# Designing Drive Systems for Low-Web-Speed Applications

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This paper will look at the solutions that current drive and motor technology can offer in not only improving tension control but also solving several other issues that are inherent to systems with high ratio gearing.

## Design Considerations

***Tension control modes.*** Center-driven winders can be operated in one of three modes of tension control:

1. Indirect torque control
2. Torque-limiting control
3. Speed set-point correction control

*Indirect tension control* is technically the simplest, but least accurate of the modes. Indirect control does not make use of a tension feedback sensor. The tension control is open-loop, directly based on the tension set-point factored by a measured or calculated diameter. Inertia and friction compensation can be a control feature of this mode.

*Torque-limiting tension control* is closed-loop, based on the tension set-point reference, factored by the actual diameter and compensated by the actual tension error through a PI (tension) control loop as a rotational force. Inertia and friction compensation are normally a control feature of this mode.

*Speed set-point correction tension control* is closed-loop, based on the actual web speed reference, factored by the actual diameter and compensated by the transformed tension error through a PI (tension) control loop as a speed set-point addition. Inertia and friction compensation can be a control feature of this mode.

***Determining tension control mode.*** The selection of the winder tension control mode can be influenced by several factors, including machine specifications or design, type of tension sensor used, if any; but normally, the major determining factor will be the compliance of the web material being transported.

The modes of torque control are commonly adapted when the web material has a very low compliance or is "non-extensible;" e.g.—heavy paper, steel, aluminum or other metals. With a non-compliant web the increased system gain of

