Rope Life: From Theory to Field Practice, Part 1

The history, the problem, the theory and the solutions

by Nick O'Donnell

Introduction

Rope life is the service interval between installation and mandatory replacement of the suspension ropes in a traction elevator. Replacement is not optional. ASME A17.1/CSA B44 requires ropes to be discarded when they reach the criteria in ASME A17.6 Section 1.10, including excessive crown or valley wire breaks per lay length, diameter reduction below the allowable limit, internal corrosion ("rouge") or other visible degradation that compromises the safety factor. When one rope in a set reaches discard, the entire set is replaced unless the code exception for partial replacement is documented and followed exactly.

Twenty-five years ago, full rope replacement was relatively uncommon unless there was damage or poor maintenance. Ropes just worked. A set of 6x19 or 8x19 ropes on a geared or large gearless machine with a 30 in. or larger drive sheave could deliver 10, 15 or even 20 years of service with routine lubrication and equalized tension. The phrase "rope life" was barely in the vocabulary on the jobsite.

So did the industry simply forget how to do it right? No — the machines and systems changed.

Learning Objectives

After reading this article, you should have learned how to:

- Connect modern MRL and compact gearless design choices to increased bending fatigue and shorter rope-replacement intervals.
- Interpret ASME A17.1/A17.6 discard criteria as field limits and relate them to fatigue mechanisms such as crown breaks, diameter loss and rouge.
- Describe how bending fatigue, tensile load, lubrication condition and statistical discard criteria fit together in the Feyrer rope-endurance model.
- Apply the EN 81-50 equivalent-pulley method to compare rope paths, groove profiles, reverse bends and sheave-size effects from an application standpoint.
- Identify how drive-sheave diameter relative to rope diameter, D/d, affects bending strain, required safety factor and predicted cycles to discard.

The shift started in Europe in 1996 when KONE introduced the MonoSpace®, the first commercial machine-room-less (MRL) traction elevator.[1] By eliminating the machine room and shrinking the drive to a compact permanent-magnet gearless unit, the industry gained floor space, reduced energy use and lowered construction cost. Bids are won or lost on first cost, so manufacturers responded naturally: smaller sheaves (~16–20 in. in many cases), fewer or thinner ropes where the safety factor allowed it, tighter roping arrangements and more reverse bends to fit the machine inside the hoistway. ASME A17.1 added comprehensive guidance for MRL designs in the 2007 edition,[2] and North America followed the European wave. What began as a niche product for low- and mid-rise buildings is now a common choice for new traction installations in many markets.[3]

The first large wave of gearless MRLs installed in the 2000s is now 15–20 years old and entering heavy modernization territory. At the same time, the installed base of traditional geared machines continues to age out. The result is a perfect storm: Thousands of elevators were designed primarily to save space and up-front cost rather than to optimize long-term rope service life. Rope replacement frequency has climbed noticeably. Where a traditional geared job often provided a decade or more of rope life, many modern MRL applications are reaching discard in as few as two-to-three years — sometimes sooner when tension, alignment, groove wear or lubrication is off by even a little.

That is not simply a maintenance failure; it is physics meeting economics. Smaller drive-sheave diameters increase bending stress. More bends, especially reverse bends, multiply fatigue cycles. Competitive pressure pushed designs toward the lowest initial cost while still meeting the minimum code safety factor. The industry is now living with the consequences.



Value: 1 contact hour (0.1 CEU)

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The good news is that rope life can be improved. The first step is understanding why it has decreased in modern systems instead of treating ropes as inevitable short-life consumables. This article reviews the core engineering principles — the EN 81-20/EN 81-50 equivalent-pulley method and the Feyrer fatigue model behind many rope-life calculators — and then translates those principles into practical decisions on sheave sizing, groove profiles, rope selection, installation practices and modernization strategy.

Fundamentals of Rope Fatigue Theory

Rope discard can result from several failure modes, but typical rope-life calculations focus on bending fatigue combined with tensile stress. As the car and counterweight travel, the suspension ropes repeatedly bend over the system sheaves. Each passage creates alternating tensile and compressive stresses in the individual wires: Outer wires stretch while inner wires compress. Over hundreds of thousands of cycles, those repeated strains initiate micro-cracks that grow until visible wire breaks, diameter loss or other discard criteria trigger replacement under ASME A17.6.

This is different from static applications. In elevators, the dominant damage comes from bending over sheaves rather than pure tension fluctuation, although load changes add secondary effects. Visible bending-fatigue signs typically appear first as broken crown wires — the outer wires on the rope surface that contact the sheave groove — as described in A17.6 Section 1.10.1.2.[4]

ASME A17.1/CSA B44 and ASME A17.6 set mandatory discard criteria — broken wires per lay length, diameter reduction, rouge and related conditions — but they do not provide a predictive life-calculation method for design. Instead, they rely on minimum D/d ratios (40:1 for drive sheaves on suspension ropes) and safety factors to keep ropes within acceptable endurance under normal use.

European standards take a more quantitative approach. EN 81-20, with detailed calculations in EN 81-50 Clause 5.12 and Annex E, requires a minimum safety factor on suspension ropes that explicitly accounts for cumulative bending severity through an "equivalent number of pulleys." This method adjusts for groove type, reverse bends and relative sheave diameters. It is based on extensive laboratory testing and is intended to ensure the rope can endure the expected number of bending cycles before reaching discard.

The most widely accepted predictive model for bending endurance in running ropes comes from Prof. Klaus Feyrer at the University of Stuttgart. *Wire Ropes: Tension, Endurance, Reliability* (2nd ed., Springer, 2015) presents regression equations derived from thousands of controlled bending-over-sheave tests. The model predicts the mean number of bending cycles (N) to rope discard, defined as the point where broken wires or measurable degradation trigger code criteria.

