How to Size an Actuator for Unwind and Rewind Web Guides

Article

Introduction

This is Part 2 of our series on unwind and rewind web guides. In Part 1, we covered the design and installation of the guide frame — the shifting stand, linear bearings, and mechanical layout. Now we turn to the actuator: the component that converts electrical energy into the precise linear motion needed to position the web.

We manufacture the RLA Series actuators specifically for terminal web guiding applications, so we have a direct interest in helping you size them correctly. But the force model presented here is universal. You can use it to size any linear actuator — hydraulic, pneumatic, or electric — for any shifting-stand web guide. We would rather you size the job correctly and choose a competitor's product than undersize one of ours and have it fail in your mill.

What follows is our engineering methodology for determining the minimum thrust required to move a loaded roll stand reliably. We will build the force model from first principles, account for the real-world forces that simplified models often omit, and work through a realistic example with numbers you would actually encounter on a converting or paper line.

The Complete Force Model

Most actuator sizing guides present a clean two-term equation: friction plus inertia. That model is correct in a classroom, but it is incomplete for an installed web guide. Experienced web handling engineers know that the real force budget includes several additional terms that can individually exceed the friction force.

We will build the model in layers, starting with the textbook terms and then adding the forces that show up on the plant floor.

The Full Force Equation

$$F_{total} = F_{friction} + F_{inertia} + F_{tension} + F_{umbilical} + F_{grade} + F_{misalignment}$$

Where:

Term	What It Represents
$F_{friction}$	Rolling or sliding friction of the linear bearing system
$F_{inertia}$	Force required to accelerate the total moving mass
$F_{tension}$	Lateral force from web tension acting through the steering angle
$F_{umbilical}$	Drag from cables, hoses, and conduit that flex as the stand moves
F_{grade}	Gravitational component if the floor is not perfectly level
$F_{misalignment}$	Additional resistance from real-world installation tolerances

After computing the comprehensive total, we apply a true safety factor for genuine unknowns:

$$\mathrm{F_{required} = F_{total} \times FoS}$$

This is an important distinction. The safety factor should account for unknowns the model cannot capture — transient shock loads, degradation over time, contamination — not compensate for forces we failed to model. By accounting for all known forces explicitly, we can use a more rational FoS rather than inflating it to cover model deficiencies.

Term 1: Bearing Friction — $F_{friction}$

$$\mathrm{F_{friction} = \mu \cdot m_{total} \cdot g}$$

Where: - m_{total} — combined mass of the full roll and the shifting stand (kg) - μ — coefficient of friction for the bearing system - g — gravitational acceleration, 9.81 m/s²

Catalog Friction Coefficients

Bearing / Guide Type	Catalog μ	Notes
Profile linear rail (recirculating ball)	0.003–0.005	Most common in modern guide frames
Circular linear bearing (round shaft)	0.004–0.006	Round shaft guides
Cam rollers on machined steel plate	0.002–0.005	Common in legacy guide frames
Needle roller bearings	0.004–0.005	Track rollers, cam followers
Precision roller bearings	~0.0018	High-precision applications
Plain bushing / bronze sleeve	0.07–0.12	Legacy equipment, high friction

For profile linear rails — the most common bearing type in modern shifting stands — catalog friction is very low. On a 13,000 lb (5,897 kg) stand, $F_{friction}$ at $\mu = 0.004$ computes to only 52 lbf (231 N). That number is correct for a clean, perfectly aligned rail in a catalog. It is not necessarily what you will see on the plant floor, which is why Term 6 (misalignment) exists.

Term 2: Inertia — $F_{inertia}$

$$F_{\text{inertia}} = m_{\text{total}} \cdot a$$

Where: - a — desired linear acceleration of the stand (m/s²)

Web guiding does not require high acceleration. Typical correction speeds are 0.25–1.0 in/s (6–25 mm/s), and the stand accelerates over 50–200 ms. A reasonable design acceleration for most terminal guides is 1–3 in/s² (0.025–0.076 m/s²).

Even at these modest rates, inertia can dominate the bearing friction term when the load is heavy. At 2 in/s² (0.051 m/s²) with a 13,000 lb total load, $F_{inertia}$ is approximately 67 lbf (300 N) — already larger than the friction term for profile linear rails.

Term 3: Web Tension Lateral Force — $F_{tension}$

This is the force most commonly omitted from simplified sizing models, and it can be the largest single term in the equation.

