Tension Brakes and Clutches

Design Considerations and Selection

Brakes and clutches used for tensioning (constant slip) have one thing in common. Generally, heat dissipation capacity is the primary criteria for sizing, followed by torque capacity. Beyond this, each has unique sizing requirements that differ greatly. Information on particular Warner Electric tension brakes and clutches start on page 68.

Brakes (Unwinds or Payoffs)

Thermal Requirements

Thermal requirements for a brake equals web HP; which is

$$HP = \frac{\text{Tension (lbs.)} \times \text{Linear Speed (FPM)}}{33{,}000}$$

This energy is constant throughout the unwinding process. Although energy is a function of torque and slip speed, slip speed is at its slowest when torque required is at its greatest (full roll), and slip speed is at its fastest when torque required is at its least (core). All that is needed, then to determine thermal capacity required in an unwind brake is tension and linear speed.

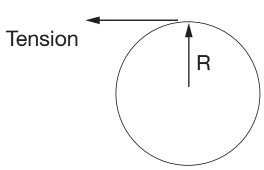
Caution should be taken, however, on machines that run more than one material at different line speeds. All combinations of tensions and line speeds should be checked to insure that brake sizing satisfies the most demanding condition (i.e. – the highest web HP).

Torque Requirements

There are generally three conditions under which a brake must supply sufficient torque: running torque, E-Stop (or emergency stop) torque and controlled stop torque (normal deceleration).

a. Running Torque

This is the torque required to maintain constant tension at any point in the roll being unwound. Since torque is force x distance, with force being tension and distance being roll radius, then torque must change as radius changes if tension is to remain constant. Moreover, the maximum running torque will be at full roll, since that has the largest radius.



b. E-Stop Torque, Web Break

This is the torque required to stop the roll in the event of a web break or a safety related machine stop. There are basically two types of stop conditions to be considered: web break where only the roll inertia stop time and RPM are major considerations, and controlled E-Stop where stopping is required due to some safety related issue, but web tension must be maintained.

During web break E-Stop controlling tension is not a major concern, but getting the roll stopped in a specified time to minimize spillage. The time frame to stop may be a company specification or an OSHA requirement.

For a web break E-Stop, the torque required is a function of roll inertia, roll RPM and E-Stop time requirements.

$$T(\text{torque}) = \frac{WR^2 \times RPM}{308 \times t}$$

where T = Torque (lb.ft.)
t = E-Stop time requirement of machine

Since the roll inertia is greatest when the roll is full, this condition is normally used for calculating the worst-case E-Stop web break torque. RPM can be determined by dividing the linear speed by the roll diameter x pi (3.1416). E-Stop times as short as 2 seconds are not uncommon.

Note that if the control system is open loop (i.e. – ultra-sonic, manual, etc.), maximum E-Stop torque must be obtained by having the S-Stop switch on the machine turn the brake to full on, otherwise the torque available will only be running torque. In the closed loop mode (dancer or load cell), maximum E-Stop torque will automatically be applied.

c. E-Stop Torque, controlled

In a controlled stop, the brake must stop the roll during the time the machine stops, all the while maintaining tension on the unwind roll. This differs from web break E-Stop torque in that the brake must stop the inertia as well as continue to maintain running torque or tension.

$$T = \frac{WR^2 \times RPM}{308 \times t} + \text{Maximum Running Torque}$$

where T = Torque (lb.ft.)
t = E-Stop time requirements of machine

It should be noted that controlled stops can only be accomplished in the closed loop mode, as feedback is required to maintain tension.

For the same stopping times, the controlled E-Stop will require more torque than the web break E-Stop, due to the additional load of maintaining tension. Controlled E-Stop torque is the worst case as the stop is the much faster than normal deceleration times.

E-Stop whether it be for controlled purposes or web break is generally a set function of the machine. Caution should be made in that the faster the E-Stop requirements, the more torque that is required of the system and the more stress that is placed on the components in the machine.

All categories must be investigated to determine the maximum torque capacity required for the application.

Other Considerations

In some instances, it may be desirable to have a gear ratio from the roll shaft to the brake, with the brake on the higher speed shaft. In addition to providing a torque multiplication equal to the gear ratio, this also serves to reduce the effective inertia that the brake sees, as reflected roll inertia is reduced by the square of the ratio. Note, however, that with brakes that have a specified drag, or minimum torque, that drag torque is also multiplied, which could result in inability to address minimum running torque at or close to core diameter.

Also, it is important to realize that employing a gear ratio **DOES NOT** reduce the heat dissipation requirement of the brake.

Another instance where a gear ratio may be needed is when any friction type brake is required to run at very low speeds, usually below 50 RPM. Although today's friction materials have been perfected to the point where static and dynamic coefficients or friction are very close, a certain amount of "sticktion" or stick slip phenomena may occur to the extent that precise control of tension may be compromised. Employing a speed-up gear ratio can make the brake operate at a more efficient speed.

Clutches (Rewinds or Winders)

Although motor drives are the more common choice for winders, clutches can be used quite successfully, and offer a more economical alternative. Typically, the input to the clutch will be a fixed RPM, and can be a take-off from the main machine drive, or an independent motor. RPM input should normally be a least 10% higher than the fastest output. To calculate this, determine the core RPM at fastest line speed, and increase this by at least 10%.

The output of the clutch will start at core RPM, and will gradually decrease as the diameter builds. As in the unwind brake, torque will vary in proportion to the diameter change, but unlike the brake, torque must increase as the diameter builds and the slip speed INCREASES. Slip speed increases because the fixed input RPM doesn't change, but the output RPM keeps decreasing as the roll diameter builds.

Energy dissipation capacity is the most critical sizing criteria in a winder clutch. Creation of heat is highest at full roll, since this is where slip speed AND torque are at their maximum. Maximum heat, or thermal HP, can be found by the following formulae:

$$HP = \frac{\text{Torque(lb.ft.) @ full roll} \times \text{Slip RPM @ full roll} \times 2 \times \text{Pi}}{33{,}000}$$

After the clutch size is selected based on the above thermal calculation, clutch torque capacity should be checked by calculating maximum torque required, which is maximum tension times full roll radius.

Taper Tension

With some materials, taper tension may be required. This is a means by which tension is gradually decreased as the roll diameter builds, and is employed if there is a risk of crushing cores due to build-up of internal pressure within the roll, or if telescoping (slippage to one side) of the wraps might occur. This becomes a function of the control, as the rate of torque increase must be reduced as diameter increases.

In single zone machines, where the unwind brake controls winder tension, taper tension can be handled in a similar fashion.

Control of the clutch can be either open loop (manual adjust or diameter compensation) or closed loop (dancer or load cell), depending upon the degree of precision needed.

For detailed sizing and selection for unwind, intermediate and rewind applications, see sizing selection section on pages 16 through 32.

Design considerations and selection can be broken down by the type of system being selected and the function it must perform. Sizing and application for an unwind will be different than that for a rewind. Also, depending on whether it will be for a clutch, or brake or for a drive, certain system parameters will be required.

Additionally, will the system require a simple remote/analog control, or will it require the option of a closed loop dancer or load cell controller? These factors must taken into consideration when sizing the proper system.

No matter which type of system is being considered, certain application parameters are necessary to make the calculations for selecting the proper components. The selection process is straight forward if the necessary data has been obtained.

An application data sheet should be used for each application to insure the necessary data is available when doing the calculations. In many cases, three or four data sheets may be used for a particular machine. Although this may seem excessive, parameters will often vary between unwind, intermediate, or rewind sections of the machine.

Unwind Sizing Tension Brakes

Once the selection data has been obtained, sizing and calculations can be started. An application example is included for both a brake sizing and a drive sizing, showing the comparison of the two type systems.

Application Data

Material: Paper; 30 lb. Basis weight
Tension: 36 lbs. max.
Roll weight: 1,100 lb. avg.
Web Width: 24 inches
Linear Speed: 800 ft./min.
Core diameter: 3.00 inches
Max. roll diameter: 42.00 inches
Machine Acceleration Time: 15 seconds
Machine Deceleration Time: 15 seconds
Machine E-Stop Time: 3.8 seconds

Note: Tension = Material Tension (PLI) X Web Width

Sizing for a Unwind Tension Brake System

1. **Energy Rate**

   Energy Rate = Tension x Linear Speed

   ER = 36 X 800

   ER = 28,800 ft. lbs./minute

2. **Thermal Horsepower**

   $$\text{Thermal HP} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

   $$\text{HP} = \frac{28{,}800}{\mathbf{33{,}000}}$$

   HP = 0.873 HP

3. **Minimum Roll Speed**

   $$\text{Min. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Max. Roll Diameter (in.)}}$$

   $$\text{Min. Roll Speed} = \frac{800 \times \mathbf{3.82}}{42}$$

   Min. Roll Speed = 72.76 RPM

4. **Maximum Roll Speed**

   $$\text{Max. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Core Diameter (in.)}}$$

   $$\text{Max. Roll Speed} = \frac{800 \times \mathbf{3.82}}{3}$$

   Max. Roll Speed = 1,018.67 RPM

5. **Selection Speed**

   $$\text{Selection Speed} = \frac{\text{(Max. Roll Speed – Minimum Roll Speed)}}{\mathbf{10}} + \text{Min Roll Speed}$$

   $$\text{Selection Speed} = \frac{(1{,}018.67 - 72.76)}{\mathbf{10}} + 72.76$$

   $$\text{Selection Speed} = \frac{945.91}{\mathbf{10}} + 72.76$$

   Selection Speed = 94.591 + 72.76

   Selection Speed = 167.35 RPM (Selection Speed)

   Ref: Appropriate thermal curves on various catalog pages for possible brake selections (Selection Speed vs. Thermal)

6. **Minimum Roll Torque**

   $$\text{Minimum Roll Torque} = \text{Tension} \times \frac{\text{Core Dia (in.)}}{\mathbf{24}}$$

   $$\text{Minimum Roll Torque} = 36 \times \frac{3}{\mathbf{24}}$$

   Minimum Roll Torque = 36 x 0.125

   Minimum Roll Torque = 4.5 lb. ft.