The Feyrer formula (formula 3.55 from his book) for running ropes under combined bending and tension is:[5]

$$lgN = b_0 + \left(b_1 + b_3 + lg\frac{D}{d}\right)\cdot\left(lg\frac{S}{d^2} - 0.4\cdot lg\frac{R_0}{1770}\right) + b_2\cdot lg\frac{D}{d} + lgf_d + lgf_L + lgf_c$$

Where:

d is nominal rope diameter, in mm

D is sheave diameter, in mm

S is rope tensile force, in N

R_0 is nominal tensile strength, in N/mm^2

b_0, b_1, b_2 and b_3 are regression coefficients for specific rope constructions detailed in Table 3.14

f_d is an endurance factor accounting for rope diameter scale effect

f_L is an endurance factor for length of the most stressed rope zone

f_C is an endurance factor for rope core type

The formula is presented in base-10 logarithm form, which is standard for endurance curves in wire-rope literature. Several assumptions are built into Feyrer's regression:

- Tests were performed with constant tensile load and repeated bending over a single sheave.
- The groove was round, steel and had a radius of 0.53d (plain U-groove baseline).
- The rope was generously lubricated with oil or Vaseline in dry indoor conditions; dry operation can reduce bend-cycle life by roughly 70–80%.
- No fleet angle was present, although Feyrer introduced a separate factor to account for it.
- Discard was defined by visible wire breaks or diameter loss matching code criteria.
- Results are statistical: N is the mean, and N_{10} is the value that 90% of ropes are expected to exceed with 95% confidence.

EN 81-20 and EN 81-50 build directly on Feyrer's experimental database and regression model. Working groups under CEN/TC 10 translated the full regression equation into a practical design procedure for complex elevator roping systems. Using Feyrer's endurance data, they developed tables that assign an "equivalent number of simple bends" to different sheave groove profiles and apply correction factors for reverse bends and differing sheave diameters. The resulting procedure is published in EN 81-50:2014, Clause 5.12 and Annex E (formerly Annex N in EN 81-1).[6,7]

In elevator systems, the rope experiences multiple bends as it travels over the traction sheave and various deflection and compensation pulleys. Some bends are more damaging than others. EN 81-50 defines a "reverse bend" when two conditions are met simultaneously: The distance between the points of rope contact on two consecutive pulleys with fixed axes is less than 200 times the nominal rope diameter (200d), and the planes of bending are rotated through more than 120°.

To account for the combined effect of sheave quantity and groove geometry, EN 81-50 uses the "equivalent number of pulleys," denoted Nequiv. This single value represents the cumulative bending severity in the most heavily loaded section of the rope path. It is calculated as:

$$N_{equiv} = N_{equiv(t)} + N_{equiv(p)}$$

Where:

$N_{equiv}(t)$ is the equivalent number contributed by the traction sheave, taken from Table 2 of Clause 5.12.2.1 based on

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groove type and opening angle. A semi-circular U-groove (radius ≤ 0.53d) is assigned the baseline value of 1. Undercut U-grooves and V-grooves receive higher values due to increased contact pressure and bending severity.

$N_{equiv}(p)$ accounts for deflection pulleys using the number of simple bends (N_{ps}) and reverse bends (N_{pr}) in the most loaded rope section, adjusted by the fourth power of the diameter ratio between the traction sheave and the deflection pulleys:

$$N_{equiv}(p) = (D_t/D_p)^4 \cdot (N_{ps} \cdot 4N_{pr})$$

The total equivalent number of pulleys (N_{equiv}) and the ratio of traction-sheave diameter to rope diameter (D_t/d) are then used to determine the minimum required safety factor ($S_{f,min}$) according to EN 81-50 Clause 5.12.3. Higher N_{equiv} values or lower D_t/d ratios require a higher minimum safety factor. For example, with $D_t/d = 40$ and $N_{equiv} = 7$ (about a 95° undercut U-groove), the required minimum safety factor is approximately 16.[10] By comparison, A17.1 requires a safety factor of 11 for a 1:1 car moving at 700 ft/min.

The CEN/TC 10 working group calibrated these minimum safety factor values using Feyrer's endurance data to provide adequate service life, corresponding to approximately 300,000 return trips (away from the lobby and back) under typical passenger elevator duty conditions.[8]

North American codes do not mandate this exact equivalent-pulley calculation, and the discard criteria are not perfectly identical across standards. However, the bending-fatigue physics are the same. The EN 81-50 method highlights the dominant variables: drive-sheave diameter relative to rope diameter, number and severity of bends, groove profile and the diameters of the other system sheaves. These variables are the basis for the practical design and modernization decisions discussed below.

Key Design and Application Factors Affecting Rope Life

The EN 81-50 equivalent-pulley method and Feyrer fatigue model identify five dominant variables that control rope endurance in traction elevator systems:

1) Drive-sheave diameter relative to rope diameter
2) Traction-sheave groove profile
3) Number and type of bends, especially reverse bends
4) Nominal wire tensile strength
5) Diameters of the other sheaves in the system

Each variable influences either the total equivalent number of pulleys (N_{equiv}) or the resulting minimum safety factor ($S_{f,min}$) required to achieve approximately 300,000 return trips under typical passenger duty. Using the medium usage category from ISO 25745-2, this can be approximated for this article as roughly two years of bend cycles under constant maximum-use traffic, which is generally conservative.[9]

In the following subsections, each factor is examined independently while holding the others constant. This isolates its effect on both required safety factor and predicted rope life. The charts use the same standardized calculations and provide the quantitative basis for the field and modernization decisions covered later in the article.