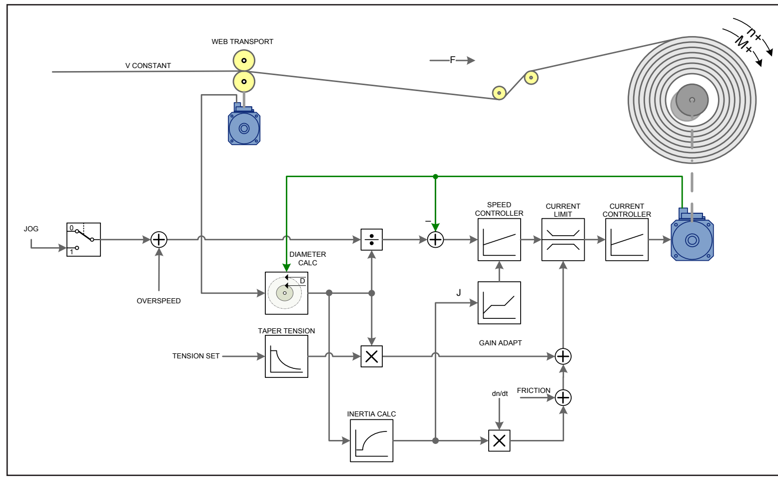


**Figure 1** Indirect torque control.

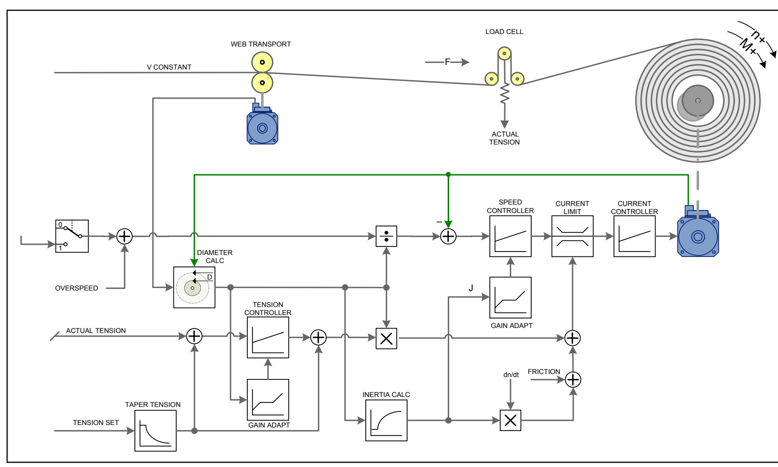


**Figure 2** Torque-limiting control (with load cell sensing).

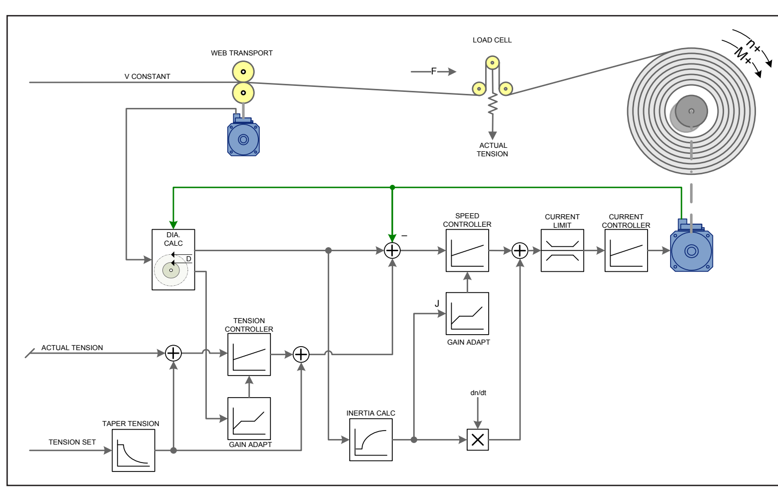


**Figure 3** Speed set-point correction control (with load cell sensing).

a speed-controlled system will make the system tend towards instability and much more complex to optimize tuning. The current or torque-regulated modes of control tend to be more stable over a wider range of conditions. When implementing the torque modes consider torque limiting with tension feedback if accuracy is important.

The mode of tension regulation via speed control is ideally implemented when the web is highly compliant or "extensible." In this mode, the added system gain from additional control loop (speed controller) and transformation from force (load cell) to speed (speed loop) to torque (current loop) is a help in meeting the additional system demands.

***Drive-sizing.*** In the engineering and design of web-handling drive systems the traditional practice is to optimize the drive and motor sizes as close as possible to the web power requirement. This is accomplished by selecting a mechanical gear ratio that will enable the motor to run as close as possible to its base speed in a web-transporting-driven section (non-winder), and well into the constant power range for center-driven wind or unwind (when an induction motor is used).

The optimum gear ratio is determined by:

- Optimum motor speed rpm/maximum load speed rpm

For a driven section with a fixed diameter roll, the power requirement is determined by the maximum torque requirement at maximum speed. The power requirements for a fixed-diameter transporting roll (sectional drive/pull roll) are from the following:

maximum torque (lbf-in) = (driven roll diameter [in.] /2) * maximum web tension [lb.]/(gear ratio * gearing efficiency)

maximum speed (rpm) = maximum web speed fpm/(∏ * driven roll diameter [in.])/12) * gear ratio

The work done by center-driven winds or unwinds is constant power. The torque and speed requirements change throughout the building of the roll. Torque is highest at full roll, with speed the lowest. At the core or smallest diameter, speed is highest and torque is lowest. We can consider the following as the main sizing criteria for a center-driven unwind or rewind:

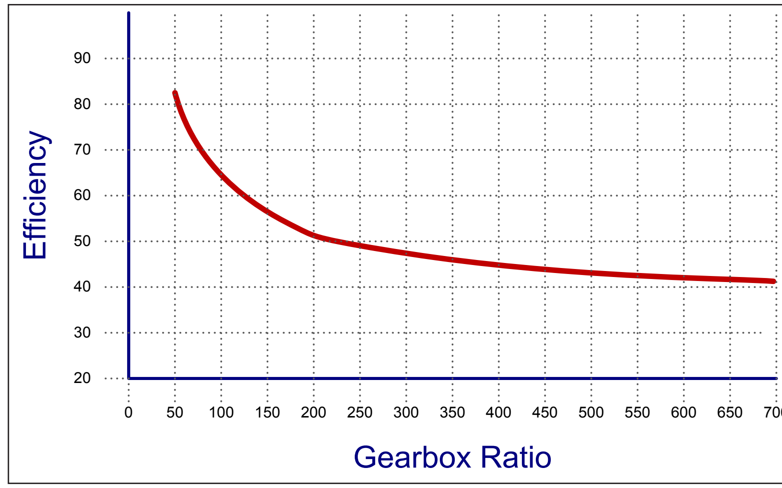


**Figure 4** Typical multistage high-ratio gearbox efficiency.

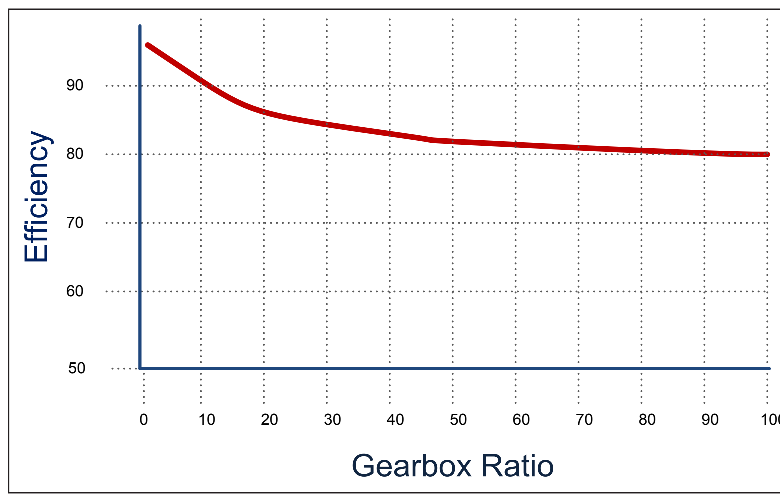


**Figure 5** Typical planetary gearbox efficiency.

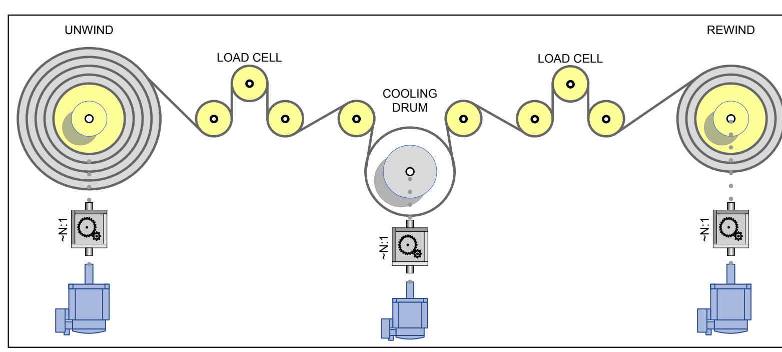


**Figure 6** Example: machine configuration.

**Example Machine Specifications**

| WEB TENSION | | |
|---|---|---|
| Min. | 1.00 | Pli |
| Max. | 10.00 | Pli |

| MATERIAL / SPINDLE | | |
|---|---|---|
| Core | 6.75 | in. |
| Full | 20.00 | in. |

| LINE SPEED | | |
|---|---|---|
| Min. | 0.25 | Fpm |
| Max | 3.00 | Fpm |
| Jog | 12.00 | Fpm |

| COOLING ROLL | | |
|---|---|---|
| Dia. | 42.0 | in. |
| Weight | 2000.0 | lb. |

| WEB WIDTH | | |
|---|---|---|
| Min. | 6.00 | in. |
| Max. | 14.00 | in. |

maximum torque at full roll (lbf-in) = (full roll diameter [in]/2)* maximum web tension (lb)/(gear ratio * gearing efficiency)

maximum speed at core (rpm) = maximum web speed (fpm)/(∏ * core diameter [in])/12) * gear ratio