When a web guide steers the stand, the guide roller deflects the web at a small angle. The web, under tension, pulls back laterally. The lateral force is proportional to the web tension and the sine of the steering angle:

$$F_{tension} = 2 \cdot T_{web} \cdot \sin(\alpha)$$

Where: - T_{web} — total web tension (lbf or N) - α — steering angle at the guide roller (radians or degrees) - The factor of 2 accounts for the web entering and leaving the guide roller

The steering angle α is typically small in web guiding — often 0.5° to 2°. But web tensions can be substantial. Consider a paper mill running 4 PLI on a 120-inch web:

- T_{web} = 4 PLI × 120 in = 480 lbf
- α = 1° (0.0175 rad)
- $F_{tension}$ = 2 × 480 × sin(1°) = 2 × 480 × 0.01745 = **16.8 lbf**

At 2° steering angle, that doubles to approximately 33.5 lbf. On a high-tension process — steel, foil lamination, or heavy board — web tensions of 1,000–3,000 lbf are common, and the lateral force scales proportionally.

For converting applications with lighter webs (film, nonwoven), this term may be only 5–15 lbf. For paper and board at moderate tensions, expect 15–60 lbf. For metal foil and high-tension processes, this term can reach 50–150 lbf or more.

Important: This is a sustained force, not transient. The actuator must overcome it continuously during any active correction, not just during acceleration.

Term 4: Umbilical and Cable Drag — $F_{umbilical}$

Every shifting stand has connections that move with it: electrical cables for sensors and drives, pneumatic or hydraulic hoses for brakes or tension systems, grease lines, signal conduit. These umbilicals resist motion through bending stiffness, friction where they slide or hang, and inertia of the cables themselves.

There is no clean formula for this force because it depends entirely on the installation — how many hoses, what diameter, how they are routed, whether they are in a cable carrier or draped freely. In practice:

Installation Complexity	Typical $F_{umbilical}$
Light — a few signal cables, cable carrier	5–15 lbf (22–67 N)
Moderate — signal cables plus 1–2 pneumatic hoses	15–30 lbf (67–133 N)
Heavy — hydraulic lines, multiple hoses, rigid conduit	30–60 lbf (133–267 N)

If you cannot measure the umbilical drag directly (a spring scale pull test on the disconnected stand is the most reliable method), use 20–50 lbf as a reasonable estimate for a typical converting or paper line with moderate hose connections. This term is often underestimated.

Term 5: Floor Slope — F_{grade}

Mill floors are not flat. Settling, construction tolerances, and intentional drainage grades mean the stand's travel axis may be on a slight slope. The gravitational force component along the travel direction is:

$$F_{grade} = m_{total} \cdot g \cdot \sin(\theta)$$

Where: - θ — angle of the floor slope along the travel axis

This force is constant and directional — it helps motion in one direction and opposes it in the other. The actuator must be sized for the opposing direction.

How large is this in practice? A 0.5° slope — barely perceptible to the eye — on a 13,000 lb load:

$$F_{grade} = 5{,}897 \times 9.81 \times \sin(0.5°) = 5{,}897 \times 9.81 \times 0.00873 = 505 \text{ N} \approx 113 \text{ lbf}$$

That single term exceeds the combined friction and inertia forces from the textbook model. If your mill floor has even a minor grade along the actuator's stroke axis, you must account for it.

For most installations, measure the slope with a precision level. If measurement is impractical, assume 0.25°–0.5° as a conservative estimate.

Term 6: Misalignment and Binding — $F_{misalignment}$

This is the term that separates textbook calculations from field reality. Catalog friction coefficients assume perfect rail alignment, proper preload, clean surfaces, and correct mounting torque. Installed systems have:

- Rail parallelism errors from shimming and floor irregularities
- Carriage preload changes from bolting to an imperfect subframe
- Contamination from paper dust, oil mist, metal fines
- Thermal effects causing differential expansion

Industry experience — widely documented in linear motion engineering references — is that installed friction is typically **2 to 5 times** catalog values for well-maintained systems, and can reach **10 times** catalog values for poorly maintained or contaminated installations.