Drive-Sheave Diameter (D_t/d)

Of all the variables in EN 81-50 and Feyrer's regression, the ratio of traction-sheave diameter to rope diameter (D_t/d) is the strongest lever for rope endurance. It determines the minimum required safety factor in EN 81-50 Clause 5.12.3 and appears directly in the Feyrer endurance equation, where lg N increases strongly with lg(D/d). Larger D_t/d reduces bending strain per cycle and dramatically increases the number of bending cycles the rope can endure before reaching ASME A17.6 discard criteria.

For any given total equivalent number of pulleys, the required safety factor rises sharply as D_t/d falls below approximately 40. Conversely, increasing D_t/d allows a lower safety factor while still achieving the calibrated target of roughly 300,000 return trips. ASME A17.1/CSA B44 mandates a minimum D_t/d of 40 for suspension ropes on traction sheaves, but that is only a code floor; it does not guarantee long service life in modern compact machines.

Modern compact gearless machines, including MRL designs, are commonly sized at this minimum ratio of $D_t/d = 40$. Common early examples include 320 mm sheaves with 8 mm ropes, 400 mm sheaves with 10 mm ropes, and 20 in. sheaves with ½ in. ropes. Older geared machines often used 24–30 in. sheaves ($D_t/d \geq 50$). Holding all other parameters constant — fixed rope tension S, wire tensile strength R_0, V50 groove and $N_{equiv} = 9$ for a system with one deflection sheave, two car

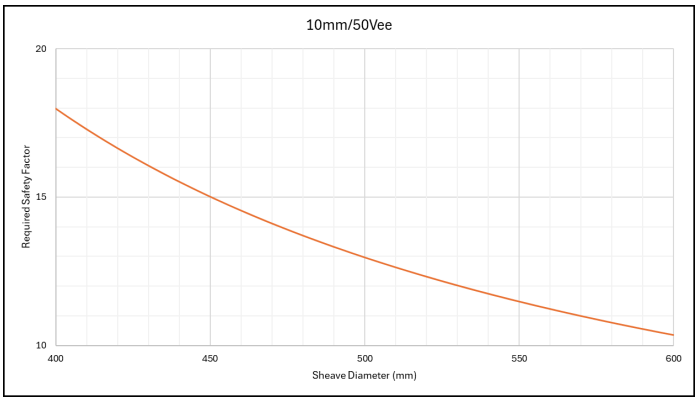


Figure 1: Reduction in required safety factor to achieve EN 81-50 return trips for a V50 10-mm rope system

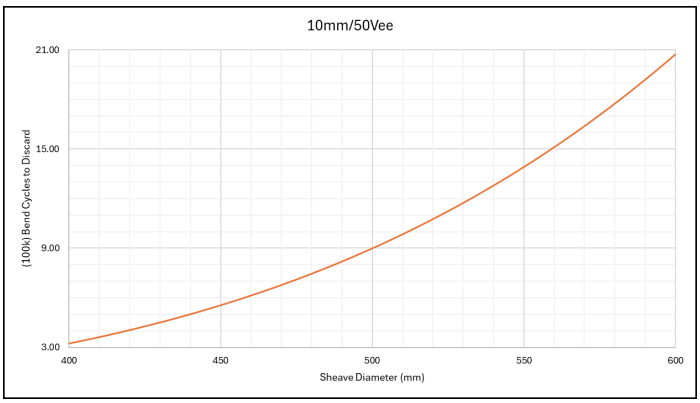


Figure 2: Increase in bend cycles until discard criteria are met for a V50 10-mm rope system

compensating sheaves, and one counterweight compensating sheave with no reverse bends — the EN 81-50 tables show that moving from $D_t/d = 40$ to $D_t/d = 60$ reduces the required minimum safety factor from about 18.0 to 10.4, while the Feyrer-predicted bending cycles to discard increase by roughly five to six times.

The practical takeaway is simple: When layout, application torque and structural constraints allow, select the largest practical drive-sheave diameter. In new designs, this may mean accepting a modest increase in first cost to gain years of additional rope service. In modernizations, upsizing the sheave — where motor torque, brake capacity and machine-beam loading permit — is one of the highest-return-on-investment (ROI) interventions available. Even a 2–4 in. increase in sheave diameter can extend rope life materially with no change in rope construction or maintenance practice.

This single parameter explains much of the difference between early compact MRL rope life and the long service intervals still seen on well-maintained large-sheave machines.

Traction-Sheave Groove Profile and Equivalent-Bend Method

Traction-sheave groove profile has one of the largest individual effects on bending severity. EN 81-50 Clause 5.12.2.1 assigns an equivalent number of simple bends contributed by the traction sheave, $N_{equiv}(t)$, based on groove geometry and opening angle. This value is added directly to the deflection-pulley contribution to produce total N_{equiv} for the Clause 5.12.3 safety-factor table. Groove geometry also affects the groove factor in Feyrer's endurance equation, so the same change can shift both the required minimum safety factor and predicted bend cycles to discard.

The table below follows the EN 81-50 values reproduced in Tavaslıoğlu's 2023 ELEVATOR WORLD article, "Reverse Bending and Its Effects on Lift Ropes":[11]

V-grooves	V-angle (γ)	35°	36°	38°	40°	42°	45°	50°
	$N_{equiv}(t)$	18.5	16	12	10	8	6.5	5
Undercut U-grooves	U-angle (β)	75	80	85	90	95	100	105
	$N_{equiv}(t)$	2.5	3.0	3.8	5.0	6.7	10.0	15.2

Table 1: Equivalent number of traction-sheave bends for V-grooves and undercut U-grooves

Two notes are important. First, a plain U-groove with no undercut is assigned $N_{equiv}(t) = 1$. Second, values for groove geometries not explicitly listed in the table can be obtained by linear interpolation or a best-fit curve.