***Gearbox efficiency.*** The gearbox selection will determine the efficiency factor component of the torque calculations. Gearbox efficiency is simply the ratio of the output power (power transmitted through the gearbox as usable work) to the input power. As no mechanical device is 100% efficient, this numeric value of efficiency will always be less than one. If a speed reducer were 100%—efficient, all torque being applied to the input shaft would be applied to the output shaft.

gearbox efficiency = (actual output torque/ theoretical output torque )*100

Typical gearing options are planetary, helical, worm and pulley/timing belt. Generally, planetary gearboxes are limited to 100:1 ratio, with about a maximum ratio of about ~ 8:1 per stage (although there are exceptions to this rule). Efficiencies for planetary gearing can be considered at ~ 90—95% per stage.

For gear ratios over 100:1 the options are typically multistage worm, helical or combination gearboxes (Figs. 4 and 5).

***Reference machine example.*** Consider a machine with: three driven sections; an unwind; a cooling drum as the master and web transport; and a winder. The cooling drum is the system master and is responsible for transporting the web. The unwind and wind spindles are tension-controlled with tension feedback from load cell transducers. The web material being transported is a stainless steel foil. Since the web is non-extendable the mode of tension control for the spindles will be configured as direct torque control.

Power requirements or web hp from the above specifications would be analyzed in the following manner:

max web hp = (10 pli * 14 in)*12.00 fpm/33,000 = 0.1508 hp

Considering the very low power requirements of this system, and that industry tends to use synchronous servo motors in the drive systems when power requirements are at fractional hp and below, we will consider synchronous servo motors for each axis. This type of motor/drive system is common industry practice. Servo motors in this size range will typically have a rated or maximum speed of 4,500 rpm to 6,000 rpm.

Instead of considering power as the sizing criteria, the following sizing examples will select motors based on the criteria of maximum torque requirements at the full roll maximum tension and the maximum speed requirement at maximum speed at core. Since the master (cooling drum) will be speed-regulated, as well as the system lead section, the following exercises concentrate only on the unwind and rewind spindles.

## System Design: Unwind/Rewind

***Step 1. Determine the spindle gear ratios (for a power optimized system).***

From the max rpm at the load and the motor base speed, we can determine an optimum gear ratio for a power-optimized system as:

***maximum rpm@core*** = maximum fpm/(∏ * [Core Dia. "/12]) = 12.00/ (∏ * [6.75"/12]) = 6.79 rpm

Considering a 4,500 rpm motor, we find an optimized gear ratio as:

4,500 rpm/6.79 rpm = 662.73:1.

For this example we will consider a gearbox ratio of 650:1; this ratio will set the speed at the motor:

***maximum motor speed at core*** (rpm) = maximum web speed/([∏ * core diameter»])/12)* gear ratio fpm/ ([∏ * 6.75»])/12) * 650.00 = 4,413.90 rpm

***minimum motor speed at full roll*** (rpm) = maximum web speed/([∏ * core diameter»])/12) * gear ratio 0.100 fpm/ ([∏ * 20»])/12) * 650.00 = 31.04 rpm