Rather than trying to model these effects individually, the practical approach is to apply an installation multiplier $k_{install}$ to the bearing friction term:

$$F_{misalignment} = (k_{install} - 1) \cdot F_{friction}$$

Where $k_{install}$ ranges from:

Installation Quality	$k_{install}$
Excellent — precision-ground subframe, clean environment, maintained	2–3
Typical — structural steel subframe, moderate contamination	3–5
Poor — legacy frame, heavy contamination, no maintenance program	5–10

This formulation keeps the misalignment force proportional to the load (heavier loads have more rail deflection and more sensitivity to alignment errors) while acknowledging that it is an estimate. If in doubt, use $k_{install}$ = 3 for a new installation with reasonable alignment practices.

Practical note: A spring-scale pull test on the actual stand, loaded with a representative roll, will give you a measured friction force that inherently includes the misalignment term. If you can perform this test, it is always preferable to calculation.

Applying the Safety Factor

With all known forces modeled explicitly, the factor of safety can serve its intended purpose: providing margin for true unknowns.

$$F_{required} = F_{total} \times FoS$$

With a comprehensive force model, a FoS of **1.5** is reasonable for well-characterized installations. Use **2.0** if significant uncertainty remains — for example, if you estimated multiple terms rather than measuring them, or if the operating environment is harsh.

Situation	**Recommended FoS**
All forces measured or well-characterized	1.25–1.5
Some forces estimated, clean environment	1.5–2.0
Multiple estimates, harsh/contaminated environment	2.0–2.5

Compare this to the common practice of using a simplified (two-term) model with FoS of 3–5. Both approaches can arrive at the same required force — but the comprehensive model tells you *why* you need it, which makes troubleshooting easier if the actuator struggles in service.

From Motor Torque to Linear Thrust

The RLA actuators use a reverse-parallel belt-and-ballscrew mechanism: a toothed belt connects the motor to the ballscrew through a pulley reduction, and the ballscrew converts rotary motion to linear thrust.

The thrust conversion factor:

$$k = R_{belt} \cdot \frac{2\pi}{L} \cdot \eta$$

$$F_{thrust} = T_{motor} \cdot k$$

Where: - R_{belt} — belt/pulley reduction ratio - L — ballscrew lead (m) - η — ballscrew mechanical efficiency (approximately 0.90 for rolled ballscrews) - T_{motor} — available motor torque at operating speed (N·m)

Linear speed:

$$v_{linear} = \frac{n_{motor}}{R_{belt}} \cdot L$$

A Note on Belt Drive Compliance

The belt-and-ballscrew architecture is a design tradeoff we chose deliberately. The belt reduction allows a smaller, lower-cost motor to deliver high thrust from a compact package — the RLA-200 produces 2,000 lbf from a frame only 111 mm square. A direct-drive motor coupled to a ballscrew without a belt would eliminate the compliance (slight elasticity) the belt introduces, but would require a significantly larger motor and housing for the same thrust.

For web guiding, this tradeoff is favorable. Web guiding corrections are slow (typically under 1 in/s) and do not require the zero-backlash stiffness of a CNC axis. The belt's compliance is well within the tolerance of the control loop. If your application requires nanometer-level positioning or extremely high dynamic stiffness — uncommon in web guiding — a direct-drive actuator may be more appropriate.

The Speed-Torque Relationship

Stepper motors and BLDC motors share a characteristic: torque decreases as speed increases, primarily due to back-EMF reducing the effective voltage driving current through the windings.

At low speeds, the motor delivers its maximum torque. As speed increases beyond a threshold that depends on supply voltage and winding inductance, torque falls. The actual deliverable thrust at any operating speed is:

$$F_{\mathrm{actual}}(v) = T_{\mathrm{motor}}(n) \cdot k$$

Where $T_{motor}(n)$ is the motor torque at the speed n corresponding to the desired linear velocity.

The RLA actuators use stepper motors with intelligent microstepping drivers that optimize current delivery across the speed range. This extends the usable torque curve compared to basic constant-voltage drivers, but the fundamental speed-torque tradeoff still applies.

What this means for sizing: The rated thrust values in the comparison table below are continuous-duty values at rated speed. The actuator can produce higher thrust at lower speeds, and lower thrust at higher speeds. Our actuator sizing calculator applies the full speed-thrust curve for the specific model and operating conditions. The methodology in this article gives you the force-side analysis; the calculator handles the actuator-side verification.

Continuous vs. Peak Ratings and Thermal Limits

Actuator thrust ratings are not all defined the same way across manufacturers. For the RLA Series:

- **Rated thrust** is the continuous-duty thrust the actuator can sustain indefinitely at rated speed without exceeding the motor's thermal limits (typically 80°C winding temperature rise at 40°C ambient).
- **Peak thrust** — available at low speeds and low duty cycles — can be 1.5–2× the rated value, but thermal limits restrict the duration.