7. **Maximum Roll Torque**

   $$\text{Maximum Roll Torque} = \text{Tension} \times \frac{\text{Max. Roll Dia. (in.)}}{\mathbf{24}}$$

   $$\text{Maximum Roll Torque} = 36 \times \frac{42}{\mathbf{24}}$$

   Maximum Roll Torque = 36 x 1.75

   Maximum Roll Torque = 63.00 lb. ft.

Note: Refer to appropriate Running Torque vs. Speed Curves

Note: Constant values in formulas are in bold.

8. Full Roll Inertia, WR^2

$$\text{Full Roll Inertia} = \frac{\text{Weight} \times \text{Max. Dia. (in)}^2}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}100 \times (42)^2}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}100 \times 1{,}746}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}940{,}400}{\mathbf{1152}}$$

Full Roll Inertia = 1,684.38 lb. ft.^{2}

9. Roll Deceleration Torque (Normal Controlled Stop)

$$\text{Roll Decel Torque} = \frac{\text{Roll Inertia} \times \text{Min. Roll Speed}}{\mathbf{308} \times \text{Machine Decel Time}}$$

$$+ \text{ Max. Running Torque}$$

$$\text{Roll Decel Torque} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 15} + 63$$

$$\text{Roll Decel Torque} = \frac{122{,}555.49}{4{,}620} + 63$$

Roll Decel Torque = 26.53 + 63

Roll Decel Torque = 89.53 lb. ft.

10. Roll E-Stop Torque, Web Break

$$\text{Roll E-Stop Torque, Web Break} = \frac{\text{Roll Inertia} \times \text{Min Roll Speed}}{\mathbf{308} \times \text{Machine E-Stop Time}}$$

$$\text{Roll E-Stop Torque, Web Break} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 3.8}$$

$$\text{Roll E-Stop Torque, Web Break} = \frac{122{,}555.49}{1{,}170.4}$$

Roll E-Stop Torque, Web Break = 104.71 lb. ft.

- This formula can also be used to check tension during acceleration. Using acceleration time of 15 seconds, torque =

  $$\frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 15} = 26.5 \text{ lb. ft.}$$

  Dividing this torque by the radius give tension, so

  $$\text{Tension} = \frac{26.5}{(42/\mathbf{24})} = 15.0 \text{ lbs.}$$

Since tension requirement is 36 lbs., acceleration is OK. If acceleration tension exceeds specified tension, a powered unwind should be considered or changing the time requirements.

11. Roll E-Stop Torque, Controlled

$$\text{Roll E-Stop Torque, Controlled} = \frac{\text{Roll Inertia} \times \text{Min Roll Speed}}{\mathbf{308} \times \text{Machine E-Stop Time}}$$

$$+ \text{ Max. Running Torque}$$

$$\text{Roll E-Stop Torque, Controlled} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 3.8} + 63$$

$$\text{Roll E-Stop Torque, Controlled} = \frac{122{,}555.49}{1{,}170.4} + 63$$

Roll E-Stop Torque, Controlled = 104.71 +63

Roll E-Stop Torque, Controlled = 167.71 lb. ft.

Refer: Appropriate torque vs. speed curves for selection of possible brakes.

Final brake sizing is determined by thermal vs. selection speed and torque vs. speed for both running and E-Stop conditions. These specifications are found in the brake selection sections starting on page 68.

A cross check of minimum running torque to minimum torque of the unit selected must also be made. If the brake minimum torque value is above the minimum running torque value, then either gearing between the unwind roll and the brake will be required, or a larger core diameter or higher tension value must be used.

Note: Not all types of tension brakes in this catalog may be suited for a particular application. Selecting a brake that is not capable of handling the system requirements will result in premature wear out or failure.

If in doubt about sizing and selection, contact your local Warner Electric Distributor, Warner Sales Representative, or the factory.

Note: Constant values in formulas are in bold.

Sizing for an Unwind Tension Drive System

Sizing for an unwind tension drive system is similar to a brake system; however, a few additional calculations are required to insure that the proper motor is selected. As before, the same system data is used to make the calculations and selection.

1. Energy Rate

$$\text{Energy Rate} = \text{Tension} \times \text{Linear Speed} \times \left\{\frac{\text{Max. Dia.(in.)}}{\text{Min. Dia (in.)}}\right\}$$

$$\text{Energy Rate} = 36 \times 800 \times \frac{42}{3}$$

Energy Rate = 36 x 800 x 14

Energy Rate = 403, 200 ft. lbs./minute

2. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{403{,}200.00}{\mathbf{33{,}000}}$$

Thermal Horsepower = 12.22 HP

3. Minimum Roll Speed

$$\text{Min. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Max. Roll Diameter (in.)}}$$

$$\text{Min. Roll Speed} = \frac{800 \times \mathbf{3.82}}{42}$$

Min. Roll Speed = 72.76 RPM

4. Maximum Roll Speed

$$\text{Max. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Core Diameter (in.)}}$$

$$\text{Max. Roll Speed} = \frac{800 \times \mathbf{3.82}}{3}$$

Max. Roll Speed = 1,018.67 RPM

5. Minimum Roll Torque

$$\text{Minimum Roll Torque} = \text{Tension} \times \frac{\text{Core Dia (in.)}}{\mathbf{24}}$$

$$\text{Minimum Roll Torque} = 36 \times \frac{3}{\mathbf{24}}$$

Minimum Roll Torque = 36 x 0.125

Minimum Roll Torque = 4.5 lb. ft.

6. Maximum Roll Torque

$$\text{Maximum Roll Torque} = \text{Tension} \times \frac{\text{Max. Roll Dia. (in.)}}{\mathbf{24}}$$

$$\text{Maximum Roll Torque} = 36 \times \frac{42}{\mathbf{24}}$$

Maximum Roll Torque = 36 x 1.75

Maximum Roll Torque = 63.00 lb. ft.

7. Full Roll Inertia, WR^2

$$\text{Full Roll Inertia} = \frac{\text{Weight} \times \text{Max. Dia. (in)}^2}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}100 \times (42)^2}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}100 \times 1{,}746}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}940{,}400}{\mathbf{1152}}$$

Full Roll Inertia = 1,684.38 lb. ft.^{2}

8. Acceleration Torque to Start Full Roll

$$\text{Acceleration Torque} = \frac{\text{Inertia} \times \text{Min Roll Speed}}{\mathbf{308} \times \text{Machine Accel Time}} + \text{Max. Roll Torque}$$

$$\text{Acceleration Torque} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 15} + 63$$

$$\text{Acceleration Torque} = \frac{122{,}555.49}{4{,}620.0} + 63$$

Acceleration Torque = 26.53 + 63.00

Acceleration Torque = 89.53 lb.ft.

9. Roll Deceleration Torque (Normal Controlled Stop)

$$\text{Roll Decel Torque} = \frac{\text{Roll Inertia} \times \text{Min. Roll Speed}}{\mathbf{308} \times \text{Machine Decel Time}} + \text{Max. Roll Torque}$$

$$\text{Roll Decel Torque} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 15} + 63$$

$$\text{Roll Decel Torque} = \frac{122{,}555.49}{4{,}620} + 63$$

Roll Decel Torque = 26.53 + 63

Roll Decel Torque = 89.53 lb. ft.

10. Roll E-Stop Torque, Web Break

$$\text{Roll E-Stop Torque, Web Break} = \frac{\text{Roll Inertia} \times \text{Min Roll Speed}}{\mathbf{308} \times \text{Machine E-Stop Time}}$$

$$\text{Roll E-Stop Torque, Web Break} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 3.8}$$

Note: Constant values in formulas are in bold.

$$\text{Roll E-Stop Torque, Web Break} = \frac{122{,}555.49}{1{,}170.4}$$

$$\text{Roll E-Stop Torque, Web Break} = 104.71 \text{ lb. ft.}$$

11. Roll E-Stop Torque, Controlled

$$\text{Roll E-Stop Torque, Controlled} = \frac{\text{Roll Inertia x Min Roll Speed}}{\mathbf{308} \text{ x Machine E-Stop Time}} + \text{Max. Running Torque}$$

$$\text{Roll E-Stop Torque, Controlled} = \frac{1{,}684.38 \times 72.76}{\mathbf{308} \times 3.8} + 63$$

$$\text{Roll E-Stop Torque, Controlled} = \frac{122{,}555.49}{1{,}170.4} + 63$$

$$\text{Roll E-Stop Torque, Controlled} = 104.71 + 63$$

Roll E-Stop Torque, Controlled = 167.71 lb. ft.

Not only does horsepower have to be calculated on thermal capacity, but horsepower must also be calculated based on both running and E-Stop torque requirements. In many cases, this will dictate a larger horsepower rating than was previously calculated for thermal capacity.

Generally, most AC and DC motors used with a drive, as is the case with most tension systems, produce 3 lb.ft. of torque over the entire speed range. The drives also provide increased current capacity for acceleration and deceleration for short time periods in the range or 150% of nominal ratings. This translates to a torque rating of 4.5 lb. ft. per horsepower.