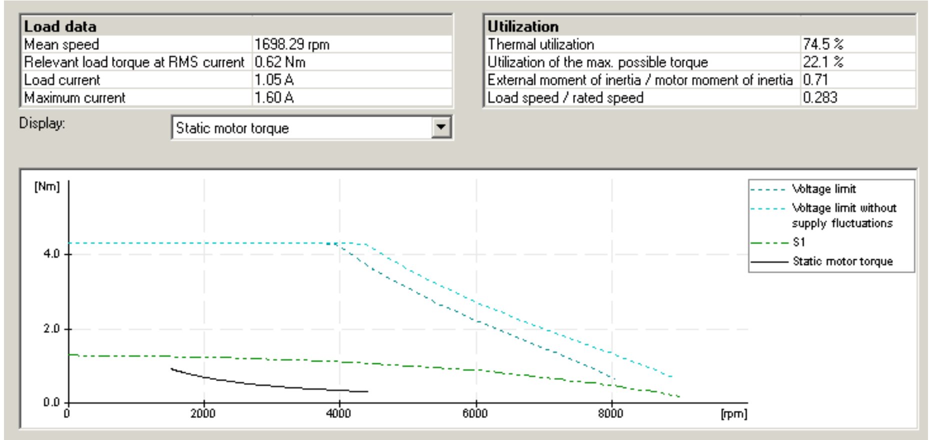


**Figure 7 Motor selection/load requirements.**

Choices for gearboxes in the range of 650:1 are limited and require selection of a multi-stage gearbox to achieve a ratio of that magnitude. We can consider that the typical efficiency of a multi-stage gearbox with a ratio of 650:1 will be in the area of ~ 33 percent.

***Step 2. Determine the spindle torque requirements.***

***maximum torque at core*** (lbf-in) = (core diameter/2)* (maximum web tension/(gear ratio * gear efficiency); (6.75"/2) * (140 lb/(650* 0.33)) = 2.20 lbf-in (Note*: a system with 90% efficiency would require .807 lbf-in at core*)

***Max torque at full roll*** (lbf-in) = ((full roll diameter/2)* (maximum web tension/(gear ratio * gear efficiency)); ((20"/2) * (140 lb/(650* 0.33)) = 6.53 lbf-in

(*Note: a system with 90% efficiency would require 2.39 lbf-in.*)

With these load criteria we can consider a synchronous motor with the minimum of 2.20 lbf-in of torque at the maximum speed at core and 6.53 lbf-in of torque at minimum speed @ full roll. Based on the data, we select a standard motor rated for (1.3 Nm) 11.5 lbf-in at 6,000 rpm (Fig. 7).

## Issues with High-Ratio Gearing

***Isolation: friction losses.*** The gearbox-related friction losses in efficiency, in effect, isolate the motor from the load. We can see the result of that in how the losses in the high-ratio, multi-stage gearbox have increased the mo-

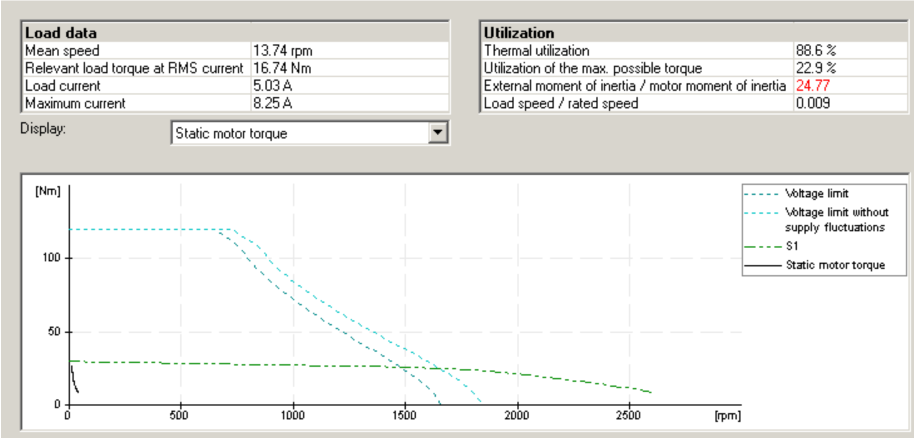


Figure 8 Motor selection/load requirements.

tor torque requirements. It has in fact tripled both the motor and drive size from ~ 2.15 lbf-in to 6.53 lbf-in.

***Active torque control component to friction losses.*** The ratio of the active torque control component to the friction losses should be considered in the design process. Consider that in closed-loop tension control modes only a small component of the tension control signal is active, or supplied, by the tension controller. The majority of the tension set-point is provided from the set-point modified by the actual diameter. Normally we can consider the active portion of the tension component signal at about 5% of the tension torque component. Because the friction losses are not static and will, with time, at a constant speed, also vary with speed and load, the tension control system has to work through the dynamic peaks and valleys of the losses.

