Modular Control Units

MCS2000 Series Digital Web Tension Controls

The MCS2000 Web Tension Controller handles all winding, intermediate zone and unwinding applications. MCS2000 easily interfaces to the appropriate clutch/brake driver or motor drive. The digital controller ends the problem of handling large diameter ratios greater than 10:1. See page 46.

- P-I-D parameter programming
- Automatic P-I-D parameter adaption
- Dual outputs in either current or voltage operation modes
- Auto-splice circuit
- Optically isolated I/O
- PLC compatible
- Auto ranging of sensors
- Programmed via hand held programmer or Windows PC program
- Programmable based parameters may be saved on a plug-in memory card
- Multilingual programming
- Usable for unwind/zone/rewind: Electric or Pneumatic Clutches and Brakes, AC, DC, Servo or Stepping Motor Drives.

Analog Controls

TCS Series Analog/Manual Controls

The TCS-200 is a manual analog control for the Electro Disc Tensioning Brake. The control is a constant-current output type that uses a front panel or remote potentiometer to adjust the output. The TCS-200-1/-1H is a manual analog control for any 24 VDC tension brake. It can also accept a 0-10 VDC or 4-20mA analog input for adjusting the output. See page 56.

TCS-200

- Input: 24–30 VAC, 50/60 Hz
- Output: 0-270 mA continuous per magnet up to 12 electro disc magnets, adjustable 3.24 amps
- Torque adjust, brake on, run, brake off switch on front panel
- Remote torque adjust, roll follower inputs

TCS-200-1 Selectable Voltage

- Input: 115/230 VAC, 50/60 Hz
- Output: 0-24 VDC adjustable, 4.25 amps continuous
- Torque adjust, brake on/off, run switch
- Remote torque adjust, roll follower inputs

TCS-200-1H

- Input: 115/230 VAC, 50/60 Hz
- Output: 0-24 VDC adjustable, 5.8 amps continuous
- Torque adjust, brake on/off, run switch
- Remote torque adjust, roll follower, analog voltage or current option

MCS-204 Analog Tension Control

The MCS-204 is a solid-state control designed for manual or analog input to operate one or two 24 VDC tension brakes. It is designed for use with the MCS-166 power supply. See page 57.

- Input 24–28 VDC @ 3 amps
- Operates from torque adjust control knob on front, remote potentiometer, roll follower, or current loop
- Panel mount with exposed wiring or wall/shelf mount enclosure with conduit entrance.

TCS-220 Analog Tension Control

The TCS-220 operates an Electro Disc or other electromagnetic tension brake from an analog input (customer supplied) or the manual setting of the “Torque Adjust” dial on the control face. See page 58.

- Input: 48 VDC. 1.6 amps continuous, 6 amps intermittent. Analog inputs from roll follower or current loop.
- Output per magnet is 0–270 mA running, 270–500 mA stopping
- Cabinet mounting enclosure with exposed wiring or wall/shelf mounting enclosure with conduit entrance.

MCS-208 Analog Tension Control

The MCS-208 operates pneumatic tension brakes through an E to P transducer, which varies air pressure accordingly. Control output is based on an analog input (customer supplied) or the manual setting of the “Torque Adjust” dial on the control face. See page 59.

- Input: 26 VDC. Analog inputs from roll follower or current loop
- Output: 1–9 VDC; 1–5 mA, 4–20 mA, or 10–50 mA, depending on transducer needs
- Cabinet mounting enclosure with exposed wiring or wall/shelf mounting enclosure with conduit entrance.

TCS-320 Analog Splicer Control

The TCS-320 is a solid state splicer control that operates two Electro Disc or other electromagnetic tension brakes, one brake controlling and one brake holding, or two tension brakes operating simultaneously. It can also be used as a dual brake control operating up to 24 MTB brake magnets. See page 60.

- Input: 48 VDC, 3.2 amps continuous, 12 amps intermittent
- Output per magnet is 0–270 mA running, 270–500 mA stopping, 9–90 mA holding
- Available as open frame or with NEMA 4 enclosure

Dancer Controls

MCS-203 Dancer Control

The MCS-203 automatically controls web tension through a dancer roll and sensor. It has 24 VDC output for use with TB, ATTB & ATTC, and Magnetic Particle clutches and brakes. See page 61.

- Operates two 24 VDC tension brakes in parallel when using dual MCS-166 power supplies
- Full P-I-D loop adjustment and system gain adjustment for optimum control.
- Available in open frame or enclosed wall/shelf mount enclosure.

TCS-210 Dancer Control

The TCS-210 automatically controls web tension through a dancer roll and position sensor. It outputs to an Electro Disc or other electromagnetic tension brake. See page 62.

- Input: 48 VDC, 1.6 amps continuous, 6 amps intermittent
- Output per magnet: 0–270 mA running, 270–500 mA stopping
- Cabinet mounting enclosure with exposed wiring or wall/shelf mounting enclosure with conduit entrance.

MCS-207 Pneumatic Dancer Control

This control provides automatic web tensioning using a dancer roll and pivot point sensor. See page 63.

- Operates most pneumatic clutches and brakes
- Automatic control for precise tensioning with minimal operator involvement
- Full P-I-D loop and system gain adjustments for optimum control
- Switch selectable output operates E to P transducers (0–10VDC) or I to P transducers (1–5mA, 4–20mA, 20–50mA) with zero and span adjustments.

TCS-310 Dancer Splicer Control

The TCS-310 is an automatic splicer control that operates two Electro Disc or other electromagnetic tension brakes, one brake controlling and one brake holding, or two tension brakes operating simultaneously. It can also be used as a dual brake control operating up to 24 MTB brake magnets. See page 64.

- Input: 48 VDC, 3.2 amps continuous, 12 amps intermittent
- Output per magnet is 0–270 mA running, 270–500 mA stopping, 0–90 mA holding
- Available as open frame or with NEMA 4 enclosure

Power Supplies

MCS-166 Power Supply Module

The MCS-166 Power Supply Module provides power for the MCS-203, MCS-204, MCS-207, or MCS-208 control modules. See page 65.

- 120V/220V/240 VAC, 50/60 Hz
- 24 VDC, 1.5 amp output
- May be connected in parallel for increased current capacity.

TCS-167 Power Supply

The TCS-167 Power Supply provides power for either the TCS-210 or TCS-220 control modules. See page 65.

- 120V/240 VAC, 50/60 Hz operation, switch selectable
- Output: 9 VDC @ 1.5 amps and 48 VDC @ 1.6 amps continuous, 6 amps intermittent
- Internally fused for protection.
- Available in open frame or enclosed wall/shelf mount enclosure.

TCS-168 Power Supply

The TCS-168 Power Supply provides power to either the TCS-310 or 320 dancer tension controls. See page 65.

- Input switch selectable for 120 or 240 VAC, 50/60Hz
- Output 3.2 amps continuous, 12 amps intermittent

Electric Brakes & Clutches

TB Series Basic Tension

Annular style 24 VDC tension brakes for light to medium duty unwind tension applications.

- Sizes: 1.7" to 15.25" diameter
- Torque range: 0.50 lb.ft. to 256 lb.ft.
- Thermal range: .019 HP to 1.09 HP

ATT Series Advanced Technology

Designed for intermediate web tension ranges. Three size ranges.

- One piece clutch design for easy shaft mounting
- Brakes are flange mounted and the armature is the only rotating member
- Clutch torque ranges 7 to 41 lb.ft. Brake torque ranges from 8 to 62 lb.ft.
- Replaceable friction faces and armature rings.

MTB Series Modular Tension

Modular Tension Brakes (Electro-Disc) are modular caliper type electric brakes used for unwind tensioning. Torque is varied by disc diameter and by changing the number of magnets on the friction disc(s).

- 10", 13", 15" and 20" diameters
- Torque ranges to 1120 lb.ft.
- Thermal capacities to 8 HP
- Brakes rebuildable by changing only friction pads and armature disks.

M Series Permanent Magnet

Permanent magnet brakes and clutches are ideal for light tensioning applications, such as film and fine wires. They require no external power, have a wide range of torque adjustment, have no friction surfaces to wear, and offer chatter-free torque control even at very low speeds.

- Torque range from 1 oz.in. through 65 lb.in.
- Manual torque adjustment
- Constant torque with varying speeds.

Magnetic Particle

Self-contained magnetic particle clutches and brakes for a wide range of unwind/rewind applications offer smooth operation at very low speed and electronic control compatibility.

- Torque range from 2 lb.in. through 578 lb.ft.
- Shaft or flange mounting
- Fan cooled in largest sizes.

Pneumatic Brakes & Clutches

Mistral

Mistral Pneumatic Tension Brakes' compact design meets the special needs of the corrugating industry.

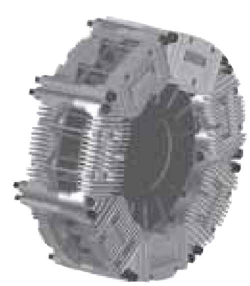
- Fan cooled for longer life
- Three sizes for multiple applications
- Torque range: 1 lb.ft. to 442.5 lb.ft.
- Thermal capacity to 3.5 HP
- Three sizes from 9" to 16" diameter. Eases handling small roll ends.

Magnum

Note: Being Discontinued.

AD Series Air Disc Brakes

Note: Being Discontinued



ModEvo

Modular Pneumatic Tension Brake allows for a wide range of tension applications with the modular design. Actuator configuration with different friction material coefficients allow for much greater range capabilities.