12. Horsepower Based on Running Torque

$$\text{Running Horsepower} = \frac{\text{Maximum Running Torque}}{\mathbf{3.0}}$$

$$\text{Running Horsepower} = \frac{63.00}{\mathbf{3.00}}$$

Running Horsepower = 21 HP

13. Horsepower Based on E-Stop Torque

Normally controlled E-Stop torque will be the worst-case conditions for calculating this horsepower requirement.

$$\text{E-Stop Horsepower} = \frac{\text{E-Stop Torque, Controlled}}{\mathbf{3.0 \times 1.5}}$$

$$\text{E-Stop Horsepower} = \frac{167.71}{\mathbf{4.5}}$$

E-Stop Horsepower = 37.27 HP

As can be seen, the horsepower requirements for torque are much higher than those calculated for just thermal capacity. The motor and drive must be selected based on the largest of the three horsepower requirements.

14. Motor HP Comparisons for Thermal and Torque

Thermal HP = 12.22 HP

Running Torque HP = 21.00 HP

Accel/Decel Torque HP = 19.89 HP

E-Stop Torque HP = 37.27

Based on the largest of the three requirements, in this case the E-Stop requirements of 37.27 HP; a 40 HP motor and drive system is required.

Note: Often a service factor will be added that will further increase the motor and drive size. This will generally depend on the severity of the application, environment, etc.

Service factors of 1.25 to 2.5 are typical for most applications.

Sizing and selection for different types of unwind systems, whether they be electric or pneumatic brakes, AC or DC drive systems, is basically the same. Though some differences may exist in the sizing and selection processes, most of the differences are revealed in the actual calculations, which are based on the type of system being considered. Acceleration, deceleration, and E-Stop requirements must be calculated for dancer and load cell type systems.

With analog or manual type systems, sizing process differences are not a factor, as the signal providing the control is a function of roll diameter only, and true machine function feedback is provided.

If deceleration and E-Stop capabilities are necessary to maintain accurate tension, then either a dancer or load cell type system must be considered. These are the only type systems that employ the full closed loop feedback needed for deceleration and E-Stop.

Control systems can be selected from the appropriate tables, page 44.

Note: In some cases a reducer or gearbox may be required between the motor or brake and the unwind roll spindle.

When sizing a reducer or gearbox, the speed is increased by the ratio and the torque is reduced by the ratio. Additionally, the efficiency of the reduction must be taken into account as this will slightly increase the required torque.

Note: Constant values in formulas are in bold.

Intermediate Sizing

Intermediate sizing and selection typically involves a roll that retards or pulls the web to create tension.

A brake usually provides the retarding force, while a clutch driven by a constant speed motor or a variable AC or DC drive system provides pull force.

A few additional parameters are considered in addition to those used in sizing and selecting an unwind.

Application Data

Material: Paper; 30 lb. Basis weight
Tension: 36 lbs. max.
Roll weight: 1,100 lb. avg.
Web Width: 24 inches
Linear Speed: 800 ft./min.
Core diameter: 3.00 inches
Max. roll diameter: 42.00 inches
Machine Acceleration Time: 15 seconds
Machine Deceleration Time: 15 seconds
Machine E-Stop Time: 3.8 seconds
Location of Controlling Element: Nip Rolls, S-Wrap
Roller Diameter: 6.00 inches
Roller Width: 30.00 inches
Roller Weight: 100 lbs.
Nip Roll Pressure: 25 lbs.

Sizing an Intermediate Tension Brake System

1. Nip Roll Speed

$$\text{Nip Roll Speed} = \frac{\text{Linear Speed} \times \mathbf{3.82}}{\text{Nip Roll Diameter}}$$

$$\text{Nip Roll Speed} = \frac{800 \text{ X } \mathbf{3.82}}{6.00}$$

Nip Roll Speed = 509.33 RPM

2. Tension Torque

$$\text{Tension Torque} = \text{Tension} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Tension Torque} = 36 \times \frac{6.00}{\mathbf{24}}$$

Tension Torque = 36 x 0.25

Tension Torque = 9.00 lb. ft.

3. Torque Due to Nip Roll Pressure

$$\text{Nip Roll Torque} = \text{Nip Roll Force} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Nip Roll Torque} = 25 \times \frac{6.00}{\mathbf{24}}$$

Nip Roll Torque = 25 x 0.25

Nip Roll Torque = 6.25 lb. ft.

Note: Constant values in formulas are in bold.

4. Torque Required for Tensioning

Total Torque = Tension Torque – Nip Roll Torque

Total Torque = 9.00 – 6.25

Total Torque = 2.75 lb. ft.

5. Energy Rate Required from Brake

Energy Rate = 2 x Pi X Nip Roll Speed x Nip Roll Torque

Energy Rate = 2 x 3.1415927 x 509.33 x 2.75

Energy Rate = 8,800.59 ft. lbs./minute

6. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{8{,}800.59}{\mathbf{33{,}000}}$$

Thermal Horsepower = 0.267 HP

Initial brake sizing is based on thermal requirements and operating speeds from the appropriate speed vs. thermal curves for the brake type being considered. This information is found in the brake selection section starting on page 68.

7. Normal Deceleration Torque

$$\text{Deceleration Torque} = \frac{\text{Nip Roll Inertia} \times \text{Nip Roll Speed}}{\mathbf{308} \times \text{Machine Deceleration Time}} + \text{Total Running Torque}$$

$$WR^2 = \frac{\text{Nip Roll Diameter}^2 \times \text{Nip Roll Weight}}{\mathbf{1152}}$$

$$WR^2 = \frac{6^2 \times 100}{\mathbf{1152}}$$

$WR^2 = 3.125 \text{ lb.ft.}^2$

$$\text{Deceleration Torque} = \frac{3.125 \times 509.33}{\mathbf{308} \times 15} + 2.75$$

$$\text{Deceleration Torque} = \frac{1591.66}{4620} + 2.75$$

Deceleration Torque = 0.345 + 2.75

Deceleration Torque = 3.095 lb. ft.

8. E-Stop Torque

$$\text{E-Stop Torque} = \frac{\text{Nip Roll Inertia} \times \text{Nip Roll Speed}}{\mathbf{308} \times \text{Machine E-Stop Time}} + \text{Total Running Torque}$$

$$\text{E-Stop Torque} = \frac{3.125 \times 509.33}{\mathbf{308} \times 3.8} + 2.75$$

$$\text{E-Stop Torque} = \frac{1591.66}{1170.4} + 2.75$$

E-Stop Torque = 1.36 + 2.75

E-Stop Torque = 4.11 lb. ft.

Final brake selection is based on running torque and E-Stop torque, based on torque vs. speed curves. The brake must have sufficient torque capability to handle the application. The appropriate curves for the brake type being considered should be consulted.

Note: Not all brake types will be suitable for a given application.

Sizing an Intermediate Tension Clutch System

Clutch sizing for an intermediate tension system is similar to brake sizing except the clutch input speed is recommended to be 50 to 100 RPM higher than the maximum output speed to assure proper controllability.

Using the same parameters as that for the brake sizing, sizing a clutch is as follows:

1. Nip Roll Speed

$$\text{Nip Roll Speed} = \frac{\text{Linear Speed} \times \mathbf{3.82}}{\text{Nip Roll Diameter}}$$

$$\text{Nip Roll Speed} = \frac{800 \times \mathbf{3.82}}{6.00}$$

Nip Roll Speed = 509.33 RPM

2. Tension Torque

$$\text{Tension Torque} = \text{Tension} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Tension Torque} = 36 \times \frac{6.00}{\mathbf{24}}$$

Tension Torque = 36 x 0.25

Tension Torque = 9.00 lb. ft.

3. Torque Due to Nip Roll Pressure

$$\text{Nip Roll Torque} = \text{Nip Roll Force} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Nip Roll Torque} = 25 \times \frac{6.00}{\mathbf{24}}$$

Nip Roll Torque = 25 x 0.25

Nip Roll Torque = 6.25 lb. ft.

4. Total Torque Required for Tensioning

Total Torque = Tension Torque + Nip Roll Torque

Total Torque = 9.00 + 6.25

Total Torque = 15.25 lb. ft.

5 Clutch Input Speed

$$\text{Clutch Input Speed} = \frac{k \times \text{Linear Speed}}{\text{Nip Roll Diameter}}$$

k = 4.2 for 50 RPM Slip Difference

k = 4.57 for 100 RPM Slip Difference

$$\text{Clutch Input Speed} = \frac{4.57 \times 800}{6}$$

$$\text{Clutch Input Speed} = \frac{3656}{6}$$

Clutch Input Speed = 609.33 RPM

6. Energy Rate

Energy Rate = 2 x(Pi) π x Total Torque x Slip Speed Difference

Energy Rate = 2 x 3.1415927 x 15.25 x 100

Energy Rate = 9,581.86 ft. lbs./minute

7. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{9{,}581.86}{\mathbf{33{,}000}}$$

Thermal Horsepower = 0.3 HP

8. Acceleration Torque

$$\text{Acceleration Torque} = \frac{\text{Nip Roll Inertia} \times \text{Nip Roll Speed}}{\mathbf{308} \times \text{Machine Acceleration Time}} + \text{Total Running Torque}$$

$$\text{Acceleration Torque} = \frac{3.125 \times 509.33}{\mathbf{308} \times 15} + 15.25$$

$$\text{Acceleration Torque} = \frac{1591.66}{4620} + 15.25$$

Acceleration Torque = 0.345 + 15.25

Acceleration Torque = 15.595 lb. ft.