The torque required to produce maximum tension without considering loses is 2.15 lbf-in. This computes to an active tension control component of 0.05 * max tension control requirement = .05 * 2.15 in lb = 0.1075 lbf-in.

With friction losses of (6.53 lbf-in – 2.15 lbf-in = 4.38 lbf-in) 4.38 lbf-in, we find that the output related to the losses is over 40 times the magnitude of the active tension control component output. (4.38 lbf-in/0.1075 lbf-in) = 40.74

For effective tension control in the direct-torque-control mode, the active tension control torque signal should be at least equal or greater to the torque required to overcome the gearbox losses.

***Torque ripple-induced tension error.*** Another concern for effective tension control in the torque-regulated modes is the tension error that can be caused from the motor's inherent torque ripple when factored through the gearbox ratio. Synchronous servo motors can have a typical torque ripple from ~ 2.5% to < 1% of M0, or the rated motor torque. In this example the motor selected has a rated torque of 11.5 lbf-in. Considering a torque ripple of 2.5%, the ripple at the motor shaft will be ~ .025 * 11.5 lbf-in = .2875 lbf-in.

This relates to 650/1 * .02875 lbf-in = 186.875 lbf-in on the output of the gearbox and 186.875 lbf-in/10 in. (full roll radius) = 18.687 lb of open=loop tension disturbance on the web at full roll.

Consider also how an oversized motor will add to the open-loop tension error induced by the motor. Additionally, external web tension disturbances will manifest through the system in the same manner.

***Back-driven efficiency.*** High ratio gearing can have a much worse efficiency from the load side in comparison to the input that in effect isolates load changes from output shaft significantly.

***Backlash and compliance.*** Any lost motion between motor and load, be it backlash or compliance, will have negative effects on the control of the load. Compliance can cause mechanical resonance that reduces servo response and renders the system unstable and difficult to tune. Torsional compliance acts as a spring and also causes resonances. The larger the shaft diameter the stiffer or lower compliance it will have. The more mechanical sections, meaning couplings, gearbox stages, etc. in the system, the more compliance and backlash present.

***Dynamic performance.*** There are always disturbances in a system; they can come from torque ripple, out-of-round rolls, tuning, etc. It is possible for a system that is geared to match the lowest web speed to not have enough dynamic response to compensate for the natural disturbance.

***Excess output torque from motor over-sizing.*** As ratios increase, any additional torque in the selected motor size from optimal can raise the issue of too much output torque at the output of the gearbox. The outcome can be machine damage with web jams or web breaks. (In this example the motor torque requirement was 6.53 lbf-in and the selected motor was 10.5 lbf-in).

## An Alternative Solution

Utilizing the lowest possible gearbox ratios, or, if practical, direct-driven motors can help eliminate the issues of load isolation through friction losses, reduce the losses to active tension control component, and reduce torque magnification.

Recommendations for gearing, when required, would be to consider an inline single-stage planetary gearbox or timing belt. When considering a timing belt the limiting factor will be the distance between pulley centers.

Table 1 Encoder resolution: minimum speed regulation

| Encoder Type | Minimum speed (Synchronous Motor) | |
|---|---|---|
| Resolver (16 Bit) | 20.000 | RPM |
| 1024 Pulse HTL (Square Wave) | 10.000 | RPM |
| 2048 Pulse HTL (Square Wave) | 5.000 | RPM |
| 4096 Pulse HTL (Square Wave) | 2.500 | RPM |
| 2048 Pulse Sin/Cos Encoder (22 bit) | 0.250 | RPM |
| 8192 Pulse Sin/Cos Encoder (24 bit) | 0.125 | RPM |

In most cases ratios in the range of 4:1 or less will be the maximum for timing belt gearing arrangements.