- Torque range from 16 lb.ft. to 3180 lb.ft.
- Optional guards and cooling fan assemblies
- Thermal capacities to 18 HP
- Optional high speed armatures

Sensing Devices

Ultrasonic Sensors

- Analog outputs with selectable 0–10V – 4–20mA
- Input voltage 20–30VDC
- Range control zero and span
- Short circuit protected
- 80" max. distance
- Response time 50 mSec

Pivot Point Sensors

The TCS-605-1 and TCS-605-5 pivot point sensors close the feed back loop to the tension control by sensing dancer roll position.

- TCS-605-1 is a single turn potentiometer with a resistance of 1KΩ for normal dancer operating ranges within 60° of arm rotation.
- TCS-605-2 is a single-turn potentiometer with a resistance of 5KΩ for normal dancer operating within a 60° range used with AC & DC drives.
- TCS-605-5 is a five-turn potentiometer with a resistance of 1KΩ for festooned dancer systems, with a 300° rotational range.

Load Cell Sensors

These devices are used in tension systems to provide closed loop feedback of the actual tension on the web.

FM – Foot Mounted

The foot mounted style load cells (used with pillow blocks) provide easy and convenient mounting to the roll that is being measured. It is a strain gauge style unit that is ideal for heavy tension applications.

- Load ratings: 22, 56, 112, 225, 562, 1122, 2248 lbs.
- Sensitivity (output): 1 mV/V at nominal load
- Power Supply: 10 to 15 VDC

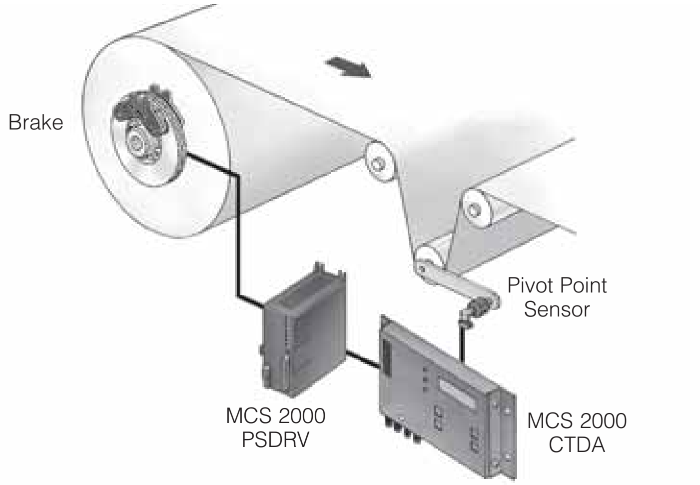
ES – End Shaft Mounted

The end shaft style load cells mount to the end of the roll that is being measured. It is a LVDT (Linear Variable Differential Transformer) style which can withstand overloads up to 10 times its rated load capacity. There are several models offered: dead shaft (no bearing), live shaft and cantilever where a single load cell can be used to measure the tension on the roll. Some units are powered with DC voltage and other units are powered with AC voltage. The AC units offer a price advantage over the DC.

- Load Ratings: 20, 50, 90, 200, 500
- Sensitivity (output): 3VDC at nominal load
- Power Supply: ±12 to ±15 VDC, ±5%

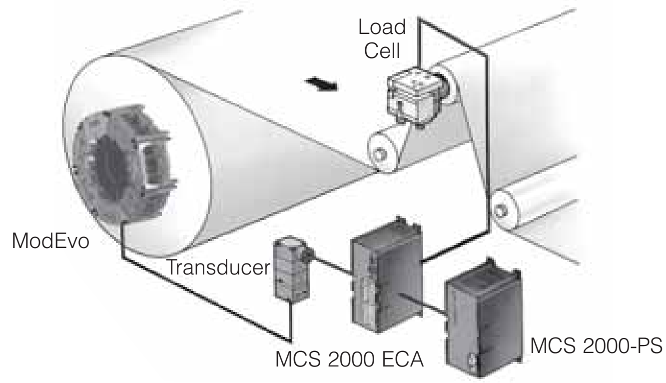
Dancer Control

The dancer control system consists of a power supply, dancer control, pivot point sensor, and controlling element, i.e., tension brake or clutch. Dancers provide the web tension while the control and controlling element stabilize dancer operation for unwind, intermediate zone or rewind tension.



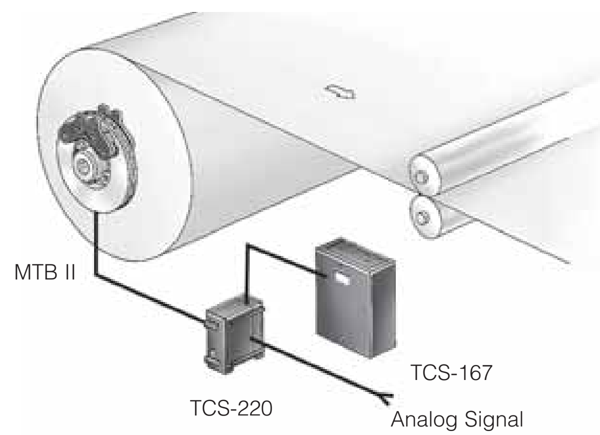
Load Cell Control

Load cell control system consists of the load cell controller, power supply, load cells and controlling element, i.e., tension brake or clutch. Load cells measure the pull force on the web and compare that force to the set point tension in the control. The control increases or decreases the retarding force. Load cells are used for unwind, intermediate zone or rewind tension control.



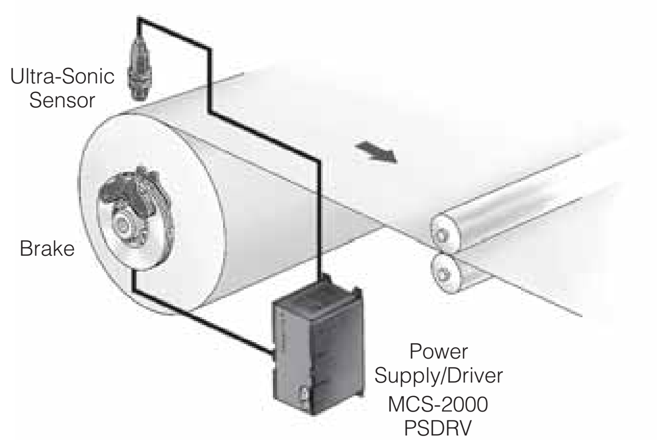
Analog Control

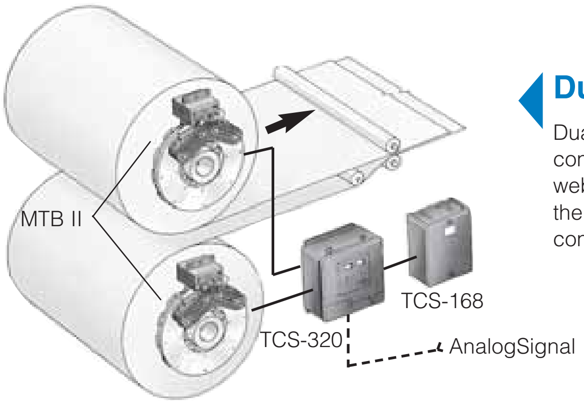
The analog system consists of a control module, power supply, and a controlling element, i.e., tension brake or clutch. The analog controller provides output proportional to the input signal for use in unwind, intermediate zone or rewind tensioning.



Electronic Control

Electronic control systems are very similar to analog control systems with the exception of using an electronic sensing element such as an ultrasonic or photoelectric sensor. The sensor monitors diameter change in either the unwind or rewind rolls, and provides a corresponding change in output.



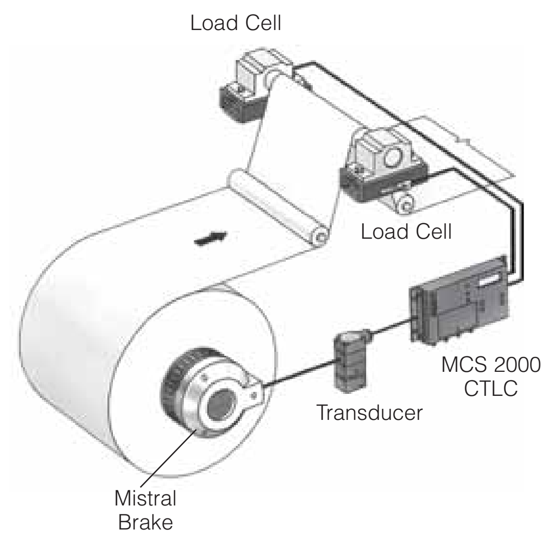


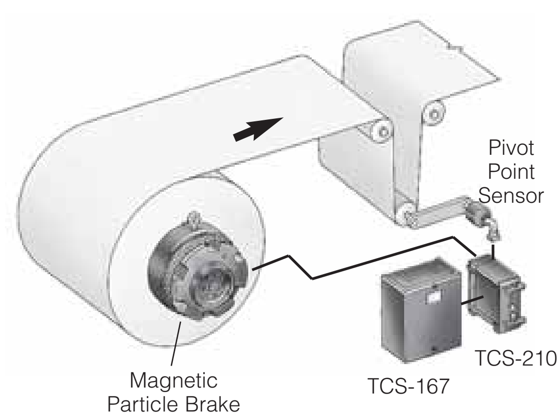
Dual Brake Unwind

Dual brake unwind incorporates modular tension brakes and an analog control system. The brakes retard the unwind roll, creating tension in the web. An external, customer-provided signal adjusts the output current to the brakes to maintain the proper tension. The dual channel controller controls each brake independently or simultaneously.

Single Roll Pneumatic Brake Unwind

Pneumatic brake retards the rewind roll, providing the required tension. Tension is set by the loading force applied to the load cells, which send a signal to the controller. The controller signal to the electric/pneumatic transducer controls the air pressure to the brake.