Web guiding is inherently a low-duty-cycle application. The actuator moves only when the web wanders, and corrections are typically small, brief motions separated by dwell periods. This thermal profile is favorable for electric actuators because the motor has time to dissipate heat between corrections. However, during initial roll threading or during a sustained disturbance (roll eccentricity, for example), the actuator may run continuously for an extended period. Size for continuous thrust to ensure reliability in these sustained-demand scenarios.

Competing Approaches — Where Electric Actuators Fit

Electric ballscrew actuators are not the only option for moving a shifting stand. Understanding the alternatives helps you choose the right tool for the job.

Hydraulic cylinders are common on very heavy stands (30,000+ lb) where thrust requirements exceed what a compact electric actuator can deliver. They provide extremely high force density and are robust in dirty environments. The tradeoffs are hydraulic infrastructure (pump, reservoir, filtration, piping), potential for leaks near the web, and the need for proportional valves and closed-loop control hardware to achieve the positioning precision web guiding requires.

Pneumatic cylinders are sometimes used on lighter stands. They are simple and inexpensive but difficult to control precisely — air is compressible, so achieving smooth, proportional positioning requires proportional valves and often secondary feedback. They are generally not suitable for precision web guiding.

Direct-drive electric actuators (motor coupled directly to the ballscrew without a belt or gearbox) offer the highest stiffness and zero backlash. The tradeoff is size: achieving 1,000+ lbf thrust without a reduction stage requires a large, high-torque motor and a correspondingly large actuator body. These are excellent where stiffness is paramount and space is available.

The RLA Series occupies a specific niche: high thrust in a compact frame, purpose-built for the speed and precision demands of web guiding, where the belt-drive compliance is well within tolerance and the compact packaging matters for integration with standard shifting stands.

Worked Example: Paper Mill Unwind Stand

Let us size an actuator for a realistic installation using the complete force model.

Given:

Parameter	Value
Maximum roll weight	10,000 lb (4,536 kg)
Shifting stand weight	3,000 lb (1,361 kg)
Total moving mass (m_{total})	13,000 lb (5,897 kg)
Bearing type	Profile linear rail, μ = 0.004 catalog
Web width	120 inches
Web tension	4 PLI (480 lbf total)
Maximum steering angle	1.5°
Desired correction speed	0.5 in/s (12.7 mm/s)
Desired acceleration	2 in/s² (0.051 m/s²)
Umbilical complexity	Moderate (signal cables + 2 pneumatic hoses)
Floor slope (measured)	0.3° along travel axis
Installation quality	Typical ($k_{install}$ = 3)

Step 1 — Bearing Friction (catalog)

$$F_{\text{friction}} = \mu \cdot m_{\text{total}} \cdot g = 0.004 \times 5{,}897 \times 9.81 = 231\ \text{N} = 52\ \text{lbf}$$

Step 2 — Inertia

$$F_{\text{inertia}} = m_{\text{total}} \cdot a = 5{,}897 \times 0.051 = 301\ \text{N} = 68\ \text{lbf}$$

Step 3 — Web Tension Lateral Force

$$F_{\text{tension}} = 2 \cdot T_{\text{web}} \cdot \sin(\alpha) = 2 \times 480 \times \sin(1.5°) = 2 \times 480 \times 0.02618 = 25.1\ \text{lbf} = 112\ \text{N}$$

Step 4 — Umbilical Drag (estimated)

$$F_{\text{umbilical}} \approx 25\ \text{lbf} = 111\ \text{N}$$

Based on moderate hose configuration — two 3/8" pneumatic hoses plus signal cables in a cable carrier.

Step 5 — Floor Grade

$$F_{grade} = m_{total} \cdot g \cdot \sin(\theta) = 5{,}897 \times 9.81 \times \sin(0.3°) = 5{,}897 \times 9.81 \times 0.00524 = 303 \text{ N} = 68 \text{ lbf}$$

Step 6 — Misalignment

$$F_{misalignment} = (k_{install} - 1) \times F_{friction} = (3 - 1) \times 231 = 462 \text{ N} = 104 \text{ lbf}$$

Step 7 — Total Force

Force Component	lbf	N	Percentage
Bearing friction (catalog)	52	231	12.8%
Inertia	68	301	16.7%
Web tension lateral	25	112	6.2%
Umbilical drag	25	111	6.2%
Floor grade	68	303	16.7%
Misalignment	104	462	25.6%
Subtotal (F_{total})	**342**	**1,520**	**—**

Note what the complete model reveals: the two textbook terms (friction + inertia) account for only **120 lbf** — barely a third of the actual force budget. The "hidden" forces — misalignment, floor grade, umbilical drag, and web tension — contribute the remaining 222 lbf. This is why simplified models require inflated safety factors.

