the “standard” 6 or 8 strand Seale (9/9/1).

2.4 Late 1990s to Present Day

EN 81-1 [1] underwent a major revision in the late 1990s and has recently been superseded by EN 81-20 [2] and EN 81-50 [8]. These standards will be familiar to most within the industry so this section will briefly summarise the major changes from BS5655-1:1986 (EN 81-1:1985) regarding ropes:

- End terminations: spliced return loops with thimbles, gripped return loops with thimbles and at least 3 grips, metalled or resin sockets are no longer permitted, only self-tightening wedge sockets, ferrules or swaged terminations may be used.
- The method of calculating traction has changed.
- Groove pressure is no longer considered, replaced by a very complicated mandatory equation that gives a minimum permissible rope safety factor (EN 81-1:1998, Annex N). Note if this safety factor is less than 12 (three or more ropes) or 16 (two ropes) the higher figure should be used.

BS 5655-6 [9] only recommends the following it does not impose any restrictions:

- Machines should be located above if possible.
- The number of pulleys used in the system and the number of reverse bends should be minimised.
- Careful consideration should be given to the effect of the number of pulleys, the number of reversed bends, and the fleet angles of the ropes on and off the sheave or pulley.

3 WHERE HAS IT ALL GONE WRONG?

3.1 Some Anecdotal Evidence

Many people in the industry considered dropping the requirements for groove pressure from the standards to be a step backwards. Some still take it into consideration, but many do not.

One heard whispers from the mid-2000s onwards that ropes were not lasting as long as they should do, with the rope and machine manufacturers getting the blame in many cases.

One company the author worked for believed the Annex N equation only allowed a minimum rope life of three years and the safety factor should be increased.

Some of the technical people at Brugg wrote an article in Elevator World to set the record straight about the quality of modern ropes [10] and concluded that poor rope life was due to a combination of the following factors:

- High usage
- Small ratios between the rope diameter and sheave diameter
- Uneven load distribution between the ropes
- High acceleration and deceleration rates
- Poor quality sheaves
- Poor installation
- Poor maintenance

All valid points, but note fleet angle does not make the list.

3.2 It All Goes Back to Feyrer

According to Feyrer [11], the equation used to calculate the number of bending cycles is based on the following assumptions:

- The rope is well lubricated
- There is no side deflection (i.e. fleet angle)
- The grooves are steel (i.e. not lined with plastic) and the radius = 0.53 x the rope diameter
- The ropes are not twisted

If the assumptions made above do not hold true the number of bending cycles calculated in equation 3.55, it is corrected by multiplying by four endurance factors f_{N1}, f_{N2}, f_{N3} and f_{N4} which are shown in Table 3.

Table 3 Endurance Factors (Feyrer)

<table>
<tr><th>Endurance Factor</th><th>Description</th><th>Value</th></tr>
<tr><td>f_{N1}</td><td>Lubrication – assume the rope is well lubricated</td><td>1.0</td></tr>
<tr><td>f_{N2}</td><td>Fleet angle ϑ [degrees]</td><td>$$1 - \left(0.00863 + 0.00243\frac{D}{d}\right)\vartheta - 0.00103\vartheta^2 \quad (1)$$ Where D = sheave/pulley diameter, d = rope diameter</td></tr>
<tr><td>f_{N3}</td><td>Groove form</td><td>Assuming an undercut U or V groove form:<table><tr><td>Undercut β</td><td>75°</td><td>80°</td><td>85°</td><td>90°</td><td>95°</td><td>100°</td><td>105°</td></tr><tr><td>f_{N3}</td><td>0.40</td><td>0.33</td><td>0.26</td><td>0.20</td><td>0.15</td><td>0.10</td><td>0.066</td></tr></table>(refer to Feyrer for other groove forms)</td></tr>
<tr><td>f_{N4}</td><td>Twisted ropes - assume the rope twist is negligible</td><td>1.0</td></tr>
</table>

Where lifts are concerned, the only one of these factors that are generally considered is the one for the groove form. Indeed, the value of the equivalent number of traction sheaves ($N_{equiv(t)}$) in the evaluation of the minimum rope safety factor is groove form dependent. Despite the lessons of experience, it must be assumed the ropes will be properly lubricated, installed without twist and fitted with anti-twist lanyards (although this is by no means certain given the diminishing skills of site personnel), but what about the fleet angle?