Single Roll Magnetic Particle Brake Unwind

The magnetic particle brake retards the unwind roll, maintaining tension provided by the dancer roll's weight. The pivot point sensor signals the controller to vary the current to the brake.

Technical Considerations – Tension Zones

I. A tension zone in a web processing machine is defined as that area between which the web is captured, or isolated. Virtually any machine can be broken down into tension zones, and it is important to do so to properly address maintaining the tension required.

Simple machines, such as rewinders or inspection machines, may have only one zone (see Fig. 1). The primary goal here is to control tension so that the rewound package is accurately wound. Typically, the winder (A) would be a simple line speed motor drive, with tension controlled by a brake system at the unwind (D). The method of brake control (i.e.: open or closed loop) would be determined by the accuracy demands of the application. For simple diameter compensation, an ultrasonic sensor measuring the diameter of the roll can produce satisfactory results. Greater accuracy may require closed loop feedback, such as from a dancer or load cell.

II. More commonly, a machine will have driven nip rolls in the center, or processing section (see Fig. 2). A simple slitter/rewinder is an example. In this case, there are two separate tension zones to deal with and the tension levels may be different in each zone. Different tension levels are possible because the web is captured at the driven nip rolls, thus creating separate and distinct unwind and rewind zones. The driven nip rolls (B) will typically be powered by a motor drive that establishes machine line speed. Processing tension will be controlled by a brake system at the unwind (D), and a clutch or motor drive will control the winder tension (A). Again, the method of control will be dictated by the accuracy of tension control required in each zone. If process tension levels can vary by 10% or greater, a simple open loop brake control system may suffice. More accurate control would require a closed loop system, such as dancer or load cell feedback. Likewise, in the winder zone, open loop control may be sufficiently accurate, or closed loop or taper tension control may be required.

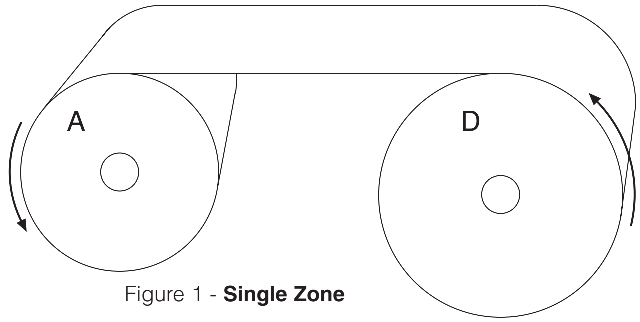


Figure 1 - **Single Zone**

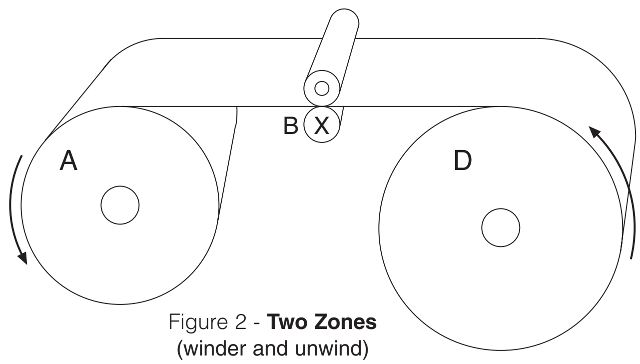


Figure 2 - **Two Zones**
(winder and unwind)

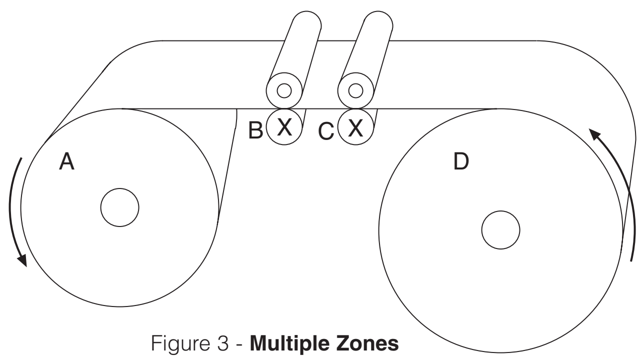


Figure 3 - **Multiple Zones**
(winder, intermediate, unwind)

III. More complex machines will usually have multiple intermediate zones in addition to the unwind and rewind zones (see Fig. 3). One of the intermediate zone drives will typically establish line speed, and the control of drive rolls for the other zones will relate to this drive. In some instances, a simple master/slave relationship with a speed differential ratio will provide the draw tension necessary in that zone (i.e. Fig. 3 – B & C). In other cases, this may be accomplished with closed loop (dancer or load cell) trim. The rewind (A) and unwind (D) would be handled as described in II. Multiple intermediate zones can become very complex, particularly if high degrees of accuracy are required. As a general rule of thumb, control of any zone should be accomplished at one end of the zone only. Control systems at both ends of the zone (for that zone) will generally result in instability of tension levels.

Reliable and accurate control for all system design layouts

Open loop tension control systems provide the least expensive manner to provide a degree of web tension control with the minimal amount of components. Open loop tension control can apply to unwind, intermediate, or rewind tension applications.

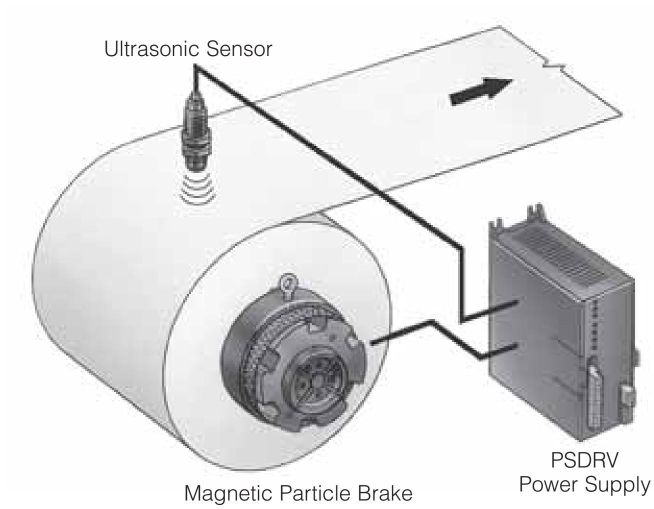
Although not as sophisticated as most closed loop tension control systems, a degree of controllability is achieved. Using open loop tension systems, one does sacrifice such things as web storage for acceleration, deceleration, and E-stop conditions. Tension variations during machine start or stop are common with this type of system.

The most common of the various tension systems are generally comprised of the controlled device; i.e., brake, clutch, etc., a simple controller or power supply, and a controlling element, i.e., a potentiometer or some type of analog sensor.

Because of system simplicity, tension is maintained for diameter compensation only in an unwind or rewind system, and no compensation is provided for acceleration, deceleration, E-stop or out of round roll conditions. Tension variations of 25% or more may be possible during acceleration or deceleration, and 10% or more during running due to out of round rolls or variations in the process machines.

These types of systems lend themselves nicely to applications where tension variations are not a concern, and hold back on a rewind role or scrap wind up is needed. Operator adjustments are usually required when material tensions or roll diameters are changed initially.

Open Loop System



Typical Components

For the simplest of unwind systems, the following components might be used:

- Tension brake coupled to the unwind roll, i.e., ATTB, TB, magnetic particle, or MTB, or pneumatic brake
- Tension controller to provide control current or voltage to the brake, i.e., TCS-200-1, MCS-166/MCS-204, TCS-167/TCS-220, MCS-166/MCS-208
- Control, either the manually adjusted type with a control potentiometer, or through an external potentiometer coupled to a follower arm, or ultrasonic or analog proximity sensor monitoring roll diameter.

Flying Splicer

Specially designed solid state splicer control holds the unused roll stationary while tensioning the operating roll. Dancer variation sensing and subsequent adjustment are virtually instantaneous for accurate tensioning during the splice, typically at less than 1% variation.

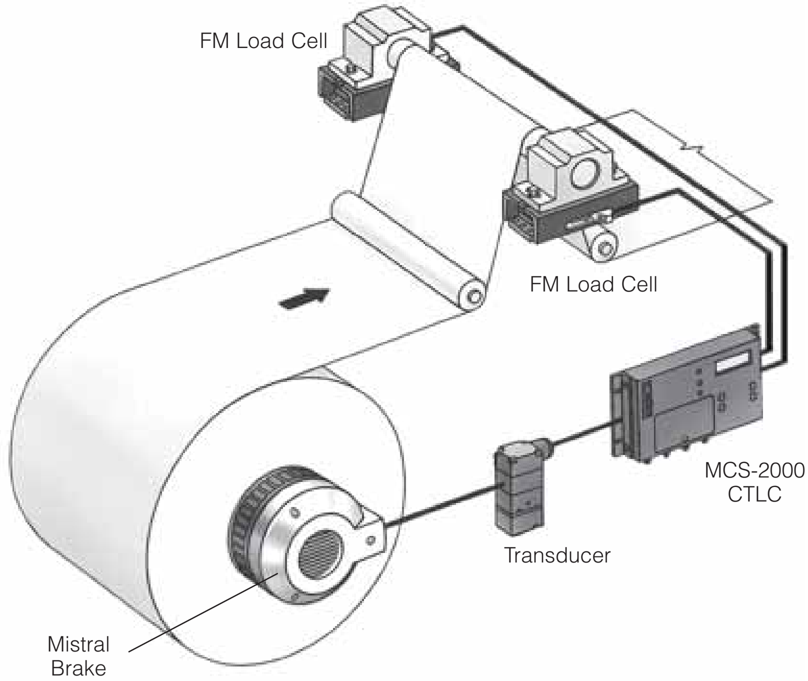
Closed Loop System

Closed loop tension systems provide very precise and accurate tension control during steady state running conditions as well as acceleration, deceleration, and E-stop conditions. Because the material web is monitored constantly, either by load cells or from a dancer by position, changes are detected immediately and the controlled device is changed instantaneously to maintain accurate tension control.