Step 8 — Apply Safety Factor

Because the floor slope was measured and the installation quality is reasonably characterized, but the umbilical drag is estimated:

$$F_{required} = F_{total} \times \text{FoS} = 1{,}520 \times 1.75 = 2{,}660 \text{ N} = 598 \text{ lbf}$$

Step 9 — Select Actuator

Candidate	Rated Thrust	Margin over $F_{required}$	Speed at Rated Thrust
RLA-050	500 lbf	Insufficient (?16%)	1.3 in/s
RLA-100	**1,000 lbf**	**+67%**	**1.3 in/s**
RLA-200	2,000 lbf	+234%	1.1 in/s

The RLA-100 at 1,000 lbf continuous thrust provides 67% margin above the calculated requirement. This is a comfortable margin that accommodates degradation over time and occasional worst-case conditions.

Notice what happened: a simplified two-term model with FoS = 2.0 would have computed $F_{required}$ = 240 lbf and suggested the RLA-050 at 500 lbf with "76% margin." The complete model shows the actual requirement is 598 lbf — which **exceeds** the RLA-050's rating. The simplified model would have led to an undersized actuator. This is a real failure mode we have seen in the field.

RLA Series Model Comparison

Specification	RLA-050-xxxx-TG	RLA-100-xxxx-TG	RLA-200-xxxx-TG
Rated thrust (continuous)	500 lbf (2,224 N)	1,000 lbf (4,448 N)	2,000 lbf (8,896 N)
Rod diameter	32 mm	40 mm	60 mm
Frame cross-section	60 × 64 mm	75 × 75 mm	111 × 111 mm
Available strokes	4", 6", 12"	4", 6", 12"	4", 6", 12"
Maximum speed	1.3 in/s	1.3 in/s	1.1 in/s
Supply voltage	24/48 VDC	24/48 VDC	24/48 VDC
Drive architecture	Ballscrew + belt reduction, intelligent microstepping driver with sensorless stall detection		
Mounting options	Pivot, rod eye, clevis		
Designed for	Roll-2-Roll Technologies shifting stands and SCU6x web guide controllers		

Rated thrust values are continuous-duty at rated speed and 40°C ambient. Peak thrust at low speed and low duty cycle can be 1.5–2× higher. Speed at full rated thrust — see the sizing calculator for the complete speed-thrust curve for each model.

Practical Sizing Recommendations

Measure whenever possible. A spring-scale pull test on the loaded stand — with all umbilicals connected and the web threaded — gives you $F_{friction} + F_{misalignment} + F_{umbilical}$ in a single measurement. This collapses three estimated terms into one measured value and dramatically improves confidence in the calculation.

Do not neglect floor grade. A precision level on the rail surface takes five minutes and can reveal the single largest force in the system.

Web tension lateral force scales with tension, not web weight. A light film under high tension generates more lateral force than a heavy paper web under low tension. Always check this term.

Size for the worst case. Maximum roll weight, maximum tension, moving uphill (if applicable). The actuator should handle the hardest correction, not the average one.

Use the sizing calculator for actuator-side verification. This article gives you the methodology to determine the force your installation requires. The actuator sizing calculator maps that force against the specific RLA model's speed-thrust curve, thermal capacity, and stroke length to confirm the selection.

Summary

Sizing an actuator for a shifting-stand web guide requires accounting for six categories of force: bearing friction, inertia, web tension lateral load, umbilical drag, floor grade, and installation misalignment. Simplified two-term models systematically underestimate the real-world force budget and compensate through inflated safety factors that obscure the actual engineering.

The complete model presented here will not give you a single, tidy number. Real installations have uncertainty in several terms. But by modeling every known force explicitly and reserving the safety factor for genuine unknowns, you arrive at a selection you can defend — and troubleshoot if something is not right.

If you have questions about sizing an RLA actuator for your application, or if your force calculation falls between models, contact our engineering team. We are happy to review your specific installation parameters.