3.3 What About the Fleet Angle?

Figure 1 shows the results of using eq. 1 to calculate the endurance factor for the fleet angle depending on the rope to sheave ratio. The maximum fleet angles recommended by CP 407 and BS 5655-6:1990 are indicated for easy reference. The chart only goes up to 4°, as this appears to be the general consensus for the maximum limit.

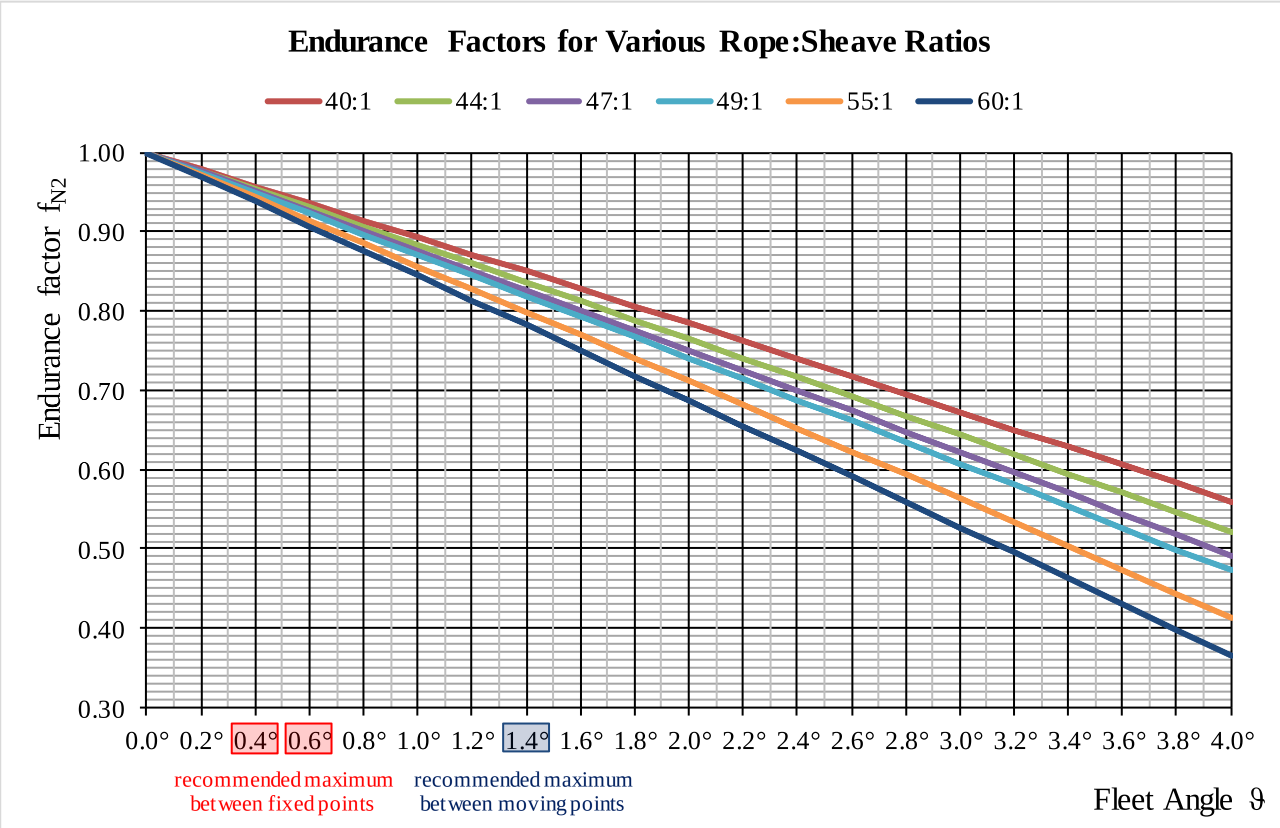

Figure 1

It is apparent that the endurance factor not only reduces as fleet angle increases but also as the rope to sheave ratios increase. To put this into context, assuming as a worst case a fleet angle of 4°, a 10mm diameter rope running over a 600mm diameter sheave would have an endurance factor approximately 35% smaller than the same rope running over a 400mm diameter sheave.

If the fleet angle is limited to the 1.4°, the CP 407 and BS 5655-6:1990 recommendation for the maximum between a fixed pulley and a moving pulley or anchorage, the endurance factor reduces by approximately 8%; and 2% if the fleet angle is reduced to the 0.4°, the figure BS 5655-6:1990 would have recommended between fixed pulleys if the typographical error hadn't occurred.