The two most common methods of providing closed loop tension control are via load cells that monitor the force on the web directly or via dancers, which provide tension by the load imposed by the dancer roll and dancer position and velocity are monitored, usually by a precision potentiometer. Even the most minute changes are sensed and compensated for in a closed loop system.

Closed loop tension control systems require the least amount of operator involvement during running. Normally, the operator sets only the tension level required for the material being run, once the system has been properly set up and adjusted. Closed loop system controllers compensate for changes in roll diameter and conditions, acceleration, deceleration, and machine variations.

Although closed loop tension control systems offer the most advantageous method of providing web tension control, be it dancer or load cell, there are some limitations to each type of system. In dancer systems, more space is required in the machine to accommodate the dancer arm and rollers, and some method, preferably an air cylinder and regulator, is required for loading. Load cell systems, on the other hand, require less space for mounting, but storage is non-existent for acceleration or deceleration, and balancing of all machine rollers. Web contact is required because of load cells' high sensitivity.

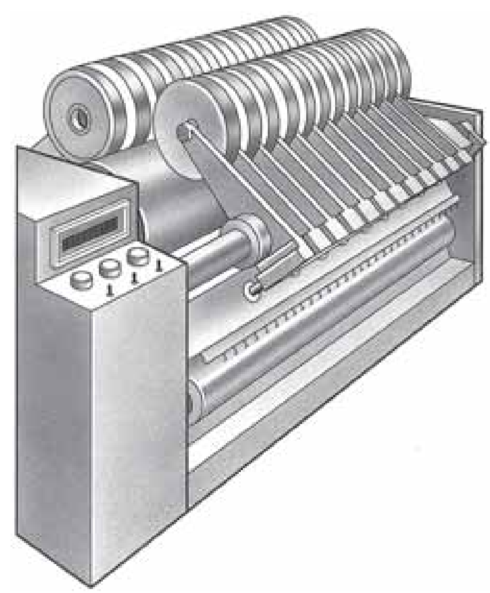


Typical System Components

The typical components of a closed loop tension system are:

- Tension brake coupled to the unwind roll; i.e., TB, MTB, magnetic particle, pneumatic brake
- Controller to provide proper signal to control device; i.e., MCS2000EAC/ MCS2000PSDRV, MCS-166/MCS-203, TCS-167/TCS-210, MCS-166/MCS-207
- Controlling element, either load cell or dancer pivot point sensor potentiometer

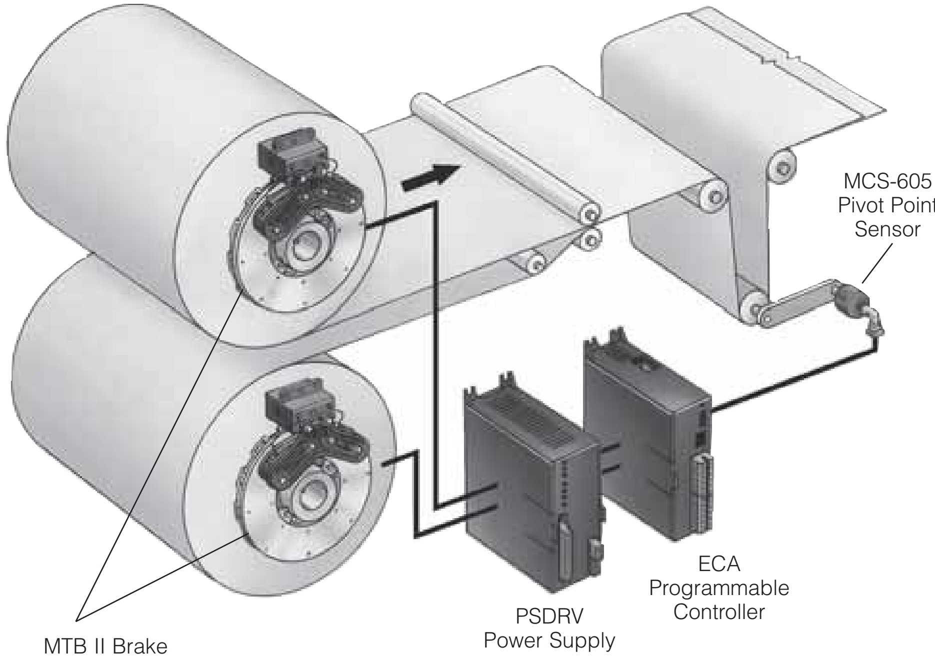
In general, closed loop tension control is the preferred method in more complex machines where precise tension control is required due to process requirements, such as precise registration, multiple color printing or coating to an exact thickness.



Slitter/Rewinder

Slitter/rewinders process an unlimited number of materials including paper, wires, and foils. Modularity and broad torque capability make Warner Electric the ideal system for the complete range of slitter/rewinder tensioning requirements.

Dual Output and Splicer System



Dual output tension control systems, often referred to as splicer controls, offer the user a multitude of options for the way they may be set up and used. Dual output tension controls have the capability of operating both outputs simultaneously from a single input or operating each output alternately, one being controlled by the sensing input and the other in a holding mode. This allows the controls to be used on either zero speed or flying splicers.

Control types include both analog, such as the TCS-310 dancer control and the TCS-320 remote/analog controller, and digital such as the MCS2000 ECA. Dual output controllers work like the single output controllers, except a few more features are included to provide switching between the output channels when operated as splicer controls.

The remote/analog splicer control provides an output proportional to the input. Typically, this is an open loop controller and does not compensate for acceleration, deceleration, or E-stops in the system. In addition, it provides no compensation for out of round roll conditions or variations associated with machine functions. This is the most basic type of controller and, in many cases, requires operator intervention to compensate for changing roll conditions.

The dancer splicer control, TCS-310, has additional features to provide automatic compensation for acceleration, deceleration, E-stop, out of round roll conditions and variations in the machine functions. A three-term control loop (P-I-D) is used to provide these functions. Set-up adjustments are provided to tune the system for optimum performance and, once set, requires no additional adjustment. With the dancer splicer system, operator involvement during a run is eliminated, and precise tension control is achieved.

The digital tension controller, MCS2000 ECA, allows the user a multitude of functions for both the type of inputs being used and the outputs for the controlled element. Because of its modularity, the user can tailor the MCS2000 system to specific application requirements. This system can be used as an open loop controller being controlled by a manual potentiometer, a roll follower pot, or some type of analog input sensor, i.e., ultrasonic or photoelectric.

The same controller can also be used with either a dancer or load cell and an optional input module for closed loop control. By changing the parameters, this is easily accomplished without having to change to a different control.

Depending on application requirements and the control selected, the optimum system for machine function and control can be selected.

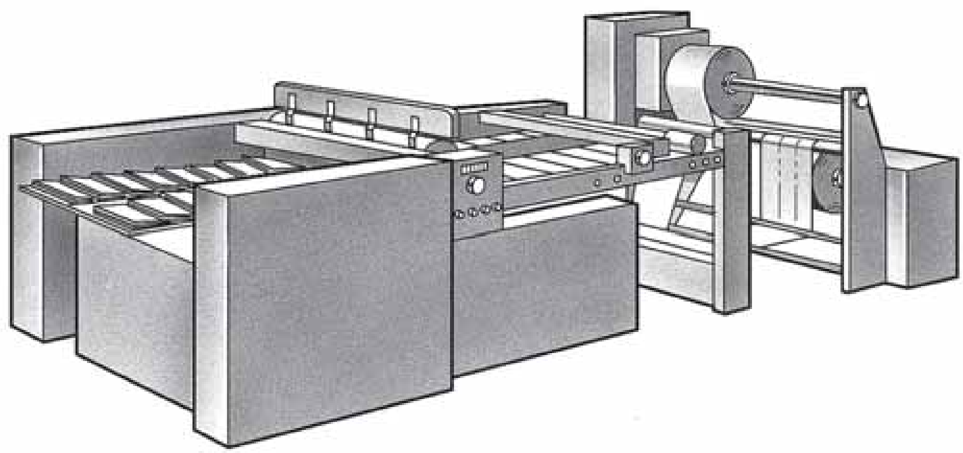
Typical Components for Splicer System

For Modular MTB Brakes Only

- Modular tension brake, MTB Series.
- Dual output tension controller, i.e., TCS-310 for dancer system, TCS-320 for remote/analog system, for providing current to brake magnets.
- Power supply, TCS-168, to provide control and brake power.
- Controlling element, i.e., pivot point sensor for dancer system; external pot, remote signal, or analog sensor for remote/analog controller.