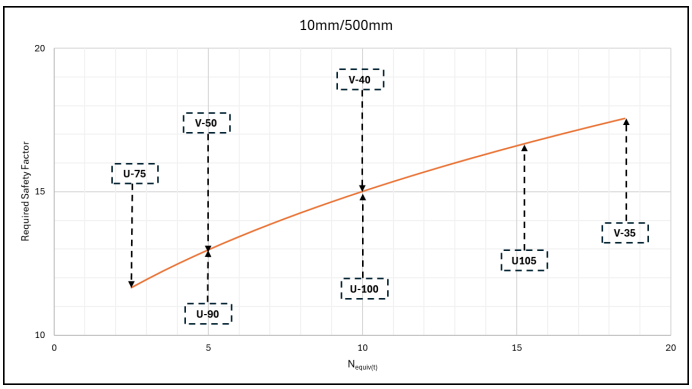


Figure 3: Increase in required safety factor as equivalent bends from traction-sheave groove profile increase

For comparison with the previous section, assume the same 10 mm rope, $D_t/d = 50$, and the same arrangement of three compensating sheaves and one deflector sheave ($N_{equiv}(p) = 4$). Varying only $N_{equiv}(t)$ from 1 (plain U-groove) to 18.5 (35° V-groove) illustrates the effect on the required safety factor under EN 81-50.

Figure 3 makes the first point clear: Lower $N_{equiv}(t)$ produces a lower required safety factor and higher predicted bend-cycle life. It does not, however, explain how to choose a groove. The missing piece is traction, which is also driven by the same V-angle (γ) and U-angle (β) parameters used in $N_{equiv}(t)$.

For the analytical approach to traction requirements in a roped elevator system, see equations 2.2 and Table 2.1 in Andrew and Kaczmarczyk's *Systems Engineering of Elevators*:[12]

$$\frac{T_1}{T_2} < e^{f\alpha}$$

Where:

T_1/T_2 is the ratio of rope tensions where traction between the ropes and drive sheave must be maintained

f is the friction factor related to groove geometry

α is the angle of wrap of the suspension ropes around the traction sheave

and:

Plain U-groove	$f = 4\mu\frac{\cos(\gamma/2)}{\pi - \gamma + \sin\gamma}$
Undercut U-groove	$f = 4\mu\frac{\cos(\gamma/2) - \sin(\beta/2)}{\pi - \gamma - \beta + \sin\gamma - \sin\beta}$
Hardened V-groove	$f = \frac{\mu}{\sin(\gamma/2)}$
Non-hardened undercut V-groove	$f = 4\mu\frac{1 - \sin(\beta/2)}{\pi - \beta - \sin\beta}$

Where:

μ is the coefficient of friction between non-lined, machined steel or iron grooves and steel ropes

γ is the groove angle, in radians

β is the undercut angle, in radians

This makes it possible to compare a groove's traction capability against its impact on rope life as a function of groove angle. The two independent variables can be plotted against each other to normalize the comparison between groove profiles.

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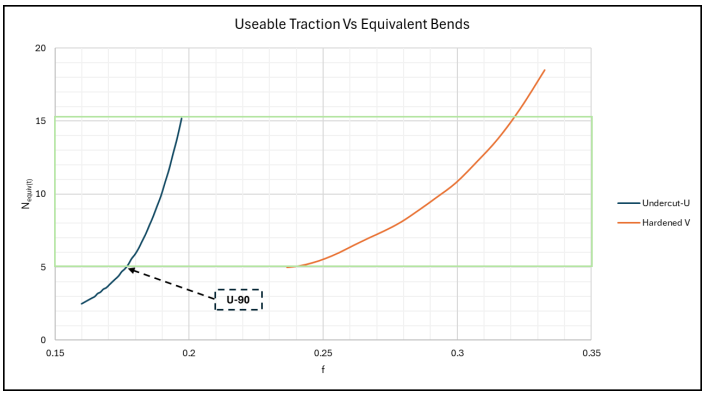


Figure 4: Normalized comparison of undercut U-groove and hardened V-groove traction performance versus rope-life impact

If rope life is the goal, Figure 4 suggests that undercut U-grooves with α greater than 90° are usually hard to justify under the equivalent-bend approach. At the same $N_{equiv}(t)$, a hardened V-groove can provide a better friction factor. With most modern suspension ropes using an outer tensile grade of 1550 N/mm² or higher, the required V-groove hardening should not, by itself, dominate real-world rope life. The decision then often comes down to whether future regrooving is desired and whether the cost of hardening is acceptable.

The next question is how to reduce the required friction factor for the elevator system. The primary lever is angle of wrap.

In many designs, angle of wrap is set by drive-sheave diameter and the distance between car and counterweight pickup points. With maximum T_1/T_2 already defined, the designer has two basic ways to satisfy the traction equation: make the groove more aggressive or increase wrap by using a double-wrap arrangement. The first increases available f; the second lowers required f. If the required traction can be reduced enough to allow a plain or mildly undercut U-groove, rope life improves significantly.

Figure 5 compares the same baseline N_{equiv} approach for double-wrapped undercut U-grooves and single-wrapped hardened V-grooves, using a base α of 165° and equal 500 mm drive and secondary sheaves.