Final clutch sizing is based on running torque and acceleration torque requirements that are based on slip RPM between input and output. The appropriate torque vs. speed curves should be consulted to insure that the clutch being considered has the necessary torque capacity for the application. See clutch information starting on page 68.

Not every model of clutch will be suitable for a given application.

Note: Constant values in formulas are in bold.

Sizing an Intermediate Tension Drive System

Sizing a tension drive system for an intermediate tension zone is as easy as sizing a clutch or brake. Often a reducer or gear head will be used between the motor and nip rolls being controlled.

Using the same application parameters as that for the previous brake and clutch, sizing a drive is as follows:

1. Nip Roll Speed

$$\text{Nip Roll Speed} = \frac{\text{Linear Speed} \times \mathbf{3.82}}{\text{Nip Roll Diameter}}$$

$$\text{Nip Roll Speed} = \frac{800 \text{ X } \mathbf{3.82}}{6.00}$$

Nip Roll Speed = 509.33 RPM

2. Tension Torque

$$\text{Tension Torque} = \text{Tension} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Tension Torque} = 36 \times \frac{6.00}{\mathbf{24}}$$

Tension Torque = 36 x 0.25

Tension Torque = 9.00 lb. ft.

3. Torque Due to Nip Roll Pressure

$$\text{Nip Roll Torque} = \text{Nip Roll Force} \times \frac{\text{Nip Roll Diameter}}{\mathbf{24}}$$

$$\text{Nip Roll Torque} = 25 \times \frac{6.00}{\mathbf{24}}$$

Nip Roll Torque = 25 x 0.25

Nip Roll Torque = 6.25 lb. ft.

4. Total Torque Required for Tensioning

Total Torque = Tension Torque + Nip Roll Torque

Total Torque = 9.00 + 6.25

Total Torque = 15.25 lb. ft.

5. Energy Rate

Energy Rate = 2 x (Pi) π x Total Torque x Nip Roll RPM

Energy Rate = 2 x 3.1415927 x 15.25 x 509.33

Energy Rate = 48,803.3 ft. lbs./minute

6. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{48{,}803.3}{\mathbf{33{,}000}}$$

Thermal Horsepower = 1.48 HP

Initial motor selection would be for a 1.5 HP. However, this must be checked to insure that the motor will have sufficient torque capacity to handle the application.

In this application, a ratio between the nip rolls and the motor would be advantageous as it will allow the motor to operate closer to its base speed of 1,750 RPM.

To determine the ratio for the reducer or gear head, assume the maximum motor speed is 1,750 RPM.

7. Reduction Ratio between Motor and Nip Rolls

$$\text{Reduction Ratio} = \frac{\text{Motor Base Speed}}{\text{Nip Roll Speed}}$$

$$\text{Reduction Ratio} = \frac{1750}{509.33}$$

Reduction Ratio = 3.44 : 1

Based on this maximum ratio of 3.44 to 1, a 3:1 ratio would be selected for use between the motor and nip rolls. This would be a standard ratio and would be more readily available in comparison to a 3.44:1 ration.

8. Acceleration Torque

$$\text{Acceleration Torque} = \frac{\text{Nip Roll Inertia} \times \text{Nip Roll Speed}}{\mathbf{308} \times \text{Machine Acceleration Time}} + \text{Total Running Torque}$$

$$\text{Acceleration Torque} = \frac{3.125 \times 509.33}{\mathbf{308} \times 15} + 15.25$$

$$\text{Acceleration Torque} = \frac{1591.66}{4620} + 15.25$$

Acceleration Torque = 0.345 + 15.25

Acceleration Torque = 15.595 lb. ft.

9. Deceleration Torque

$$\text{Deceleration Torque} = \frac{\text{Nip Roll Inertia} \times \text{Nip Roll Speed}}{\mathbf{308} \times \text{Machine Deceleration Time}} + \text{Total Running Torque}$$

$$\text{Deceleration Torque} = \frac{3.125 \times 509.33}{\mathbf{308} \times 15} + 15.25$$

$$\text{Deceleration Torque} = \frac{1591.66}{4620} + 15.25$$

Deceleration Torque = 0.345 + 15.25

Deceleration Torque = 15.595 lb. ft.

Note: Constant values in formulas are in bold.

10. E-Stop Torque

$$\text{E-Stop Torque} = \frac{\text{Nip Roll Inertia x Nip Roll Speed}}{\mathbf{308} \text{ x Machine E-Stop Time}} + \text{Total Running Torque}$$

$$\text{E-Stop Torque} = \frac{3.125 \text{ x } 509.33}{\mathbf{308} \text{ x } 3.8} + 15.25$$

$$\text{E-Stop Torque} = \frac{1591.66}{1170.4} + 15.25$$

$$\text{E-Stop Torque} = 1.36 + 15.25$$

$$\text{E-Stop Torque} = 16.61 \text{ lb. ft.}$$

Because a 3:1 reduction is used between the nip rolls and motor, the reflected torque the motor must produce is reduced by this ratio.

11. Running Torque reflected to Motor with ratio

$$\text{Motor Run Torque}_{\text{(reflected)}} = \frac{\frac{\text{Roll Running Torque}}{\text{Ratio}}}{\text{Efficiency of Reduction}}$$

$$\text{Motor Run Torque}_{\text{(reflected)}} = \frac{\frac{15.25}{3.00}}{0.85}$$

$$\text{Motor Run Torque}_{\text{(reflected)}} = 5.98 \text{ lb. ft.}$$

12. Acceleration Torque reflected to Motor with ratio

$$\text{Motor Accel Torque}_{\text{(reflected)}} = \frac{\frac{\text{Roll Acceleration Torque}}{\text{Ratio}}}{\text{Efficiency of Reduction}}$$

$$\text{Motor Accel Torque}_{\text{(reflected)}} = \frac{\frac{15.595}{3.00}}{0.85}$$

$$\text{Motor Accel Torque}_{\text{(reflected)}} = 6.12 \text{ lb. ft.}$$

13. Deceleration Torque reflected to Motor with ratio

$$\text{Motor Decel Torque}_{\text{(reflected)}} = \frac{\frac{\text{Roll Acceleration Torque}}{\text{Ratio}}}{\text{Efficiency of Reduction}}$$

$$\text{Motor Decel Torque}_{\text{(reflected)}} = \frac{\frac{15.595}{3.00}}{0.85}$$

$$\text{Motor Decel Torque}_{\text{(reflected)}} = 6.12 \text{ lb. ft.}$$

14. E-Stop Torque reflected to Motor with ratio

$$\text{Motor E-Stop Torque}_{\text{(reflected)}} = \frac{\frac{\text{Roll E-Stop Torque}}{\text{Ratio}}}{\text{Efficiency of Reduction}}$$

$$\text{Motor E-Stop Torque}_{\text{(reflected)}} = \frac{\frac{16.61}{3.00}}{0.85}$$

$$\text{Motor E-Stop Torque}_{\text{(reflected)}} = 6.514 \text{ lb. ft.}$$

Note: Constant values in formulas are in bold.

The final selection of the motor is based on the torque/HP capabilities. Motors will normally produce 3 lb.ft. of torque per HP over the speed range when used with either an AC or DC drive. Knowing this, horsepower requirements can be based on the various torque requirements and the motor selected accordingly. Additionally, most AC and DC drives provide a 150% overload capability for a limited time for acceleration, deceleration, and E-Stop conditions.

15. Motor HP based on Running Torque

$$\text{Motor HP} = \frac{\text{Running Torque}}{\mathbf{3.00}}$$

$$\text{Motor HP} = \frac{5.98}{\mathbf{3.00}}$$

$$\text{Motor HP} = 1.99 \text{ HP}$$

16. Motor HP based on Acceleration Torque

$$\text{Motor HP} = \frac{\text{Acceleration Torque}}{\mathbf{4.50}}$$

$$\text{Motor HP} = \frac{6.12}{\mathbf{4.50}}$$

$$\text{Motor HP} = 1.36 \text{ HP}$$

17. Motor HP based on Deceleration Torque

$$\text{Motor HP} = \frac{\text{Deceleration Torque}}{\mathbf{4.50}}$$

$$\text{Motor HP} = \frac{6.12}{\mathbf{4.50}}$$

$$\text{Motor HP} = 1.36 \text{ HP}$$

18. Motor HP based on E-Stop Torque

$$\text{Motor HP} = \frac{\text{E-Stop Torque}}{\mathbf{4.50}}$$

$$\text{Motor HP} = \frac{6.514}{\mathbf{4.50}}$$

$$\text{Motor HP} = 1.45 \text{ HP}$$

19. Motor HP Comparisons for Thermal and Torque

Thermal HP = 1.48 HP

Running Torque HP = 1.99 HP

Accel/Decel Torque HP = 1.36 HP

E-Stop Torque HP = 1.45

20. Minimum Motor Horsepower Selection

Minimum Motor Horsepower Selected = 2.00 HP.

This would be the absolute minimum motor horsepower that would satisfy the requirements for this application.

Note: The 2 HP motor sized does not take into account any type of service factor for the application. Typically a service factor or 1.5 to 2.5 depending on the severity of the application, environment, hours per day operated, etc. are not unrealistic.

By adding a service factor to the final requirements, you can handle any additional friction, drag, etc. that may not be known and can be handled safely. Additionally, this will also help improve the life of the motor and system as well.

Using a service factor of 1.5 in this case, the motor HP would be 2 x 1.5 = 3.00 HP for final motor size selection. This would be much more preferred over using a 2 HP in this particular application.