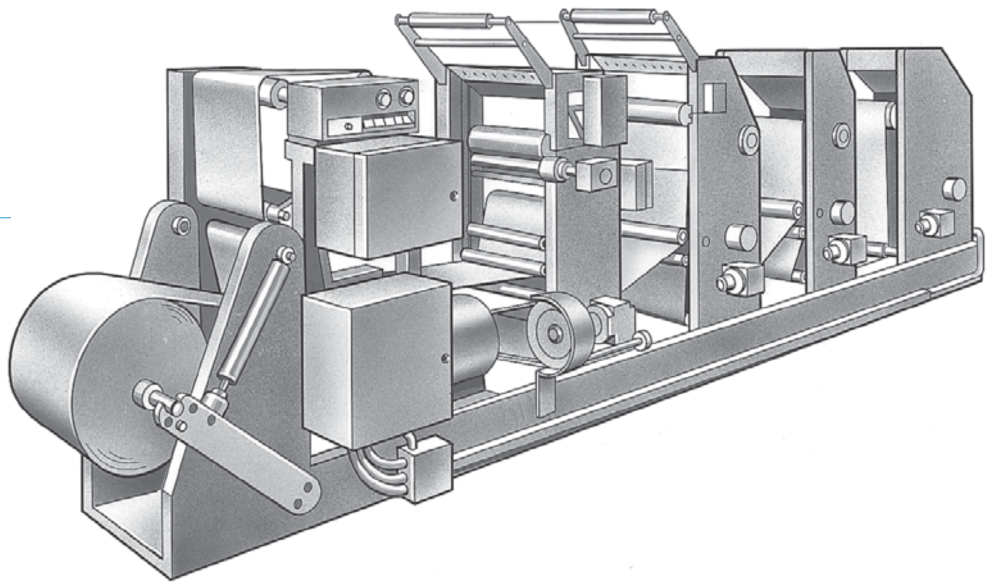
For other Brake/Clutch Systems

- Tension brake, clutch, or electronic motor drive, i.e., TB's, MTB's, ATT's, magnetic particles or pneumatic.
- Tension controllers, MCS2000 ECA and appropriate output modules and/or input modules as necessary depending on system type.
- Control element, i.e., dancer potentiometer, load cells, tachometers, or analog sensors, depending on application requirements.



Bag Making Machines

The smooth, consistent tension provided by Warner Electric tension control systems eliminates most reject bags caused by uneven reel tension. On preprinted bags, Warner Electric tension brakes and control systems allow superior registration control to keep the printed area in its optimum position.



Business Forms Press

Unique control circuitry allows Warner Electric tensioning systems to maintain exact web tension for intermittent web processing operations. From the beginning of each roll to its core, operator adjustment is unnecessary, even at the highest production speeds.

Unwind Tension Application Data Form

Warner Electric, Inc.
449 Gardner Street, South Beloit, Illinois 61080
Phone: 1-800-825-9050 • FAX: 815-389-6678 • E-mail: www.warnerelectric.com

Company Name:____________________ Date:__________

Address:______________________________

City:____________________ State:______ Zip:__________

Contact:____________________ Title:__________

Phone:____________________ Fax:____________________

E-mail:______________________________

Type of Equipment:______________________________

SYSTEM DATA:

Please check those that apply.

A. Application

- ☐ New
- ☐ Existing

If existing, what is currently being used?

____

____

B. Controlling Element

- ☐ Load Cell
- ☐ Dancer
  - ☐ Standard
  - ☐ Festoon
- ☐ Analog
  - ☐ Roll Follower
  - ☐ Sensor
- ☐ Other____________________

____

C. System Type Preference

- ☐ Brake
- ☐ Drive System
  - ☐ Center Wind
  - ☐ Surface
  - ☐ AC
  - ☐ DC
  - ☐ Other____________________

____

D. Web Motion

- ☐ Continuous
- ☐ Intermittent

If Intermittent;

Draw length:__________ in inches

Draw time:__________ seconds

Dwell time:__________ seconds

APPLICATION DATA:

A. Material:____________________

*Web Width:__________ inches

*Thickness:__________ inch, pts, mils
Circle appropriate measure

*Tension:
Pounds/Inch:__________ pounds
Total Tension:__________ pounds

B. Linear Speed:__________ ft./min.

C. Core Diameter:__________ inches

D. Max Diameter:__________ inches

E. Full Roll Weight:__________ pounds

F. Core Weight:__________ pounds

Machine Parmeters

G. Accel Time:__________ seconds

H. Decel Time:__________ seconds

I. E-Stop Time:__________ seconds

*** If additional application data is pertinent, please use second sheet.**

Intermediate Tension Application Data Form

Warner Electric, Inc.
449 Gardner Street, South Beloit, Illinois 61080
Phone: 1-800-825-9050 • FAX: 815-389-6678 • E-mail: www.warnerelectric.com

Company Name:______________________________ Date:______________

Address:______________________________

City:______________________________ State:__________ Zip:______________

Contact:______________________________ Title:______________

Phone:______________________________ Fax:______________________________

E-mail:______________________________

Type of Equipment:______________________________

SYSTEM DATA:

Please check those that apply.

A. Application

- ☐ New
- ☐ Existing

  If existing, what is currently being used?

  ____

  ____

B. Controlling Element

- ☐ Load Cell
- ☐ Dancer
  - ☐ Standard
  - ☐ Festoon
- ☐ Analog
  - ☐ Roll Follower
  - ☐ Sensor
- ☐ Other______________________________

  ____

C. System Type Preference

- ☐ Brake
- ☐ Clutch
- ☐ Drive System
  - ☐ Center Wind
  - ☐ Surface
  - ☐ AC
  - ☐ DC
  - ☐ Other______________________________

    ____

D. Web Motion

- ☐ Continuous
- ☐ Intermittent

  If Intermittent;

  Draw length: ____________ in inches

  Draw time:____________ seconds

  Dwell time:____________ seconds

APPLICATION DATA:

A. Material: ______________________________

*Web Width:____________ inches

*Thickness:____________ inch, pts, mils

Circle appropriate measure

*Tension:

Pounds/Inch:____________ pounds

Total Tension:____________ pounds

B. Linear Speed:____________ ft./min.

C. Core Diameter:____________ inches

D. Max Diameter:____________ inches

E. Full Roll Weight:____________ pounds

F. Core Weight:____________ pounds

Nip Roll Information

G. Nip Roll Matieral:______________________________

H. Nip Roll Diameter:____________ inches

I. Nip Roll Width:____________ inches

J. Nip Roll Thickness:____________ inches

K. Nip Roll Weight:____________ pounds

L. Number of Nip Rolls:______________________________

M. Nip Roll Contact Pressure:____________ pounds

Machine Parmeters

N. Accel Time:____________ seconds

H. Decel Time:____________ seconds

I. E-Stop Time:____________ seconds

*** If additional application data is pertinent, please use second sheet.**

Rewind Tension Application Data Form

Warner Electric, Inc.
449 Gardner Street, South Beloit, Illinois 61080
Phone: 1-800-825-9050 • FAX: 815-389-6678 • E-mail: www.warnerelectric.com

Company Name:______________________________ Date:______________

Address:____

City:______________________________ State:__________ Zip:______________

Contact:______________________________ Title:______________

Phone:______________________________ Fax:______________________________

E-mail:______________________________

Type of Equipment:____

SYSTEM DATA:

Please check those that apply.

A. Application

- ☐ New
- ☐ Existing

If existing, what is currently being used?

____

____

B. Controlling Element

- ☐ Load Cell
- ☐ Dancer
  - ☐ Standard
  - ☐ Festoon
- ☐ Analog
  - ☐ Roll Follower
  - ☐ Sensor
- ☐ Other______________________________

____

C. System Type Preference

- ☐ Brake
- ☐ Clutch
- ☐ Drive System
  - ☐ Center Wind
  - ☐ Surface
  - ☐ AC
  - ☐ DC
  - ☐ Other______________________________

____

D. Web Motion

- ☐ Continuous
- ☐ Intermittent

If Intermittent;

Draw length: ____________ in inches

Draw time:____________ seconds

Dwell time:____________seconds

APPLICATION DATA:

A. Material: ______________________________

*Web Width:____________ inches

*Thickness:____________ inch, pts, mils
Circle appropriate measure

*Tension:

Pounds/Inch:____________ pounds

Total Tension:____________ pounds

B. Linear Speed:____________ft./min.

C. Core Diameter:____________inches

D. Max Diameter:____________inches

E. Full Roll Weight:____________pounds

F. Core Weight:____________pounds

Machine Parmeters

G. Accel Time:____________seconds

H. Decel Time:____________seconds

I. E-Stop Time:____________seconds

Taper Tension Requirements

J. Taper Tension

- ☐ No
- ☐ Yes

If Yes, what percentage____________ %

K. Is holding required at stop?

- ☐ No
- ☐ Yes

*** If additional application data is pertinent, please use second sheet.**

Application Data Form Supplemental Information

Warner Electric, Inc.
449 Gardner Street, South Beloit, Illinois 61080
Phone: 1-800-825-9050 • FAX: 815-389-6678 • E-mail: www.warnerelectric.com

Company Name:______________________________ Date:______________

Address:____

City:____________________________ State:________ Zip:______________

Contact:____________________________ Title:______________

Phone:______________________ Fax:______________________

E-mail:____________________________

Type of Equipment:____

Additional Application Information

Application Data Form Supplemental Information

Warner Electric, Inc.
449 Gardner Street, South Beloit, Illinois 61080
Phone: 1-800-825-9050 • FAX: 815-389-6678 • E-mail: www.warnerelectric.com

Company Name:______________________________ Date:______________

Address:____

City:________________________ State:______ Zip:____________

Contact:__________________________ Title:________________

Phone:____________________ Fax:________________________

E-mail:________________________________

Type of Equipment:____________________________________

Additional Application Information

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.

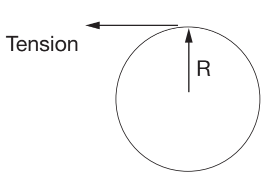
Caution should be taken, however, on machines that run more than one material at different line speeds. All combinations of tensions and line speeds should be checked to insure that brake sizing satisfies the most demanding condition (i.e. – the highest web HP).

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, WR²

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

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

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

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

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

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

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

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

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

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

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

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

Minimum Roll Torque = 4.5 lb. ft.

6. Maximum Roll Torque

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

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

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

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.

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

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

11. Roll E-Stop Torque, Controlled

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

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

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

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

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

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

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 Clutch Input Speed

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

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

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

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

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. 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**

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

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

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

$$\text{Energy Rate} = 403,200 \text{ ft. lbs./minute}$$

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**

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

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

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

Note: Constant values in formulas are in bold.