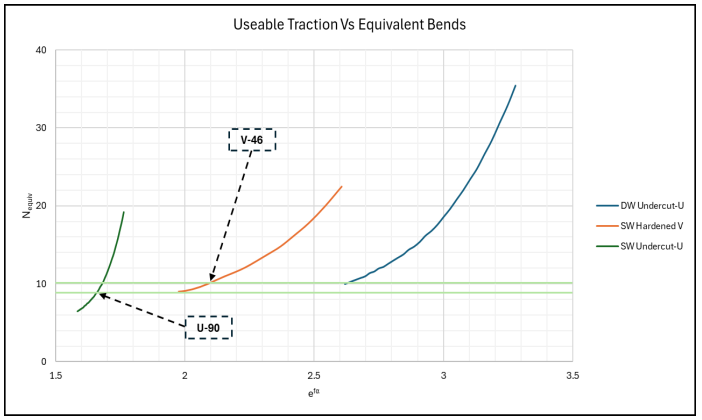


Figure 5: Normalized comparison of equivalent bends against traction capability for selected groove profiles and wraps

This comparison supports the following practical rules when rope life is the priority and a single-wrap system is preferred for installation and cost reasons:

- For very low traction requirements, use undercut U-grooves up to an undercut angle of about 90°.
- As traction requirements increase, a hardened V-groove is generally preferable up to approximately a 46° groove angle.
- Beyond the traction capability of a 46° V-groove, every effort should be made to use a double-wrap system with the smallest practical undercut angle.

These rules aim to keep the traction-sheave contribution to roughly six equivalent bends or less, corresponding to a required safety factor of about 13 or higher under the European method. Actual elevator designs still require judgment. For example, double-wrap groove offset can introduce fleet angle; according to Feyrer's endurance factors, even 1° of induced fleet angle can reduce bend cycles by about 13% in the baseline scenario. This effect is not captured in the EN 81-50 Clause 5.12.3 safety-factor table.

This helps explain why compact MRL and small gearless designs often use V-grooves or more aggressive undercut angles: Limited overhead space and higher T_1/T_2 ratios from chain or minimal compensation push the design toward more aggressive traction solutions.

Deflection of Suspension Ropes

The number and type of bends in the rope path are the next major contributor to cumulative fatigue severity. While $N_{equiv}(t)$ isolates the traction sheave, $N_{equiv}(p)$ accounts for all deflection and compounding sheaves in the most loaded rope section. EN 81-50 Clause 5.12.2.2 defines this contribution as:

$$N_{equiv}(p) = (D_t/D_p)^4 \cdot (N_{ps} \cdot 4N_{pr})$$

where N_{ps} is the number of simple bends and N_{pr} is the number of reverse bends. The fourth-power diameter-ratio term reflects the strong sensitivity of bending fatigue to sheave diameter, while the 4x multiplier on reverse bends captures the higher alternating stress that occurs when rope curvature reverses direction.

As Tavaslıoğlu explains,[11] EN 81-50 defines a reverse bend rigorously: The distance between rope contact points on two

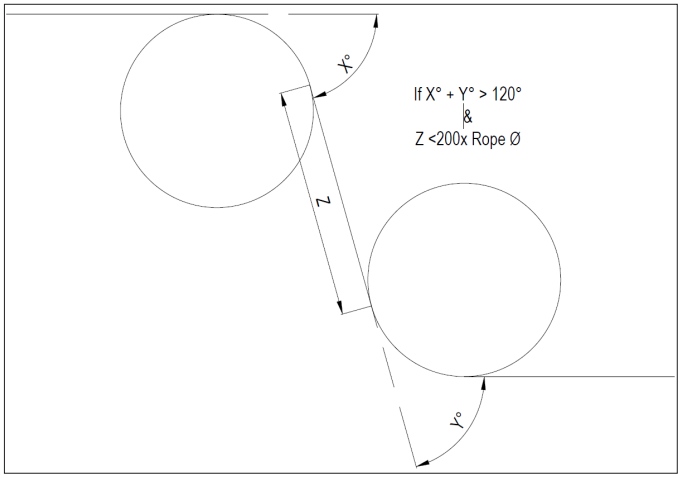


Figure 6: Reverse-bend definition

consecutive fixed-axis pulleys must be less than 200 times the nominal rope diameter (200d), and the planes of bending must rotate through more than 120°. These conditions are common in compact MRL and overhead gearless machines where space constraints force tight rope paths.

For the baseline 10-mm rope on a 400-mm drive sheave ($D_t/d = 40$, V50° groove, $N_{equiv}(t) = 5$), a typical compact gearless layout includes one deflection sheave, two car compensating sheaves and one counterweight compensating sheave ($N_{ps} = 4$, $N_{pr} = 0$). This gives $N_{equiv}(p) = 4$ and total $N_{equiv} = 9$. Adding a single reverse bend ($N_{pr} = 1$) increases $N_{equiv}(p)$ to 8 and total N_{equiv} to 13. The EN 81-50 table then requires a higher $S_{f,min}$ while Feyrer-predicted bending cycles to discard drop by roughly 2.5 to 3 times.

The field takeaway is to minimize bends — especially reverse bends — wherever hoistway geometry and structural constraints allow. In new designs, this may mean reconfiguring the roping pattern or accepting a slightly wider hoistway. In modernizations, converting a reverse bend into an ordinary bend or inserting an additional deflection sheave to split a tight reverse can be a high-ROI change without replacing the machine.