Rewind Sizing

Rewind tension systems are different from unwind tension systems only in that the material is being rewound on a roll. Many of the calculations are similar. However, rewind tension systems will use either a tension clutch or tension drive.

Selection data required for sizing a tension rewind system is similar to that of an unwind system. The application data form under the rewind section can be used for obtaining the proper data.

For purposes of our application example, the parameters used on the previous unwind and intermediate sections will be used.

Application Data

Material: Paper; 30 lb. Basis weight
Tension: 36 lbs. max.
Roll weight: 1,100 lb. avg.
Web Width: 24 inches
Linear Speed: 800 ft./min.
Core diameter: 3.00 inches
Max. roll diameter: 42.00 inches
Machine Acceleration Time: 15 seconds
Machine Deceleration Time: 15 seconds
Machine E-Stop Time: 3.8 seconds
Taper Tension Requirements: None

Note: Tension = Material Tension (PLI) X Web Width

Sizing for a Rewind Tension Clutch System

1. Energy Rate

$$\text{Energy Rate} = \text{Tension} \times \text{Linear Speed} \times \left\{\frac{\text{Max. Dia.(in.)}}{\text{Min. Dia (in.)}}\right\}$$

$$\text{Energy Rate} = 36 \times 800 \times \frac{42}{3}$$

Energy Rate = 36 x 800 x 14

Energy Rate = 403, 200 ft. lbs./minute

2. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{403{,}200.00}{\mathbf{33{,}000}}$$

Thermal Horsepower = 12.22 HP

3. Minimum Roll Speed

$$\text{Min. Roll Speed} = \frac{\text{Linear Speed} \times \mathbf{3.82}}{\text{Max. Roll Diameter (in.)}}$$

$$\text{Min. Roll Speed} = \frac{800 \times \mathbf{3.82}}{42}$$

Min. Roll Speed = 72.76 RPM

Note: Constant values in formulas are in bold.

4. Maximum Roll Speed

$$\text{Max. Roll Speed} = \frac{\text{Linear Speed} \times 3.82}{\text{Core Diameter (in.)}}$$

$$\text{Max. Roll Speed} = \frac{800 \times \mathbf{3.82}}{3}$$

Max. Roll Speed = 1,018.67 RPM

5. Clutch Input Speed

Clutch Input Speed = Maximum Roll Speed + Slip

Note: Slip Minimum = 50 RPM
Slip Maximum = 100 RPM

Clutch Input Speed = 1018.67 + 50

Clutch Input Speed = 1068.67 RPM

Note: Clutch input speed must be at least 50 RPM greater than the maximum roll speed to provide a slip difference for controlling the output. If a locked rotor condition is used, the slip torque cannot be controlled, especially at core diameter.

6. Slip Speed at Core

Slip Speed at Core = Clutch Input Speed – Maximum Roll Speed

Slip Speed at Core = 1068.67 – 1018.67

Slip Speed at Core = 50 RPM

7. Slip Speed at Full Roll

Slip Speed at Full Roll = Clutch Input Speed – Minimum Roll Speed

Slip Speed at Full Roll = 1068.68 – 72.76

Slip Speed at Full Roll = 995.91 RPM

Thermal selection curves for the appropriate clutches should be checked to insure the clutch chosen can handle the thermal requirements at the worst case slip speed. See clutch information starting on page 68.

In this example, a slip speed of 995.91 RPM and a thermal capacity of 12.22 HP would be checked against the curves to insure that the clutch selected would have sufficient capacity to handle these requirements.

8. Minimum Torque at core

$$\text{Minimum Roll Torque} = \text{Tension} \times \frac{\text{Core Dia (in.)}}{\mathbf{24}}$$

$$\text{Minimum Roll Torque} = 36 \times \frac{3}{\mathbf{24}}$$

Minimum Roll Torque = 36 x 0.125

Minimum Roll Torque = 4.5 lb. ft.

9. Maximum Torque at full roll

$$\text{Maximum Roll Torque} = \text{Tension} \times \frac{\text{Max. Roll Dia. (in.)}}{\mathbf{24}}$$

$$\text{Maximum Roll Torque} = 36 \times \frac{42}{\mathbf{24}}$$

$$\text{Maximum Roll Torque} = 36 \times 1.75$$

$$\text{Maximum Roll Torque} = 63.00 \text{ lb. ft}$$

Once maximum running torque has been determined, refer the appropriate clutch torque curves to insure that the clutch has sufficient torque at the maximum slip speed. Clutch information starts on page 68.

If the clutch selected initially does not have sufficient torque at the maximum slip speed, the next larger size unit should be checked and selected.

Acceleration torque is the final step that must be considered when selecting a clutch for a rewind application. Acceleration torque for starting the roll is in addition to the running torque needed to maintain web tension.

Worst case for acceleration torque occurs when the roll is near its maximum roll diameter. If worst-case conditions can be met, there will be no problems when starting the roll at core diameter.

10. Acceleration Torque at Full Roll

$$\text{Acceleration Torque} = \frac{\text{Full Roll Inertia} \times \text{Full Roll Speed}}{308 \times \text{Machine Acceleration Time}}$$

$$+ \text{Maximum Run Torque}$$

$$\text{Full Roll Inertia} = \frac{\text{Full Roll Weight} \times \text{Max. Roll Dia}^2\text{(in.)}}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = \frac{1{,}100 \times 42^2}{\mathbf{1152}}$$

$$\text{Full Roll Inertia} = 1{,}684.375 \text{ lb. ft.}^2$$

$$\text{Acceleration Torque} = \frac{1{,}684.375 \times 72.76}{\mathbf{308} \times 15} + 63.00$$

$$\text{Acceleration Torque} = \frac{122{,}555.13}{4620} + 63.00$$

$$\text{Acceleration Torque} = 26.527 + 63.00$$

$$\text{Acceleration Torque} = 89.53 \text{ lb. ft.}$$

This torque is required at the maximum slip speed of the clutch to insure the roll can be accelerated while under tension.

As can be seen, the thermal requirements for a rewind clutch are much higher than those required for the same application in an unwind situation.

Generally if the roll build diameter exceeds a 3:1 range, it is more than likely that a clutch will not be sufficient for a rewind application.

If in doubt during the sizing and selection, do not hesitate to contact your Warner Electric Distributor, Warner Electric Sales Representative, or the factory directly.

Sizing for a Rewind Tension Drive System

Sizing a motor for a rewind drive application is almost identical to that of an unwind system.

In this example, tension is constant to simplify sizing. In many applications, taper tension may be required due to the material being processed.

1. Energy Rate

$$\text{Energy Rate} = \text{Tension} \times \text{Linear Speed} \times \left\{\frac{\text{Max. Dia.(in.)}}{\text{Min. Dia.(in.)}}\right\}$$

$$\text{Energy Rate} = 36 \times 800 \times \frac{42}{3}$$

$$\text{Energy Rate} = 36 \times 800 \times 14$$

$$\text{Energy Rate} = 403,\ 200.00 \text{ ft. lbs./minute}$$

2. Thermal Horsepower

$$\text{Thermal Horsepower} = \frac{\text{Energy Rate}}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = \frac{403{,}200.00}{\mathbf{33{,}000}}$$

$$\text{Thermal Horsepower} = 12.22 \text{ HP}$$

3. Minimum Roll Speed

$$\text{Min. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Max. Roll Diameter (in.)}}$$

$$\text{Min. Roll Speed} = \frac{800 \times \mathbf{3.82}}{42}$$

$$\text{Min. Roll Speed} = 72.76 \text{ RPM}$$

4. Maximum Roll Speed

$$\text{Max. Roll Speed} = \frac{\text{Linear Speed X } \mathbf{3.82}}{\text{Core Diameter (in.)}}$$

$$\text{Max. Roll Speed} = \frac{800 \times \mathbf{3.82}}{3}$$

$$\text{Max. Roll Speed} = 1{,}018.67 \text{ RPM}$$

5. Minimum Roll Torque

$$\text{Minimum Roll Torque} = \text{Tension} \times \frac{\text{Core Dia (in.)}}{\mathbf{24}}$$

$$\text{Minimum Roll Torque} = 36 \times \frac{3}{\mathbf{24}}$$

$$\text{Minimum Roll Torque} = 36 \times 0.125$$

$$\text{Minimum Roll Torque} = 4.5 \text{ lb. ft.}$$

Note: Constant values in formulas are in bold.

6. Maximum Roll Torque

Maximum Roll Torque = Tension x $\frac{\text{Max. Roll Dia. (in.)}}{\mathbf{24}}$

Maximum Roll Torque = 36 x $\frac{42}{\mathbf{24}}$

Maximum Roll Torque = 36 x 1.75

Maximum Roll Torque = 63.00 lb. ft.

7. Full Roll Inertia, WR^2

Full Roll Inertia = $\frac{\text{Weight x Max. Dia. (in)}^2}{\mathbf{1152}}$

Full Roll Inertia = $\frac{1{,}100 \text{ x } (42)^2}{\mathbf{1152}}$

Full Roll Inertia = $\frac{1{,}100 \text{ x } 1{,}746}{\mathbf{1152}}$

Full Roll Inertia = $\frac{1{,}940{,}400}{\mathbf{1152}}$

Full Roll Inertia = 1,684.38 lb. ft.^{2}

8. Acceleration Torque to Start Full Roll

Acceleration Torque = $\frac{\text{Inertia x Min Roll Speed}}{\mathbf{308} \text{ x Machine Accel Time}}$

+ Max. Roll Torque

Acceleration Torque = $\frac{1{,}684.38 \text{ x } 72.76}{\mathbf{308} \text{ x } 15}$ + 63

Acceleration Torque = $\frac{122{,}555.49}{4{,}620.0}$ + 63

Acceleration Torque = 26.53 + 63.00

Acceleration Torque = 89.53 lb.ft.