4. **Maximum Roll Speed**

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

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

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

5. **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**

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

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

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

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

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

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

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

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)

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

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

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

Roll Decel Torque = 26.53 + 63

Roll Decel Torque = 89.53 lb. ft.

10. Roll E-Stop Torque, 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.

11. 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

12. Motor HP based on Acceleration Torque

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

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

Motor HP = 19.89 HP

13. Motor HP based on Deceleration Torque

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

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

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.

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

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

E-Stop Horsepower = 37.27 HP

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:

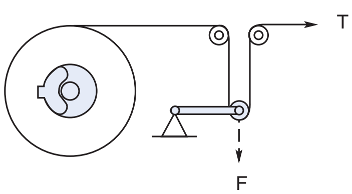
1. The controlling device, i.e. brake, clutch, or drive motor, AC or DC
2. The controller
3. A pivot point sensor which determines the position of the dancer roll
4. The dancer arm and roll assembly (customer supplied)

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

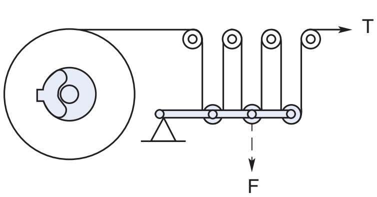
Where:

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.

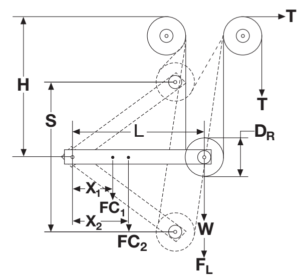


Figure 1 – Horizontal Dancer with Vertical Movement

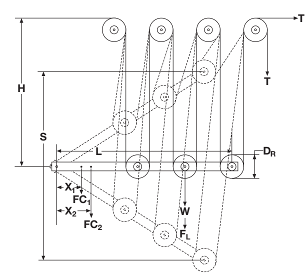


Figure 2 – Multiple Roll Dancer with Vertical Movement

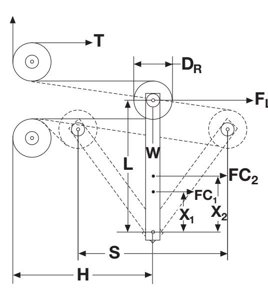


Figure 3 – Vertical Dancer with Horizontal Movement

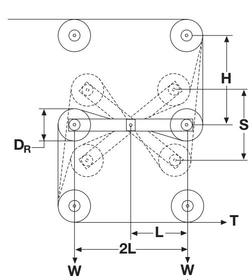


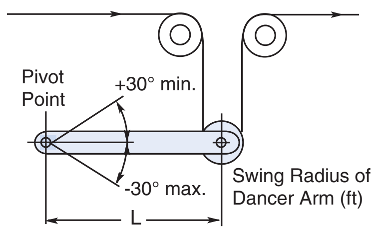
Figure 4 – S-Wrap Dancer with Vertical Movement

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

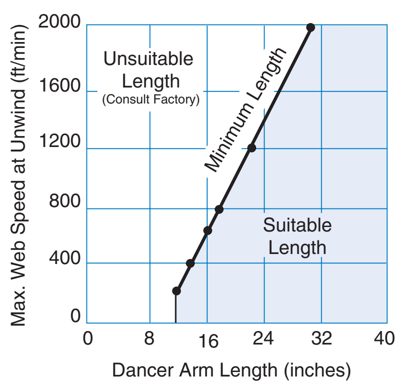


Chart 1 – Dancer Arm Length vs. Web Speed

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

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

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.

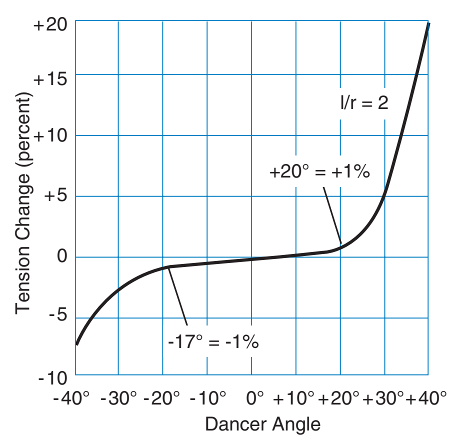


Chart 2 - Tension variation vs. dancer arm angle

* 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.

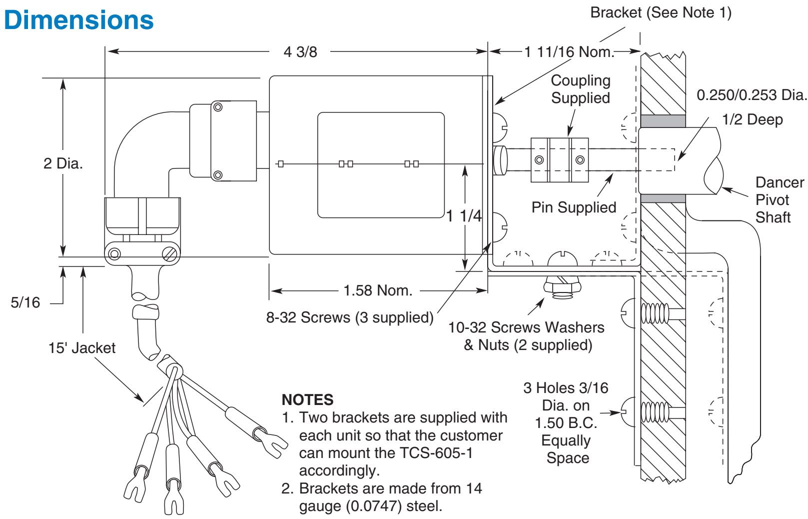
$$\text{Tension} = \frac{\text{Loading Force}}{2 \times \text{Number of Dancer Rollers}}$$

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.

TCS-605-1
TCS-605-2
TCS-605-5

Warner Electric pivot point sensor is a precision electronic positioning device which is used with the MCS-203, MCS-207, TCS-210 or TCS-310 dancer control system to provide smooth control of unwind stands operating at any speed. The sensor is mounted at one end of the dancer roll pivot shaft where it monitors the angular position, direction of travel and relative speed of dancer arm movement. TCS-605-2 used with drive systems.

Dimensions



Intermittent Motion Sensor Coupling

The Intermittent Motion Sensor Coupling is a two part coupling designed for applications where the web is started and stopped by intermittent motion. The design allows for an adjustable deadband so that the dancer arm can move before motion is translated to the pivot point sensor. This allows for smoother control of the tensioning device and prevents unwanted hunting and instability in the system. If your application requires this type of coupling, contact your Warner Electric tension specialist to determine if it is right for you.

Figure 1

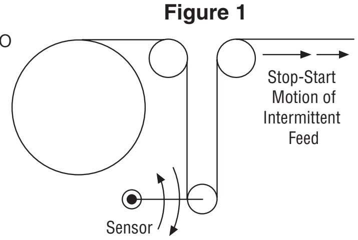
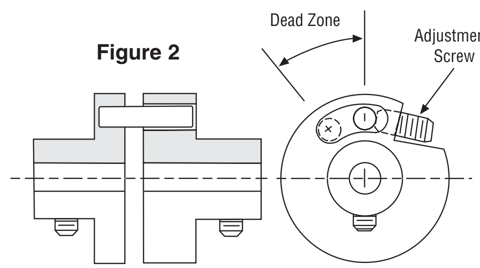


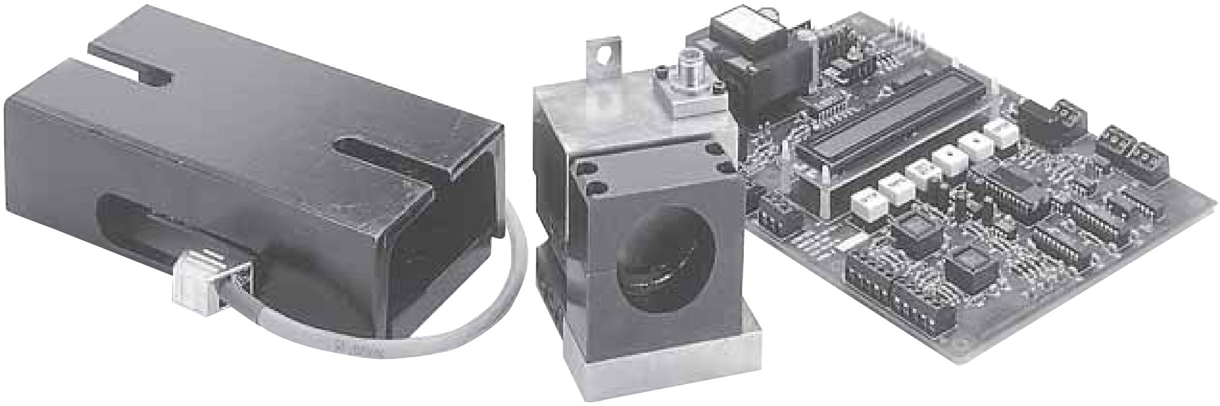
Figure 2



Specifications

Model No.	Part No.	Description
TCS-605-1	7330-448-002	Single turn potentiometer for dancer arm systems where the range of rotary motion from full-up to full-down dancer position is normally maintained within 60° (1KΩ)
TCS-605-2	7330-448-004	Single turn potentiometer for drive systems (5KΩ)
TCS-605-5	7330-448-003	Five turn potentiometer for festooned dancer systems (1KΩ)
Accessories		
	6910-101-001	Intermittent motion sensor coupling
	284-8000-003	Coupling for Pivot Point Sensors
	7330-101-001	TCS-605 Cable Assembly Only
	7330-101-002	TCS-605-1 Sensor Assembly Only
	7330-101-003	TCS-605-5 Sensor Assembly Only

Load Cell Sensors



Foot Mounted and End Shaft Mounted Series

FM Series Sensors

The foot mounted style load cells (used with pillow blocks) provide easy and convenient mounting to the roll that is being measured. It is a strain gauge style unit that is ideal for heavy tension applications.