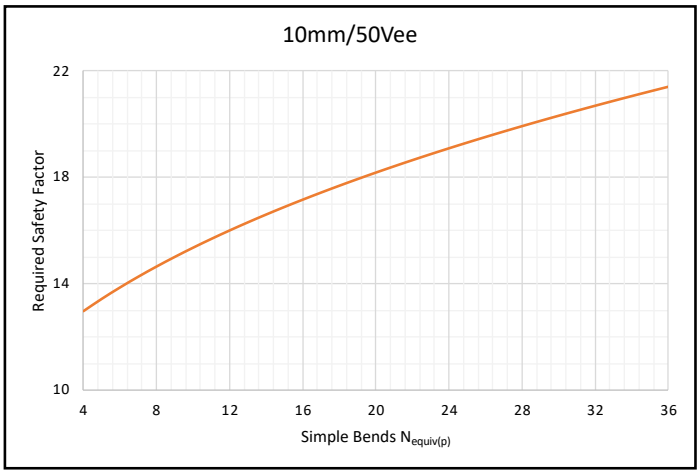


Figure 7: Increase in required safety factor as the number of simple bends increases

A second bend-related factor is fleet angle, also called side deflection or side angle. Feyrer includes a fleet-angle correction factor in his endurance model. In the full regression, endurance N is multiplied by $f\alpha < 1$, where $f\alpha$ decreases rapidly as fleet angle α increases. A practical approximation from Feyrer's test data is:

$f\alpha \approx 1 - 0.13 \cdot \alpha$ (α in degrees, valid up to approximately 4°)

Thus, a fleet angle of only 1° reduces predicted bend cycles to discard by approximately 13%. At 2°, the reduction approaches 26%. This correction is applied directly to the calculated N_{10} from Formula 3.55 and is independent of the equivalent-pulley count in EN 81-50. In other words, the EN 81-50 safety-factor approach does not capture fleet angle.

Fleet angle is most commonly introduced by groove offsetting in double-wrap machines or by minor misalignment of deflection sheaves. Even small values that appear acceptable on paper can measurably shorten rope life.

Compact gearless and MRL layouts make this worse because shorter distances between deflectors turn small alignment errors into significant fleet angles. When rope life is the priority, the message is direct: design and maintain the rope path to keep both bend count and fleet angle as low as practical while still meeting traction and space constraints.

Rope Breaking Strength and Wire Tensile Strength

Rope breaking strength enters the EN 81-50 framework in two related ways. First, the minimum safety factor $S_{f,min}$ is the ratio of the rope's minimum breaking load to the maximum tension in the most loaded rope. A rope with higher breaking strength can meet the required $S_{f,min}$ at a given operating tension, or carry higher tension while still satisfying the endurance target. Second, nominal wire tensile strength R_0 appears directly in Feyrer's regression. As R_0 increases, predicted bending cycles to discard increase for a fixed line pull S, sheave diameter D and groove condition.

In practical terms, higher wire tensile strength lowers the cyclic strain experienced by the outer wires during each bend. For the same tension S, the stress relative to the wire's ultimate strength is lower, delaying fatigue-crack initiation and growth.

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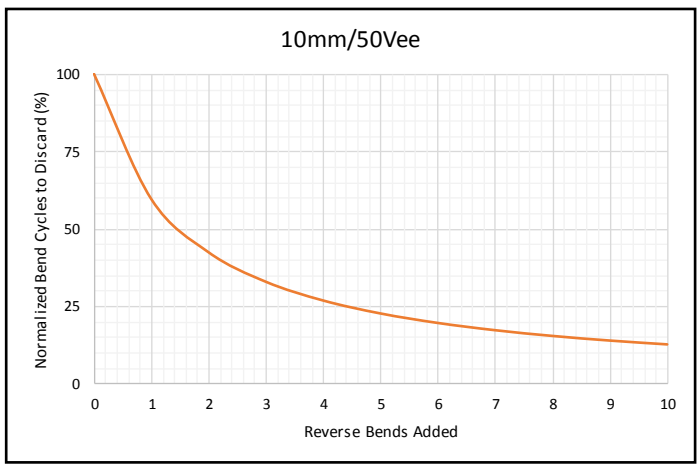


Figure 8: Predicted discard-cycle reduction per Feyrer Table 3.16 factors and the baseline case

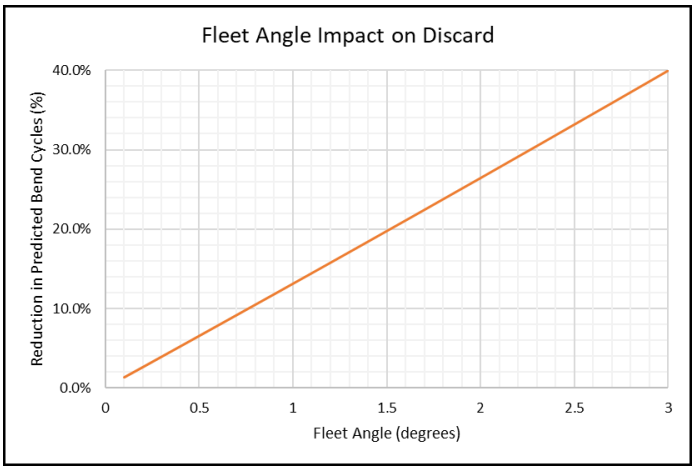


Figure 9: Percentage reduction in predicted bending cycles to discard versus fleet angle using Feyrer's correction factor

This effect is independent of N_{equiv} but compounds with the benefit of larger D_t/d and lower $N_{equiv}(t)$ or $N_{equiv}(p)$.

For the baseline 10 mm rope on a 400 mm drive sheave ($D_t/d = 40$, V50° groove, $N_{equiv} = 9$), moving from a 1,570 N/mm² wire tensile grade to 1,770 N/mm² increases Feyrer-predicted bending cycles to discard by roughly 35–50% when all other parameters are held constant. The exact multiplier depends on the specific rope construction coefficients in Feyrer Table 3.14, but the direction and scale are consistent across common elevator rope classes.