Although the fleet angle does not have as deleterious an effect on rope life as the groove form of the sheave, it will have some effect and this could be substantial depending on the circumstances.

3.4 But What Does It All Mean? – An Example

As all the standards and codes of practice recommend 1:1 roped gear above systems to maximise the rope life it has been decided to use the modernisation of a typical standard lift of this type installed in the early 1990s as an example that many people will be familiar with from their own experience. It is based on a standard “Omega” made by Express Lifts. The building is an office block, the details are summarised below:

Rated Load	:	630 kg
Rated Speed	:	1.6 m/s
Car weight	:	950 kg
Balance	:	50%
Travel	:	40 m
Number of floors served	:	12

The lift was fully compliant with the codes and standards of its day and has 6 x 11 mm diameter ropes of 8x19(9/9/1) Sz FC 1370/1770 N/mm^2; the sheave has 97° undercut “V” grooves; the ropes are terminated with sprung eyebolts with babbitted sockets at the counterweight end, and dead eyebolts with clipped returns at the car end; the other relevant details are shown in Figure 2.

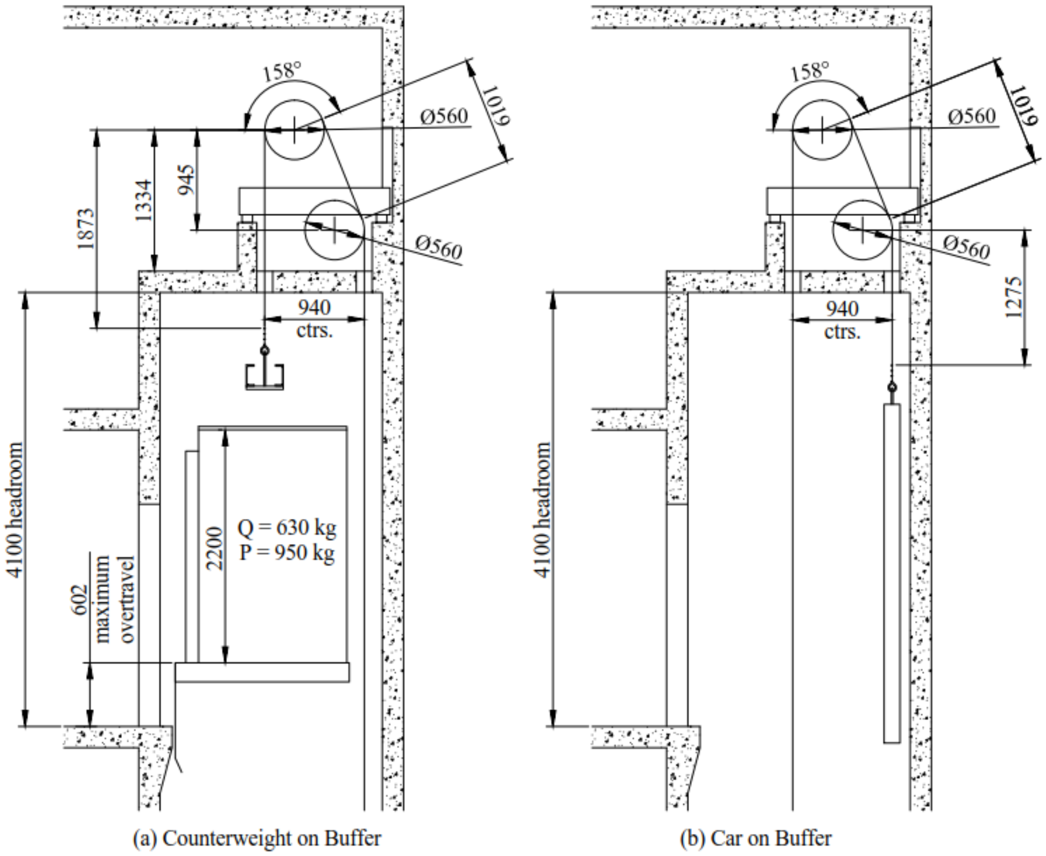

Figure 2 The Existing Arrangement

As the machine is approximately 25 years old, it has been decided to replace it, the diverter, supply a new raft re-using the existing concrete upstands, new ropes and anchorages, a new control panel and some cosmetic refurbishment to the car. The replacement of any other equipment and provision of devices to prevent the uncontrolled movement of the car have been disregarded as they will not have any bearing on the rope life.