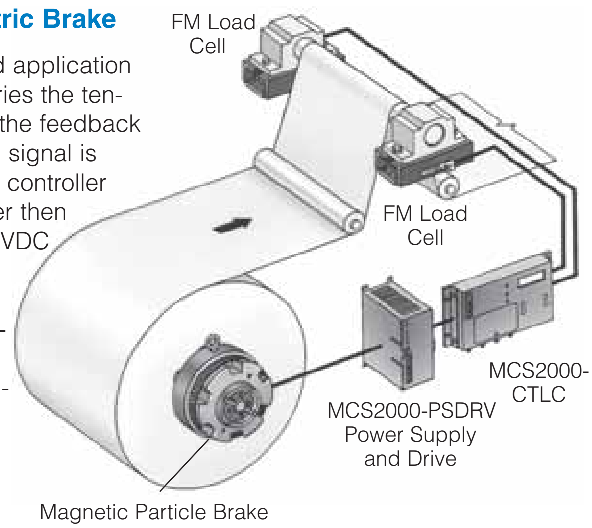
ES Series Sensors

The end shaft style load cells mount to the end of the roll that is being measured. It is a LVDT (Linear Variable Differential Transformer) style that can withstand overloads up to 10 times its rated load capacity. Several models are offered: dead shaft (no bearing), live shaft and cantilever where a single load cell can be used to measure the tension on the roll. Some units are powered with DC voltage and others are powered with AC. The AC units offer a price advantage over the DC.

Typical System Configuration Examples

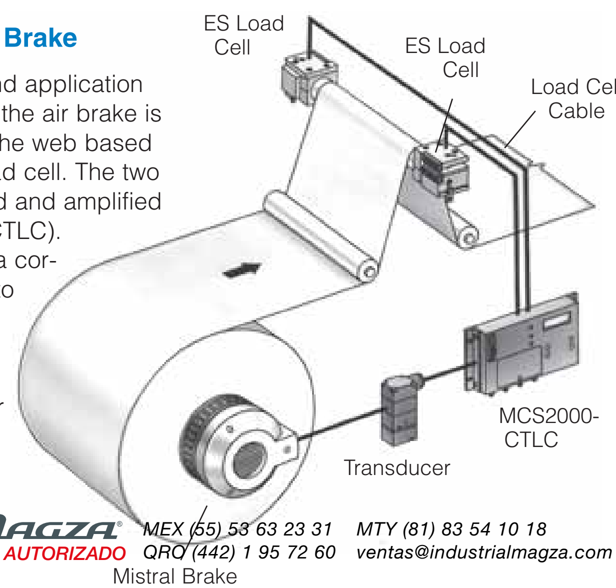
FM Load Cell with an Electric Brake

This is a single load cell unwind application example. The electric brake varies the tension on the web depending on the feedback from the load cell. The load cell signal is amplified and interpreted in the controller (MCS2000-CTLC). The controller then puts out a corresponding 0–10 VDC signal to the power supply and drive (MCS2000-PSDRV). The PSDRV then amplifies and interprets the signal from the controller and puts out a corresponding 0–24 VDC signal to the brake to apply either more or less braking.



ES Load Cell with a Pneumatically Operated Brake

This is a dual load cell unwind application example. In this application, the air brake is used to vary the tension on the web based on the feedback from the load cell. The two load cell signals are summed and amplified in the controller (MCS2000-CTLC). The controller then puts out a corresponding 0–20 mA signal to the transducer, which converts this signal from current to pressure to command the brake to apply either more or less braking.



Specifications

FM Series Foot Mounted Load Cells

Load Ratings	N	100	250	500	1,000	2,500	5,000	10K
	(lbs.)	(22)	(56)	(112)	(225)	(562)	(1,124)	(2,248)
Size		01	01	01	01	01	01	02

Input Power ±12 to ±15 VDC, ±5%

Output Signal 5 VDC factory setting at nominal load (can be rescaled for 25% load at +10 VDC output)

Deflection: 6mm at full load rating

Ambient Temperature 0–70°C (F)

Temperature Drift 0.1% of rating per °C

Non-Linearity & Repeatability <0.5%

Power Consumption 1 watt

Cable 16 ft. provided with load cell.

FM Series Model Numbers

FM 01 – 1000 – AC

FM = Model; 01 = Size; 1000 = Load in N; AC = Amplifier built in

ES Series End Shaft Mounted Load Cells

AC10 requires a power supply/amplifier

Load Ratings 60 lbs., 170 lbs., 500 lbs.

Input Power 15 Vrms @ 5 KHz

Deflection: 6mm at full load rating

Output Signal 3.2 volts AC/inch displacement/volt excitation

Output Impedance 780 ohms ±30%

Ambient Temperature –60° to +250°F (–50° to +620°C)

Temperature Drift 0.02%

Linearity & Repeatability 0.1% of full scale

Overload Protection 10 times maximum rated load of unit

Cable Two 30 ft. cables provided with load cells.

ES AC10 Series Load Ratings

A	60 lbs.
B	170 lbs.
C	500 lbs.

ES AC10 Series Model Numbers

AC 10 A 12 S _

AC 10 = Model; A = Load Rating; 12 = Shaft dia.*; S _ = Shaft Mounting Configurations

*See below for shaft diameters

Shaft Mounting Configurations

W1 = split bushing

W2 = solid bushing

S = system which includes one W1 load cell, one W2 load cell, two 30 ft. cables and a power supply (PSAC10)

PSAC10 Power Supply/Amplifier

Input Power 115/230 VAC, 50–60 Hz

Output Signal –10 to +10 VDC scaleable

Ambient Temperature 32°F to +160°F (0°C to +70°C)

Maximum cable distance between load cell and power supply board 100 feet

Part Number PSAC10 (For a 10 x 8 x 4 Housing add –H)

*ES, A30, B30 & C30 Series

Load Ratings		
	A30	8 lbs., 20 lbs., 50 lbs., 90 lbs.
	B30	8 lbs., 20 lbs., 50 lbs., 90 lbs., 140 lbs., 200 lbs., 300 lbs., 500 lbs.
	C30	8 lbs., 20 lbs., 50 lbs., 90 lbs., 140 lbs., 200 lbs., 300 lbs., 500 lbs.

Input Power 24 VDC at .040 amps (12 to 30 VDC acceptable, with LVDT output proportional)

Output Signal 3 VDC/unit

Ambient Temperature –60° to +250°F (–50° to +120°C)

Overload Protection 10 times rated load range

Deflection: 6mm at full load rating

Note: Tension cells are factory adjusted to provide an offset voltage with no load applied (no deflection). Using an input of 24 volts DC, the LVDT is set to provide an output of 3.5 volts into a resistive load of not less than 100,000 ohms. The voltage resulting from the maximum rated load then adds to or subtracts from the 3.5 volt offset. This results in an output of 6.5 volts in Compression.

ES A30, B30 & C30 Series Model Numbers

B A30 P 12 K W1

B = MS Connector; A30 = Model; P = Load Rating; 12 = Shaft Dia. Code; K = DC LVDT; W1 = Shaft Mounting Configurations

*Other sizes available if needed.

Shaft diameter

inches	¾	1	1¼	1 7/16
code	12	16	20	23

– Other diameters are available

Shaft Mounting Configurations

W1 = split bushing

W2 = solid bushing

ES A30 & C30 Series Load Ratings

M*	8 lbs.	**U**	90 lbs.	**Y**	300 lbs.
P	20 lbs.	**X**	200 lbs.	**Z**	500 lbs.
T	50 lbs.	**W**	140 lbs.		*shaft size 70 3/4 only

Load Cell Selection

The following steps should be followed to determine the proper load cell size and style for your application.

1. Determine whether you will be using one or two load cells.

It is best for two sensing heads to be used, one at each end of the sensing roll. The two individual web tension inputs are averaged in the controller, which takes care of non-central alignment of the web over the sensing roll and slack edges from a non-uniform reel. The AC10 and C30 can only be used in dual load cell applications. The FM Series and A30 can be used in single load cell applications. The A30 is designed to be used with a single pulley or sheave mounting with a projection of 1 or 2 inches. An ES style cantilever unit is also available in lengths to 18”. Consult the factory for more information.

2. Choose the load cell model that fits dimensionally.

The FM style is a foot mounted load cell (used with pillow blocks) that mounts perpendicular to the roll being measured. The ES style is an end shaft model where the mounting bolt centerline is on the axis of the measuring roll. There are two shaft mounting configurations with the ES style load cells. The “W1” cell clamps to the shaft while the “W2” cell allows for thermal expansion of the shaft. Both units have self aligning features. When using the dual load cell units (B30, C30 or AC10 series) one of each shaft mounting configuration must be used. It is recommended that a system be ordered in the AC10, B30 or C30 series (ex. AC10A12S) which will insure one “W1” load cell and one “W2” load cell is supplied as a matched pair.

The AC10 is an AC version load cell that is economically priced when compared with the other ES models, even with the added power supply board that is required to power it.

Available sizes and dimensions are listed on pages 42 & 43 for the ES or FM style units. Choose the unit(s) that will best fit the machine construction.

3. Load Cell Force Calculations

The FM style load cell can be mounted regardless of orientation, but has to work in compression. Only the perpendicular force (resultant) is measured by the load cell. The perpendicular force can be at a maximum permitted angle of ±30°. The FM style is a strain gauge load cell and the maximum tension in the web used (T) should be the potential overload force.