Rope construction also affects breaking strength for a given nominal diameter. Independent Wire Rope Core (IWRC) ropes typically provide 10–15% higher minimum breaking load than fiber-core (FC) ropes of the same nominal diameter because the steel core carries more tensile load and increases metallic fill factor. That added breaking strength can raise $S_{f,min}$, or it can allow a smaller nominal rope diameter while maintaining load capacity. The smaller diameter improves the effective D_t/d ratio and reduces absolute bending strain.

The practical implication is that when layout constraints keep D_t/d near the code minimum and N_{equiv} is elevated by groove choice or reverse bends, a higher tensile-grade rope — or a move from FC to IWRC where elongation and torque behavior permit — is one of the most direct ways to recover bending-cycle life without changing sheave diameter or roping geometry.

This lever complements the three already discussed: drive-sheave diameter, groove profile and bend count. The best results usually come from combining improvements: increase D_t/d where possible, reduce $N_{equiv}(t)$ and $N_{equiv}(p)$ and select the highest practical breaking strength consistent with traction, elongation and installation requirements.

Other System Sheave Diameters

Deflection and compounding sheave diameters enter EN 81-50 through the $N_{equiv}(p)$ term. Because this contribution is scaled by $(D_t/D_p)4$, even modest reductions in the average diameter of the other sheaves produce disproportionately large increases in total N_{equiv} and required $S_{f,min}$. Feyrer's regression reinforces the same physical point: Every sheave imposes bending cycles at its own D/d ratio, so smaller deflection sheaves add fatigue damage even though they are not part of Nequiv(t).

For the baseline 10-mm rope on a 400-mm drive sheave with one deflection sheave, two car compensating sheaves and one counterweight compensating sheave ($N_{ps} = 4$, $N_{pr} = 0$), the picture is straightforward. If all four other sheaves are also 400 mm, the diameter-ratio term is 1.0, $N_{equiv}(p) = 4$, and total $N_{equiv} = 9$. Reducing the average deflection-sheave diameter to 300 mm (a non-code-compliant example used only to show sensitivity) raises $(D_t/D_p)4$ to about 3.16. $N_{equiv}(p)$ jumps to 12.6, total N_{equiv} rises to 17.6 and predicted cycles to discard fall by roughly half. Increasing average deflection-sheave diameter to 500 mm lowers $(D_t/D_p)4$ to about 0.41, drops total N_{equiv} to about 6.6 and increases predicted cycles to discard by roughly 1.7 to 2.0 times.

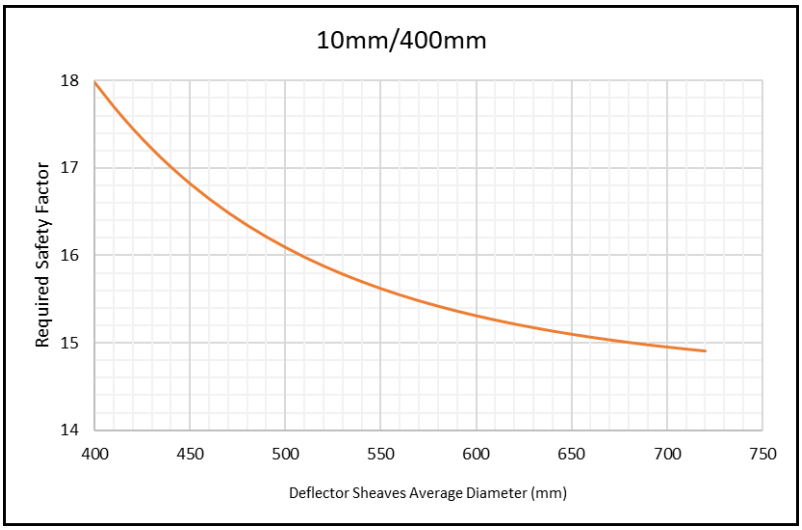


Figure 10: Required minimum safety factor reduction as average deflecting-sheave diameter increases

Compact MRL and overhead gearless machines often size deflection and compensating sheaves as small as structural clearance and weight allow. That saves space and mass, but the fourth-power sensitivity directly increases cumulative bending severity. In many modernizations, upsizing deflection or compensating sheaves — where beam loading and hoistway geometry permit — can recover a meaningful fraction of lost bend-cycle life without changing the traction sheave or rope construction.

Together, these five variables quantify why first-generation compact gearless installations often experience much shorter discard intervals than older large-sheave machines. They also provide a practical basis for new design decisions and modernization tradeoffs.

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Nick O’Donnell began his elevator industry career in 2014 as a product support engineer with TK Elevator. Over the next several years, he developed expertise in elevator system engineering, with a focus on ropes, machines and structural design before moving into engineering leadership. He went on to manage the System Engineering team and later served in a global chief engineering role focused on configuration, standardization and design automation. Today, he is director of Engineering at Hyperion Solutions, where he leads the product roadmap and R&D team across multiple elevator product lines.

Accounting of Time for Article

- The History & Problems of Rope Life: 30 Minutes – Page 104
- The Theory and the Solutions of Rope Life: 30 Minutes – Page 105 – 110

Learning-Reinforcement Questions

Use the below learning-reinforcement questions to study for the Continuing Education Assessment Exam available online at elevatorbooks.com or on page 124 of this issue.