9. Roll Deceleration Torque (Normal Controlled Stop)

Roll Decel Torque = $\frac{\text{Roll Inertia x Min. Roll Speed}}{\mathbf{308} \text{ x Machine Decel Time}}$

+ Max. Running Torque

Roll Decel Torque = $\frac{1{,}684.38 \text{ x } 72.76}{\mathbf{308} \text{ x } 15}$ + 63

Roll Decel Torque = $\frac{122{,}555.49}{4{,}620}$ + 63

Roll Decel Torque = 26.53 + 63

Roll Decel Torque = 89.53 lb. ft.

10. Roll E-Stop Torque, Controlled

Roll E-Stop Torque, Controlled = $\frac{\text{Roll Inertia x Min Roll Speed}}{\mathbf{308} \text{ x Machine E-Stop Time}}$

+ Max. Running Torque

Roll E-Stop Torque, Controlled = $\frac{1{,}684.38 \text{ x } 72.76}{\mathbf{308} \text{ x } 3.8}$ + 63

Roll E-Stop Torque, Controlled = $\frac{122{,}555.49}{1{,}170.4}$ + 63

Roll E-Stop Torque, Controlled = 104.71 + 63

Roll E-Stop Torque, Controlled = 167.71 lb. ft.

11. Horsepower Based on Running Torque

Running Horsepower = $\frac{\text{Maximum Running Torque}}{\mathbf{3.0}}$

Running Horsepower = $\frac{63.00}{\mathbf{3.00}}$

Running Horsepower = 21 HP

12. Motor HP based on Acceleration Torque

Motor HP = $\frac{\text{Acceleration Torque}}{\mathbf{4.50}}$

Motor HP = $\frac{89.53}{\mathbf{4.50}}$

Motor HP = 19.89 HP

13. Motor HP based on Deceleration Torque

Motor HP = $\frac{\text{Deceleration Torque}}{\mathbf{4.50}}$

Motor HP = $\frac{89.53}{\mathbf{4.50}}$

Motor HP = 19.89 HP

14. Horsepower Based on E-Stop Torque

Normally controlled E-Stop torque will be the worst-case conditions for calculating this horsepower requirement.

E-Stop Horsepower = $\frac{\text{E-Stop Torque, Controlled}}{\mathbf{3.0} \text{ x } 1.5}$

E-Stop Horsepower = $\frac{167.71}{\mathbf{4.5}}$

E-Stop Horsepower = 37.27 HP

15. Motor HP Comparisons for Thermal and Torque

Thermal HP = 12.22 HP

Running Torque HP = 21.00 HP

Accel/Decel Torque HP = 19.89 HP

E-Stop Torque HP = 37.27

Note: Constant values in formulas are in bold.

Not only must the motor selected be able to handle the heat dissipation of the application, but it also must be capable of providing the necessary torque to maintain proper tension.

Typically an AC or DC motor controlled by a frequency and/or vector drive, or a regenerative DC drive produces 3 lb.ft. of torque per horsepower over the rated motor speed range.

The HP ratings based on the largest of the 4 conditions of step 15 would be the HP rating selected for the application. In this case, since a 37.27 HP motor is not a standard, the next larger size motor would be selected. This application would require a 40 HP motor and drive system.

In many applications a reduction or gear head would be used between the motor and rewind roll. Often this will reduce the HP rating of the required motor as a torque advantage is realized with the reducer or gear head. It should be noted that the maximum ratio that can be used should never exceed a 30:1 ratio or problems will result at the low-end torque range of the motor possibly.

In the example above, no service factor was taken into account and in many cases a service factor of 1.25 to 2.5 may be considered. This would take into account any unknown friction, bearing drag, etc. in the system.

In this example if a service factor of 1.25 is used, then the motor HP and drive system would be 50 HP. By going to the larger system, motor life and trouble free operation would be realized.

For additional assistance in sizing and selecting a tension rewind drive system contact your Warner Electric Authorized Distributor, Warner Electric Sales Representative, or the factory technical support.

Calculating Web Tensions

For sizing any clutch, brake or drive tension system, tension must be known to perform the calculations. In many cases, the tension ranges for the materials being processed will be known. However, tensions may have to be calculated and/or even estimated for a given application.

To determine an estimated tension value when the actual value is unknown, certain parameters must be known. These are:

1. Material being processed
2. Web width of material, minimum and maximum
3. Paper weights, material thickness or gauge, or wire diameter, or paperboard points

Approximate Tension value = Web Width x Approximate Material Tension

Note: When dealing with film and foil materials, tension values given are normally pounds per mil per inch of material width.

Approximate Tension Values

The values shown are typically for unwind and intermediate tension systems. Values for rewind systems are normally 1.5 to 2 times higher in many cases, especially when dealing with slitter-rewinders.

Tension Value Charts

Material	Tension Pounds per inch of web width
Paper (Based on 3,000 sq. ft. / ream)	
15 lb.	0.50 lb./in.
20 lb.	0.67 lb./in.
30 lb.	1.00 lb./in.
40 lb.	1.33 lb./in.
50 lb.	1.67 lb./in.
60 lb.	2.00 lb./in.
70 lb.	2.33 lb./in.
80 lb.	2.67 lb./in.
100 lb.	3.33 lb./in.
120 lb.	4.00 lb./in.
140 lb.	4.67 lb./in.
160 lb.	5.33 lb./in.
180 lb.	6.00 lb./in.
200 lb.	6.67 lb./in.
Paperboard (Based on points thickness)	
8 pt.	3.00 lb./in.
10 pt.	3.75 lb./in.
12 pt.	4.75 lb./in.
15 pt.	5.63 lb./in.
20 pt.	6.00 lb./in.
25 pt.	9.38 lb./in.
30 pt.	11.25 lb./in.
35 pt.	13.13 lb./in.
40 pt.	15.00 lb./in.
45 pt.	16.88 lb./in.
50 pt.	18.75 lb./in.

Note: Typical tension is 0.375 lbs./point

Material	Tension Pounds per mil of web width
Films and Foils	
Aluminum Foil	0.5 to 1.5 lbs./mil./in. Typically 1.0 lb./mil./in.
Acetate	0.50 lbs./mil./inch
Cellophane	0.50 to 1.0 lbs./mil./in. Typically 0.75 lbs./mil./in.
Polyester	0.50 to 1.0 lbs./mil./in. Typically 0.75 lbs./mil./in.
Polyethylene	0.25 to 0.3 lbs./mil./in.
Polypropylene (Non-orientated)	0.25 to 0.3 lbs./mil./in.
Propylene (Oriented)	0.5 lbs./mil./in.
Polystyrene	1.0 lbs./mil./in.
Saran	0.05 to 0.2 lbs./mil./in. Typically o.1 lb./mil./in.
Vinyl	0.05 to 0.2 lbs./mil./in. Typically 0.1 lb./mil./in.
Mylar	0.5 lbs./mil./in.
Oriented Propylene	0.5 lbs./mil./in.
Metals and Steels	
Beryllium Copper	8.0 lbs./mil./in.
Titanium, Tungsten, High Carbon Steel, and Stainless Steel	8.0 lbs./mil./in.
Low Carbon Steels	See Chart
Non-Ferrous Metals	See Chart

Thickness	Low Carbon Steels (lbs./in. width)	Non-Ferrous Metals (lbs./in. width)
0.005	30.00	22.00
0.010	65.00	42.00
0.015	70.00	59.00
0.020	85.00	70.00
0.025	105.00	80.00
0.030	120.00	90.00
0.035	134.00	98.00
0.040	145.00	105.00
0.045	158.00	110.00
0.050	170.00	115.00
0.055	180.00	120.00
0.060	190.00	125.00
0.065	195.00	130.00
0.070	202.00	135.00
0.075	206.00	139.00
0.080	210.00	142.00
0.085	212.00	146.00
0.090	215.00	150.00
0.095	217.00	152.00
0.100	219.00	155.00
0.110	220.00	
0.120	220.00	
0.130	218.00	
0.140	214.00	
0.150	210.00	

Note: These values are for actual tensions; typically they are run at less.

Wire Tensions

AWG Wire Size	Aluminum Wire	Copper Wire
	Tension Pounds per strand of wire	
30 AWG	0.35	1.2
28 AWG	0.69	2.2
26 AWG	1.10	3.3
24 AWG	1.75	5.0
22 AWG	2.77	7.5
20 AWG	4.42	11.5
18 AWG	7.00	17.0
16 AWG	11.20	26.0
14 AWG	17.80	38.0
12 AWG	28.30	56.5
10 AWG	44.80	81.0
8 AWG	71.40	110.0
6 AWG	113.00	175.0
4 AWG	180.00	278.0

Note: In many cases, only hold back is required rather than full tensioning where there is a permanent set in the material. The actual tension values times a factor of 0.25 to 0.50 is sufficient to provide the necessary holdback.

Material Densities

When the weights of the unwind or rewind rolls are not known, they can be estimated by knowing the roll width, core diameter, maximum roll diameter, material type and material density.

Roll weights can be obtained by looking at the process tracking tags found on most rolls. When this is not possible, an estimated weight can be calculated.

Roll weight must be known to calculate roll inertia for acceleration, deceleration, and E-stop requirements for system selection.