A suitable new "Toro" machine has been determined using the software on Sassi's "Argaweb" website; using 97° undercut "U" grooves the same number, size and construction of ropes as the original will be required to meet EN81-20/50. It has been assumed the car weight will increase by 200 kg because of the refurbishment. The other relevant details are shown in

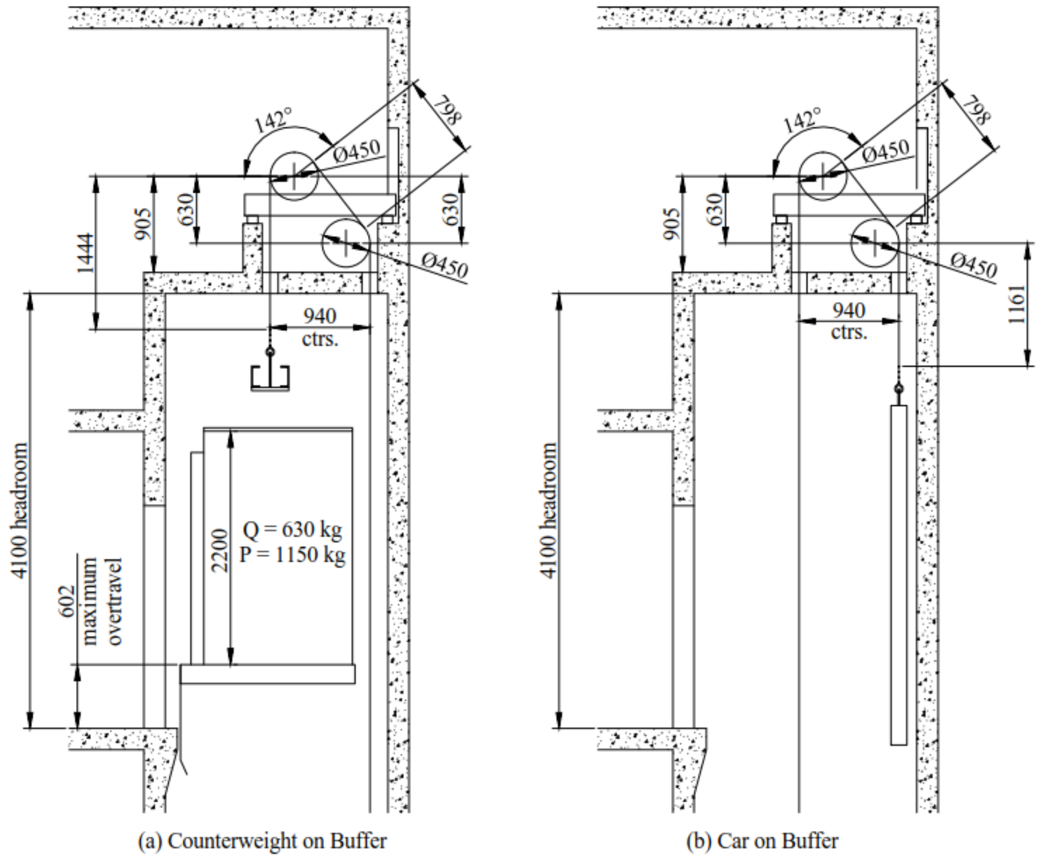

Figure 3 The Modernised Arrangement

The first problem arises trying to fit new anchorages with self-tightening wedge sockets on to the existing anchorage plates. They don't fit, so have to be replaced, with consequences for the fleet angles as shown in Figure 4 and Figure 5.