- The article argues that shortened rope life is not simply a maintenance failure. What changed in elevator system design, and why did those changes make rope fatigue more severe?
- ASME A17.1/CSA B44 and ASME A17.6 establish minimum safety and discard requirements, while EN 81-50 gives a more quantitative endurance method. Why are both perspectives useful when evaluating rope life?
- In plain language, what does the Feyrer model try to predict, and what major assumptions have to remain true for its prediction to be meaningful in the field?
- Why is D/d treated as the single strongest design lever for rope endurance? Give one new-installation example and one modernization example where this matters.
- How does the equivalent number of pulleys, N_{equiv}, turn a complicated rope path into something that can be compared from one design to another?

ELEVATOR WORLD Continuing Education Assessment Examination Questions

Read the article **"Rope Life: From Theory to Field Practice, Part 1"** (EW, October 2026, p. 104) and study the learning-reinforcement questions at the end of the article.

- To receive **one hour (0.1 CEU)** of continuing-education credit, answer the assessment examination questions found below online at elevatorbooks.com or fill out the ELEVATOR WORLD Continuing Education reporting form found overleaf and submit by mail with payment.
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1. A compact gearless job is going through ropes much faster than expected. Before assuming the rope itself is defective, which combination of checks best matches the article's core argument?
   a. Check the rope supplier first, then review the sheaves only if another rope set fails early.
   b. Verify D/d, groove severity, reverse bends, system weights, tension, alignment and lubrication.
   c. Use the ASME discard count only, because in the real world design variables no longer matter once the car is in service.
   d. Treat short rope life as unavoidable on all MRLs and focus only on scheduling replacement work and maintaining as best as possible moving forward.

2. Which statement best captures how the article uses North American code requirements together with the EN 81-50 approach?
   a. ASME provides the predictive equivalent-pulley calculation, while EN 81-50 only lists discard criteria.
   b. EN 81-50 replaces the need for ASME A17.6 discard inspection when the design safety factor is high enough.
   c. ASME defines mandatory safety and discard limits; EN 81-50 gives a useful way to quantify bending severity in design.
   d. Both approaches are treated as identical because they use the same equations, tables and inspection thresholds.

3. In the equivalent-pulley method, why is a single reverse bend treated differently from adding one more ordinary bend?
   a. It is ignored when the drive sheave already has a V-groove because groove profile controls the entire calculation.
   b. It is mainly an inspection term and has little connection to design-endurance calculations.
   d. It improves life by spreading wear across both sides of the rope instead of only the crown wires.
   d. It reverses rope curvature and is counted with a higher severity factor than an ordinary bend.

4. A designer proposes using a smaller nominal rope on the same machine to improve D/d. Which response best fits the article's theory-and-application approach?
   a. Accept it automatically because D/d is the only variable that matters once the code minimum is met.
   b. Consider the fatigue benefit, but recheck breaking strength, safety factor, groove fit and elongation.
   c. Reject it automatically because smaller ropes always reduce rope life.
   d. Treat it as an ASME discard issue only, with no connection to EN 81-50 or Feyrer-based endurance.

5. A single-wrap machine cannot meet traction with a mild groove. If rope life is still a priority, what is the most balanced next step to evaluate?
   a. Move to the smallest V-groove angle available because traction should always be solved before fatigue is considered.
   b. Reduce the number of suspension ropes so the remaining ropes sit deeper in the same groove profile.
   c. Increase the available wrap or consider double-wrap so the groove may not need to be as aggressive.
   d. Avoid lubrication because it reduces the friction factor available at the traction sheave.

6. Which factor below is a real contributor to rope life in the article, but is specifically called out as not being captured by the EN 81-50 Clause 5.12.3 safety-factor table?
   a. Fleet angle caused by groove offset or sheave misalignment.
   b. Traction-sheave groove profile.
   c. Reverse bends in the most loaded rope section.
   d. Average deflection-sheave diameter.

7. For $N_{equiv}(p)$, which change attacks the pulley contribution most directly?
   a. Use a higher tensile-grade rope and document the inspection interval more carefully.
   b. Regroove the drive sheave to a less aggressive profile.
   c. Add more deflection sheaves so that the rope bends less over each one.
   d. Increase deflection or compensating sheave diameters where the structure allows it.

8. The article emphasizes the fourth-power term on deflection and compensating sheave diameters. What practical lesson follows from that point?
   a. Small reductions in secondary sheave diameter can create a disproportionately large increase in cumulative bending severity.
   b. Deflector or secondary sheave diameter matters if reverse bends are present.
   d. The drive sheave diameter matters but deflector sheave does not.
   d. The fourth-power term applies to the number of ropes in the set.

9. A double-wrap machine shows acceptable calculated D/d and N_{equiv}, but field inspection finds measurable side deflection at the grooves. What should the reviewer be concerned about?
   a. The side deflection should improve life because it spreads rope contact across a wider portion of the groove.
   b. The concern is already fully covered by $N_{equiv}(t)$, so no separate alignment review is needed.
   c. The only likely consequence is higher building energy use, not a change in rope fatigue behavior.
   d. Fleet angle may be reducing actual bend-cycle life even though the equivalent-pulley check looks acceptable.

10. Why does the article recommend filling available traction-sheave grooves in severe compact applications when practical?
   a. It allows conversion of a V-groove into a plain U-groove by distributing the ropes over more contact points.
   b. It lowers the load carried by each rope and improves the effective safety factor for endurance.
   c. It removes the need to inspect for crown wire breaks because each rope carries less load.
   d. It directly increases D/d because the drive-sheave diameter changes when more grooves are filled.

ELEVATOR WORLD Continuing Education Reporting Form

Article title: **"Rope Life: From Theory to Field Practice, Part 1"** (EW, October 2026, p. 104)

Continuing-education credit: This article will earn you one contact hour (0.1 CEU) of elevator-industry continuing-education credit.

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