Roll weight = Roll Volume x Material Density

Volume = Max Roll Diameter ^{2} x Roll Width x 0.00045

Note: Maximum Roll Diameter and Roll Width are in inches.

Application Example

Determine the estimated roll weight of a 42 inch diameter roll, 24 inches wide, paper.

Volume = 42^2 x 24 x 0.00045

= 19.05 cubic feet

Weight = Volume x Density

= 19.05 x 57 (Density of Paper)

= 1,086 pounds

Note: This does not take into account the core spindle shaft weight. If an extremely accurate weight of all components is necessary, core spindle shaft weight can be calculated separately and added to the roll weight.

Material Densities

Material	Typical Density (lbs./ft.^{3})
Papers, Films, and Foils	
Paper	57.00-75.00
Paperboard	88.00
Acetate	81.50
Aluminum Foil	45.00
Cellophane	57.00
Polyester	78.00
Polyethylene	57.50
Polypropylene	56.00
Polystyrene	66.00
Vinyl	86.00
Saran	107.50
Mylar	112.00
Metals	
Aluminum	165.00
Beryllium Copper	514.00
Copper	542.00-576.00
Tin	407.50
Titanium	281.00
Tungsten	1,224.00
Steel (typical)	483.00-495.00

Additional Design Considerations

Considerations additional to the sizing process for the controlling device (brakes or clutches) are discussed below.

Torque

Although torque calculations are similar for unwind, intermediate and rewind tension applications, both minimum and maximum torque values of the controlling device must be considered for the application to be successful.

Minimum torque is the amount of force the controlling device must apply to maintain constant tension in the web. If the minimum torque exceeds the minimum torque necessary to maintain web tension, the system cannot control properly, web tension will increase, and waste may result.

Maximum torque is the force provided by the controlling device to maintain proper web tension in worst-case conditions. If maximum torque is less than that required by the application, tension will be less than desirable and may result in poor process.

E-Stop torque is the force the controlling device can apply during machine E-Stop conditions. This E-Stop torque depends on the type of controlling device used and the control system employed. Not all control systems or controlling devices, i.e., brakes, clutches, etc., have E-Stop capabilities. If E-Stop requirements are mandated by the application, then both the controller system and controlling device must have the capabilities to provide this.

If the controlling device cannot produce the necessary torque, then web spillage will occur and damage to machinery may result.

The controlling device must be large enough to cope with all application torque requirements. Even though most brakes and clutches have both static and dynamic torque capabilities, dynamic torque is more important than static torque in tension applications.

Heat Dissipation

When a clutch, brake, or motor operates in a slipping mode or the motor is generating torque, heat is built up as a result of the mechanical energy being converted to thermal energy. The controlling device must be able to dissipate this (heat) energy. If it doesn't, it will fail, either electrically, mechanically, or both.

The heat dissipation capacity of the controlling device must always exceed the heat produced by the application. Environmental considerations must also be analyzed to insure proper operation. High ambient temperature, enclosures surrounding the controlling device limiting the airflow, or marginal heat dissipation capacity have to be considered.

Some controlling devices may need additional cooling with fans or blowers to increase air flow.

The controlling device must be selected properly to handle the application's heat dissipation. This is probably one of the most critical factors in sizing and selection.

Speed

Brakes, clutches, and motors have minimum and maximum speed ranges. Applications must always be checked to insure that the requirements fall within the capabilities of the controlling device.

Failing to operate the controlling devices within their specifications may result in the application failing to meet the specified requirements; failure of the components mechanically and electrically, or even may result in serious damage or injury.

Selection RPM is used to properly size a unit so that over sizing is minimized and an optimum system can be specified.

Inertia

By definition, inertia is that property of a body that makes it continue in the state of motion or rest in which it may be placed until acted upon by some force.

Inertia is an important factor in tensioning applications because it has an effect in the sizing of the controlling device during acceleration, deceleration, and E-Stop conditions.

Failure to consider inertia during the calculations can definitely result in a system being undersized and unable to provide optimum performance. This may result in instability at start up and overrunning during deceleration and stopping. The end result in all cased will be poor product quality and, usually, excessive scrap.

With the exception of intermediate tension applications and analog control systems, inertias are constantly changing in unwind and rewind applications. Worst-case inertia calculations are normally used for sizing and selecting purposes.

Charts

Charts are provided for all clutches and brakes included in the catalog. They provide a means of selecting the correct controlling device for a given application. Performance charts and product specifications for brakes and clutches start on page 68.

The charts provide thermal vs. selection speed data, the means of selecting the unit based on thermal requirements.

Never select a controlling device whose thermal limits are near or equal to those of the application. The next larger size unit should always be considered or the factory should be consulted for additional options.

Selection charts are also provided for running torque vs. speed and E-Stop torque vs. speed. These charts provide a means of checking the preliminary unit selection based on thermal requirements and torques.

The appropriate charts must be used in the sizing and selection process.

Additional Calculations

Additional calculations can be made to determine roll stop time, web pay out during stop, and web storage requirements. These become important when using a dancer or load cell control system to ensure optimum performance and to insure the controlling element selected will do the job.

1. Normal Roll Deceleration Stop Time

Normal Roll Decel Stop time =

$$\frac{WR^2 \times \text{Minimum Roll RPM}}{\mathbf{308} \times [\text{Brake Dynamic Torque available} - \text{Maximum Running Torque (Full Roll)}]}$$

2. Roll E-Stop Time

Roll E-Stop Time =

$$\frac{WR^2 \times \text{Minimum Roll RPM}}{\mathbf{308} \times [\text{Brake Dynamic Torque available} - \text{E-Stop Torque Required}]}$$

Determine web payout during normal deceleration stop and E-Stop conditions to determine the amount of web spillage. The calculations that follow may signal a need to upsize the brake or improve the dancer design.

1. Determining Web Payout during normal deceleration

Web Payout during normal deceleration =

$$\frac{\text{Linear Speed (FPM)} \times \text{Roll Stop time (deceleration)}}{\mathbf{120}}$$

2. Determining Web Payout during E-Stop

Web Payout during E-Stop =

$$\frac{\text{Linear Speed (FPM)} \times \text{Roll E-Stop time}}{\mathbf{120}}$$

3. Machine Web Draw during normal deceleration

Machine Web Draw during deceleration =

$$\frac{\text{Linear Speed (FPM)} \times \text{Machine Decel time}}{\mathbf{120}}$$

4. Machine Web Draw during E-Stop

Machine Web Draw during E-Stop =

$$\frac{\text{Linear Speed (FPM)} \times \text{Machine E-Stop time}}{\mathbf{120}}$$

Once these values are calculated, web spillage can be determined and the brake selected will be found adequate or its size will have to be increased. Another alternative is dancer design improvements. See dancer design section for calculations and suggestions.

Web Spillage = Web Payout of Roll – Machine Web Draw

This should be calculated for both normal deceleration and E-Stop calculations.

Note: Constant values in formulas are in bold.

Note: If the numbers calculated are negative, then no payout or spillage will occur.

Often during E-Stop, web spillage will be evident from the above calculations. If this is not a concern and the brake selected can handle the heat dissipation and torque requirements for running and deceleration, the controlling element has been correctly selected.

It may be necessary with E-Stop requirements, to repeat calculations for torque and brake selection until a controlling element can be selected that will match all the parameters.

Selection Conclusions

No matter which type of tension system is selected, unwind, intermediate, or rewind, this is intended as a general sizing selection guide that will probably cover the vast majority of applications. Some instances will surely be encountered where the sizing and selection covered in the previous pages may not apply. In these cases, your local Warner Electric Representative can provide the necessary guidance and assistance to correctly size and select a tension control system.

The sizing and selection process is quite straightforward, although some work is involved. In summary, sizing and selection can be broken down into three simple steps:

1. Selection of the controlling device, i.e., Brake or clutch
2. Controller, Power Supply, etc., i.e., Remote/Analog, Dancer, Load Cell, or Splicer
3. Input Sensing Element, i.e., Dancer Pot, Load Cell, Analog sensor

With the wide variety of tension products available, Warner Electric can offer complete tension packages for almost any application encountered. Because of its vast experience and knowledgeable professionals, Warner Electric can solve your tensioning needs.

Web Storage

A load cell does not provide material storage for machine acceleration. As the machine draws material during the acceleration period, it is pulling against the inertia of the unwind roll. If the roll is large, the acceleration rate is high, and the material is light, the web may break. Therefore, it may be necessary to provide storage in the web path to release material as the roll comes up to speed. Another option would be to use a drive to help bring the roll up to speed. For further information or assistance, please contact your Warner Electric Distributor or Warner Electric Representative.

Designing the Optimum Dancer Storage System

For closed loop dancer controlled systems, the actual web tension is determined by the downward pressure of the dancer roll or by the loading on the dancer on the web. Consequently, special attention should go into the design of the dancer arm system to provide both consistent tension and adequate web storage for optimum web stop performance.

Load Cell vs. Dancer

Deciding between a load cell and a dancer system requires consideration of many inter-related factors. Sometimes a load cell control is selected when the material being tensioned is not flexible and will not easily wrap around a dancer roll. For example, medium to heavy gauge metals are often tensioned with load cell systems.

Load cell systems can also be selected because of space limitations in the application, or because they are easier to retrofit to existing applications. In retrofit applications, precision balance or rollers may be required if line speeds are greater than 650 feet per minute.