High-resolution motor feedback encoders are the key to improving the low-speed regulation of drive systems. With the introduction of the sin/cos optical encoders feedback resolution has been increased from thousands of counts-per-motor-revolution to ~ 4 million-counts-per-revolution for a 22-bit encoder to 16 million-counts-per-revolution for 24 bit encoders. The higher the resolution of the motor feedback sensor, the lower the speed that the drive system can effectively regulate. Table 1 offers a rule of thumb for different encoder technologies and minimum regulated speed.

Considering the same reference specifications on a drive system with a low-ratio, single- stage gearbox with a ratio of 6:1, we determine the following results:

**maximum speed at core (rpm)** = maximum web speed / ((∏ * core diameter")/12) * gear ratio 12.00 fpm/ ((∏ * 6.75")/12) * 6.00 = 40.74 rpm

**maximum speed at full roll (rpm)** = maximum web speed / ((∏ * full roll diameter")/12) * gear ratio 12.00 fpm / ((∏ * 20") / 12) * 6.00 = 13.75 rpm

**minimum speed at full roll (rpm)** = minimum web speed/((∏ * full roll diameter)/12) * gear ratio 0.10 fpm / ((∏ * 20")/12) * 6.00 = .29 rpm

**maximum torque at core (lbf-in)** = ((core diameter/2)* (maximum web tension/(gear ratio * gear efficiency)) ((6.75"/2) * (140 lb/ (6 * 0.96)) = 82.03 lbf-in

**maximum torque at full roll (lbf-in)** = ((full roll diameter/2)* (maximum web tension/ (gear ratio *gear efficiency)) ((20" / 2) * (140 lb/(100* 0.79)) = 243.06 lbf-in

With these load criteria we can consider a synchronous motor with the minimum of 82.03 lbf-in of torque at the maximum speed at core (40.74 rpm) and 243.06 lbf-in of torque at minimum speed @ full roll (0.29 rpm).

Based on the data, we select a standard motor rated for (28 Nm) 247.82 lbf-in of stall torque and (22.5 Nm) 199.15 lbf-in at 2,000 rpm (Fig. 8).

The torque required to produce maximum tension without considering loses is 233.33 lbf-in, giving an active tension control component of 0.05 * maximum tension control requirement = .05 * 233.33 in lb = 11.67 lbf-in.

With friction losses of (243.06 lbf-in – 233.33 lbf-in = 9.73 lbf-in), we find that the losses in this example are less than the magnitude of the active tension control component.

A motor is selected with a rated speed torque of (28 Nm) 247.82 lbf-in. Considering a torque ripple of 2.5% we find the ripple at the motor shaft to be \~ .025 * 247.82 lbf-in = 6.19 lbf-in.

This relates to 6/1 * 6.19 lbf-in = 37.17 lbf-in on the output of the gearbox, and 37.17/10 in (full roll radius) = 3.72 lb. of open-loop tension disturbance on the web at full roll.

Selecting low-ratio gearing with increased motor size offers a drive system with a higher level of inherent accuracy and control dynamics. We have gone from a system that had a loss to active tension control component ratio of > 40:1, to a system with a ratio of less than one. And, a system that inherently imparted disturbances of 18.687 lb. of open-loop tension disturbance on the web at full roll to a system that offers 372 lb. of open-loop tension disturbance on the web, five times less open loop tension disturbance.

## Conclusions

Considering the lowest practical gear ratio or, if practical, direct-driven motors for web handling at very low-web-speeds, can enhance machine tension control performance significantly. This is accomplished by eliminating the issues of load isolation through friction losses, reducing the losses to active tension control component ratio, and reducing the torque ripple magnification.

In systems with lower-to-moderate power requirements, the increased cost of the larger drive system in most cases will be offset by the reduced costs of the system gearing.

## References

1. Gilbert, William. "Motion Control for Converting," Aimcal Fall Conference, Charlotte, NC 2010.
2. Morozov, Denis. "Optimizing Tension Control in Center-Driven Winders," ICE Conference, Orlando FL, 2010.

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