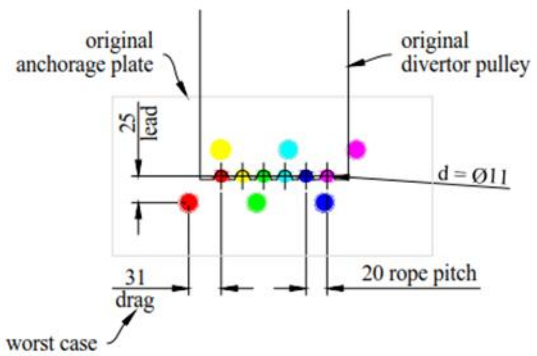

Figure 4 Original Arrangement

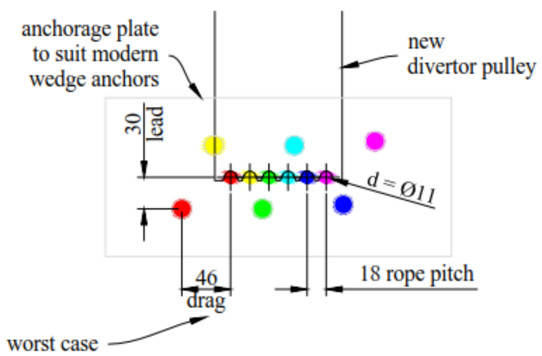

Figure 5 New Arrangement

The worst-case fleet angle was originally 1.4° (i.e. 41:1) but now has increased to 2.3° (approximately 25:1), but the rope to sheave ratio has decreased from 51 to 41 approximately. The only other parameter that will affect the rope lift calculation is the increase in the bending length.

Following the method used in Example 3.11 in Feyrer of a 1:1 roped lift (assuming the car is loaded with 50% of the rated load) for both the original lift and the modernised lift but also considering the effect of the fleet angle gives the results shown in Table 4. The total number of trips before discard ignoring the effect of the fleet angle has also been calculated for comparison.

Table 4 Calculation Results

		Original Lift		Modernised Lift	
		Sheave	Diverter	Sheave	Diverter
Rope Tensile Force	S [N]	3,347		3,831	
Endurance Factors	f_{N1}	1		1	
	f_{N2}	0.87	0.81	0.80	0.75
	f_{N3}	0.13	1	0.13	0
	f_{N4}	1		1	
Discard No Bending Cycles	N_{A10}	1,725,963	12,361,008	417,186	3,008,553
Number of trips to/from G	Z_{A10}	1,514,494		366,381	
Holeschak Ratio	f_{GF}	0.51		0.51	
Total trips before discard **with fleet angle considered**	$Z_{A10,tot}$	2,969,500		718,300	
Total trips before discard **With fleet angle disregarded**	$Z_{A10,tot}$	3,442,900		904,500	

4 CONCLUSIONS

Patrick Ryan states the modern requirements for the rope safety factor will ensure a rope life of 600,000 trips; the results above confirm this. Assuming the example lift will undertake 200,000 trips per annum the ropes on the original lift would have had a life of over 10 years (assuming proper maintenance), but the modernised lift would require the ropes to be replaced within 4 years.

As can be seen from the example in section 3.4 above using a larger rope to sheave ratio (on both the traction sheave and pulleys) has a massively beneficial effect on the rope life. The calculations indicate the ropes on the modernised lift will have an expected life of only 24% of that of the original lift with only a slight improvement if the effect of the fleet angle is ignored.

When the effect of the fleet angle is taken into consideration the anticipated life of the ropes will reduce by about 13% for the original lift, but about 20% for the modernised lift.

If the fleet angle can be shown to have this much of an effect on the rope life of a straightforward conventional 1:1 roped gear above lift, which all parties agree is the “best case scenario” for rope life, how much of an effect will it have on a modern MRL where fleet angles up to 4° are not uncommon and the number of pulleys and the roping arrangement are considerably more complex?

The rope safety factor equation in EN 81-50 is very complicated, its correct application is not explained in terms that are easily understandable and requires a level of mathematical ability many do not possess, thus making it difficult to check the calculations provided by machine manufacturers. The equation assumes the use fibre core ropes and disregards the effect of fleet angle and rope construction, perhaps it is time for it to be reviewed in the light of these concerns and consideration given to returning to a simpler system of placing limitations on fleet angles and increasing minimum rope to sheave ratios?

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BIOGRAPHICAL DETAILS

In 1979, Julia took a temporary job as a Trainee Draughtsman at the Express Lift Company in Northampton, becoming a Sales Engineer. During this time she worked on major projects in the UK and abroad (including Hong Kong, Singapore and Australia), before finally moving on to Modernisation. After the demise of Express Lifts, she worked as a Project Engineer for ThyssenKrupp, Elevator, and Kone before joining WSP as a consultant in 2013. Along the way she kept studying, eventually being awarded an MSc in Lift Engineering from the University of Northampton in 2010.