Dancer tension control is still the preferred method of control in many applications. For example, high speed printing applications may require the "forgiveness" of a dancer system to take-up or release material during the dynamically unstable conditions seen at the unwind or rewind roll. The reasons for unstable conditions include fast decelerations or accelerations, out-of-round rolls, and flying splices. A dancer system should be considered when speeds are high and tension control requires extreme precision.

Dancer Roll Design and Construction

The dancer roll and control arms are the heart of this tension control system. Dancer construction is simple, but very important.

For optimum performance, the dancer should be a thin walled tubing and be loaded by massless, low friction air cylinders. A rolling diaphragm device is most commonly used. For greatest accuracy, the wrap on the dancer roll should be exactly 180 degrees.

Anything attached to the dancer for loading will detract from the dancer's ability to act as a buffer and should be made as light and (in the case of air cylinders) efficient as possible.

Construction of Dancer Arms for Webs

Dancer arms should utilize boxed construction to provide rigidity so that the web does not cause the arms to twist. This also insures that the web will track properly over the dancer roller.

The pivot point should be bearing mounted so the dancer arm can move freely. The dancer roller should also be bearing mounted and the bearings should be small in diameter and as frictionless as possible.

This will help reduce the bearing drag and friction changes which affect good tensioning. Standard feed conveyor rollers and bearings are usually sufficient.

Construction of Dancer Arms for Wire

Wire dancers usually employ a single arm. The pivot point and dancer roller should both be bearing mounted to minimize friction and drag. Standard wire rollers are very good dancer rollers for these type systems. These rollers usually contain excellent integral bearings.

Dancer Systems

Dancer Design and Considerations

Warner Electric dancer control systems are designed to control tension in unwind, intermediate, or rewind applications for materials such as paper, foil, films, cloth, metals or wire. The system consists of four parts:

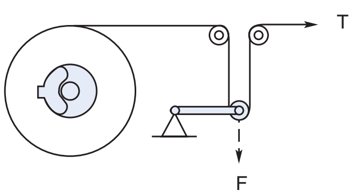
1. The controlling device, i.e. brake, clutch, or drive motor, AC or DC
2. The controller
3. A pivot point sensor which determines the position of the dancer roll
4. The dancer arm and roll assembly (customer supplied)

Dancer Arm Design

Various configurations of dancer arms exist, but their purpose is the same. The dancer provides a means of creating tension on the web by providing a force opposite to the direction the web is pulled.

The effective force applied to the arm to create the desired tension is a function of the number of dancer rollers on the dancer arm.

Single Roll Dancer



F = 2 x N x T

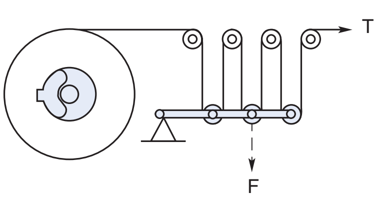
Where:

F = Effective loading force against the web

T = Tension desired in the web

N = Number of dancer rollers

Multiple Roll Dancers



F = 2 x N x T

Where:

F = Effective loading force against the web

T = Tension desired in the web

N = Number of dancer rollers

The more dancer rollers on the dancer arm, the higher the effective force must be to provide the same tension.

Dancer arms should be made of lightweight material to minimize the added effect of weight to the system as well as to keep the inertia as low as possible. Depending on the application and the amount of room available, this will dictate the type of design used and physical size.

The following figures depicting basic dancer designs are intended for guideline only. These are not the only configurations that can be used. Variations on these designs or other designs are acceptable as long as loading and storage requirements can be met.

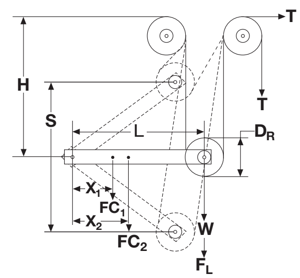


Figure 1 – Horizontal Dancer with Vertical Movement

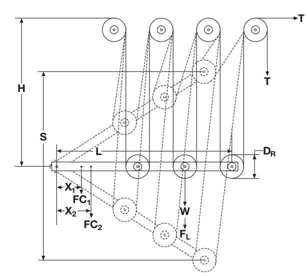


Figure 2 – Multiple Roll Dancer with Vertical Movement

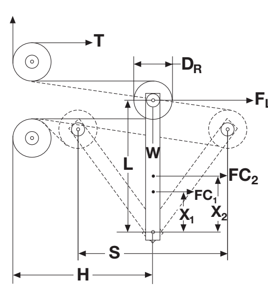


Figure 3 – Vertical Dancer with Horizontal Movement

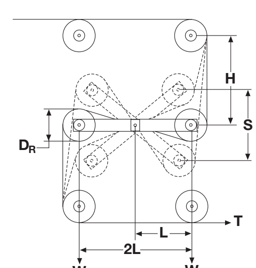


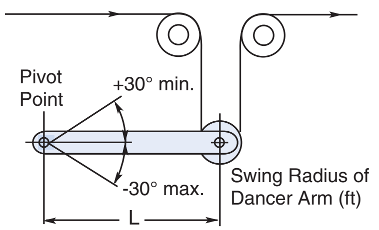
Figure 4 – S-Wrap Dancer with Vertical Movement

Dancer Systems

The following calculations offer a guide for designing a dancer arm. These will provide for an optimum system and for proper loading and storage with the system.

1. Determine Dancer Arm Length, L

This can be done by calculating the length based on the maximum operating linear speed of the system or from the chart below.



a. Calculating Length

$$L = 12 + \frac{\text{Max Web Speed (FPM)} - 200}{100}$$

Minimum L to maximum L should normally be 12" to 40".

b. Chart Determination

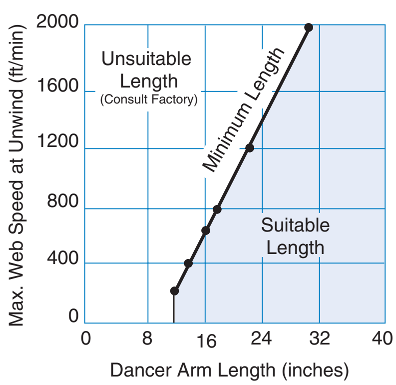


Chart 1 – Dancer Arm Length vs. Web Speed

2. Determine Swing Height of Dancer Arm, S

$S = 1.04 \times L + D_R$

Where:

L = Length of arm calculated or chosen in Step 1.

D_R = Diameter of dancer roller

3. Determine Height from edge of web to centerline of Dancer Pivot Point, H

$$H = \frac{S + D_R}{2}$$

Where:

S = Swing height calculated from Step 2.

D_R = Diameter of dancer roller

Because wide ranges of tensions are required from most systems, some type of loading is usually used to make setting the tension easier. The preferred method is to use a pneumatic cylinder [normally a low inertia, friction less type (Bello-fram) cylinder]. Weights or springs can be used, but these add weight and inertia to the system and are sometimes very difficult to stabilize.

4. Selecting the Loading Point, X

$X_{MIN} = 0.25 \times L$

$X_{MAX} = 0.33 \times L$

Where :

L = Length of the dancer arm

5.* Calculating Cylinder Force Required, F_C

$$F_C = \frac{F \times L}{X}$$

Where:

F = Effective force of the dancer

L = Length of the dancer calculated in Step 1

X = Loading point calculated in Step 4

6. Calculating Cylinder Stroke required

Stroke = 2 x X Tan30 or 1.155 x X

Where:

X = Loading point from Step 4

By following these guidelines, a dancer design with the +/- 30 degree swing will be achieved. This is the range the Warner Electric pivot point sensors require for optimum control performance.

The following chart depicts the percentage of tension variations based on the dancer position in a properly designed dancer.

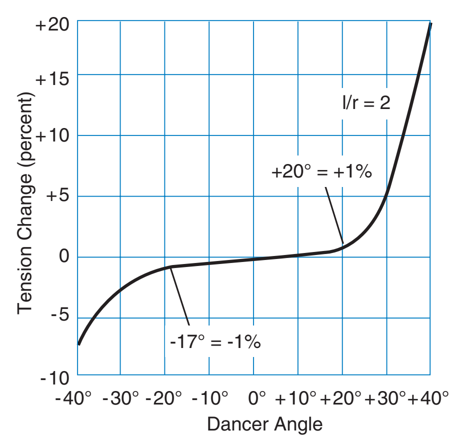


Chart 2 - Tension variation vs. dancer arm angle

* See page 157 for effective cylinder force at a given air pressure.

The following notes are provided for information purposes and should be considered in the design of a dancer arm. Following these guidelines will result in a more optimized system.

I. Horizontal Dancer with Vertical Movement

A. Downward Loaded Dancer

$$\text{Tension} = \frac{\text{Downward Loading Force}}{2 \times \text{Number of Dancer Rolls}}$$

Total Downward loading force at dancer roll =

Downward force created by loading + weight of dancer arm

In this case, the pressure required will be less because the dancer weight adds to the total loading force.

B. Upward Loaded Dancer Arm

$$\text{Tension} = \frac{\text{Upward Loading Force}}{2 \times \text{Number of Dancer Rollers}}$$

Total Upward loading force at dancer roll =

Upward force created by loading - weight of dancer arm

In this case, the pressure required will be greater because the dancer weight subtracts from the total loading force.

II. Vertical Dancer with Horizontal Movement

Dancer weight in this case is no longer a factor on the loading force on the dancer.

$$\text{Tension} = \frac{\text{Loading Force}}{2 \times \text{Number of Dancer Rollers}}$$

Caution must be used when this type dancer and diaphragm type cylinders as the rod assembly is supported by the cylinder bushing only. Secondary support is necessary to keep the cylinder shaft from binding.