The ES style load cells can be mounted at any angle around the axis of the measuring roll with any wrap angle. They work equally well in either tension or compression making it easy to adapt them to any new, retrofit, or replacement application. The mechanical structure and primary conversion element is designed to handle overloads at ten times the rated load range. Therefore, these units don't need to be oversized to provide adequate overload protection.

The following selection information is required to select a load cell:

T = maximum tension in the web (lbs.)

W = weight of the sensing roll (lbs.) acts vertically

X = wrap angle (degrees), 180° max.

Y = angle between resultant force of tension and vertical (degrees)

SF = Safety factor. Use 1 for ES style load cells and 2 for FM style load cells.

RF = Resulting force (lbs.)

4. Choose the load cell rating that is equal to or greater than the force calculation.

– Minimum rating of each cell should exceed 7% of maximum rating.

5. Choose accessories

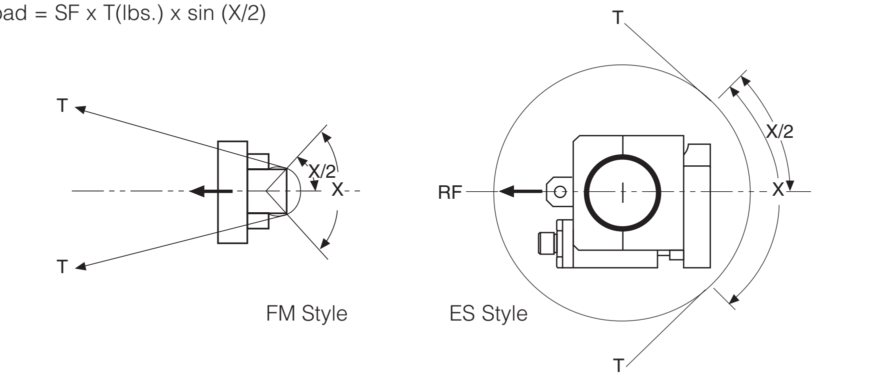
a. For ES style load cells choose shaft diameter. Chart is on page 43.

b. For the A30, B30 or C30 models choose cables L1A25 or L1A99 which are 25 or 99 ft. cables. Other lengths are available. A cable is needed for each load cell ordered.

c. For the AC10 model the PSAC10 (power supply amplifier) is needed. Specify without or PSAC10-H with housing.

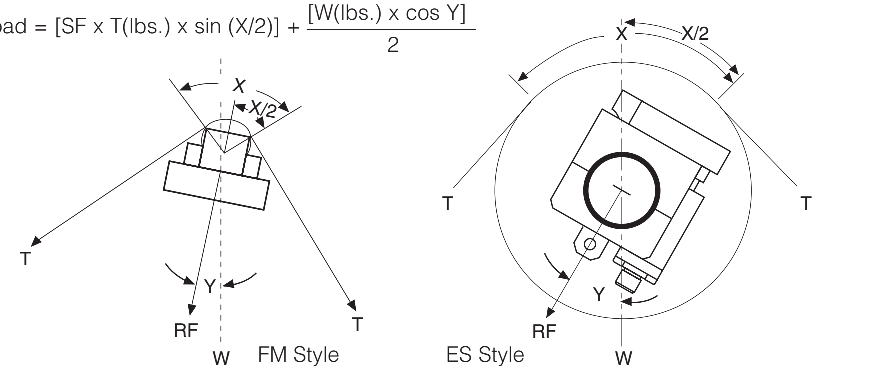
Case 1: Resultant force points horizontal

Load = SF x T(lbs.) x sin (X/2)



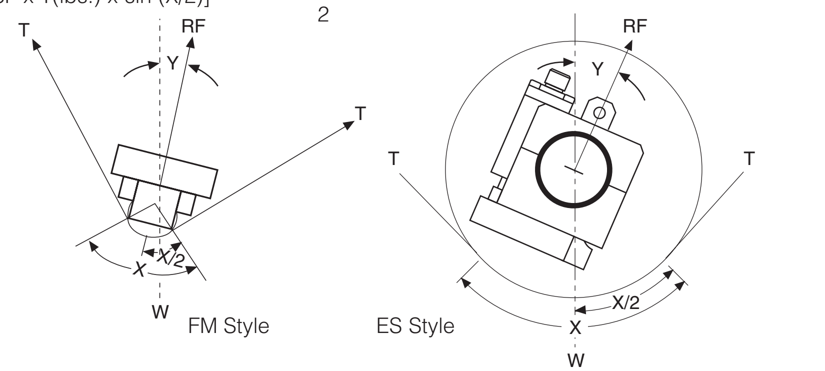
Case 2: Resultant force points down

$$\text{Load} = [\text{SF} \times \text{T(lbs.)} \times \sin (X/2)] + \frac{[\text{W(lbs.)} \times \cos Y]}{2}$$



Case 3: Resultant force points upward

$$\text{Load} = [\text{SF} \times \text{T(lbs.)} \times \sin (X/2)] - \frac{[\text{W(lbs.)} \times \cos Y]}{2}$$

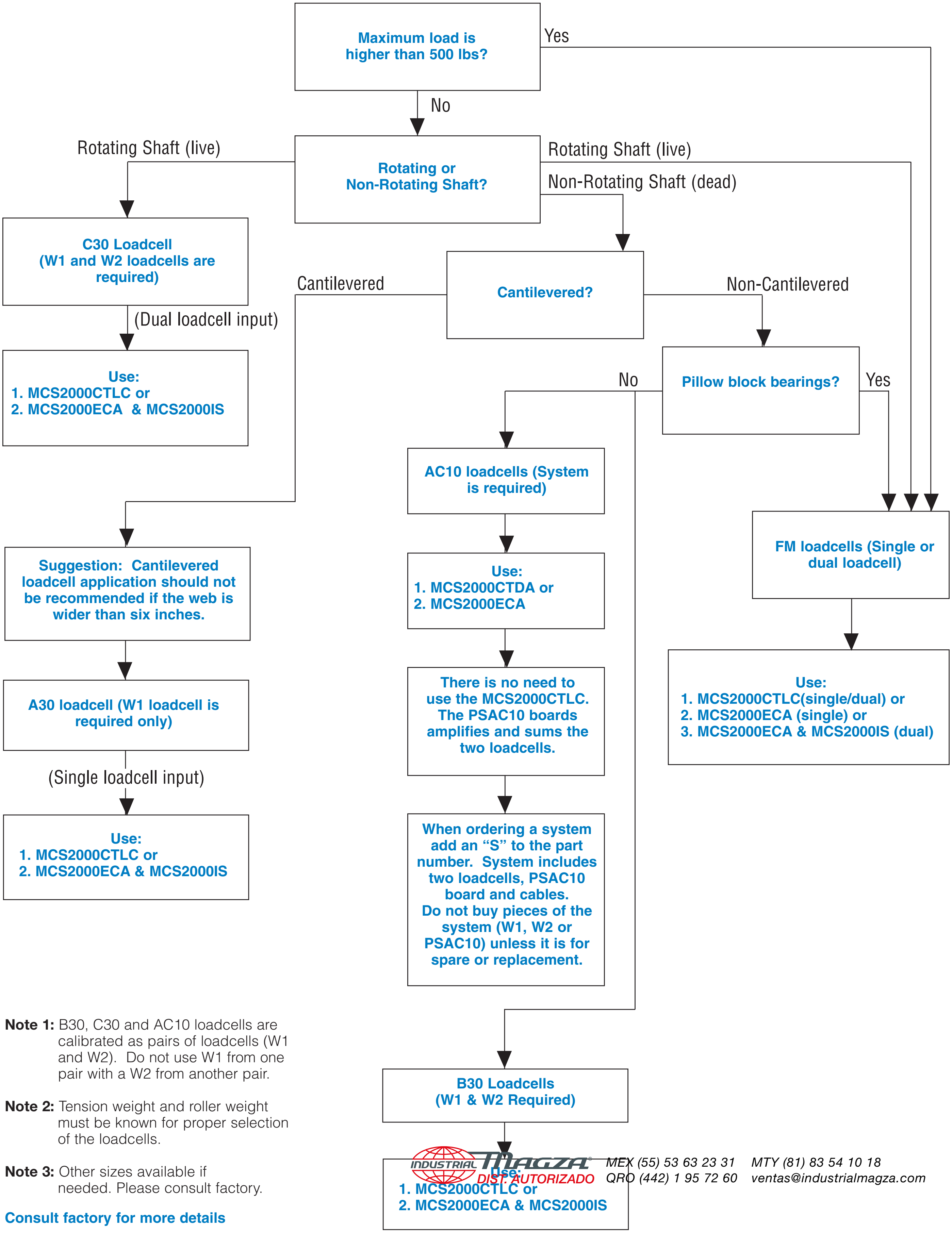


Sin/Cos Table

Degrees	Sin	Cos
0°	.0000	1.000
5°	.0872	.9962
10°	.1736	.9848
15°	.2588	.9659
20°	.3420	.9397
25°	.4226	.9063
30°	.5000	.8660

Degrees	Sin	Cos
35°	.5736	.8192
40°	.6428	.7660
45°	.7071	.7071
50°	.7660	.6428
55°	.8192	.5736
60°	.8660	.5000
65°	.9063	.4226

Degrees	Sin	Cos
70°	.9397	.3420
75°	.9659	.2588
80°	.9848	.1736
85°	.9962	.0872
90°	1.000	.0000



Note 1: B30, C30 and AC10 loadcells are calibrated as pairs of loadcells (W1 and W2). Do not use W1 from one pair with a W2 from another pair.

Note 2: Tension weight and roller weight must be known for proper selection of the loadcells.

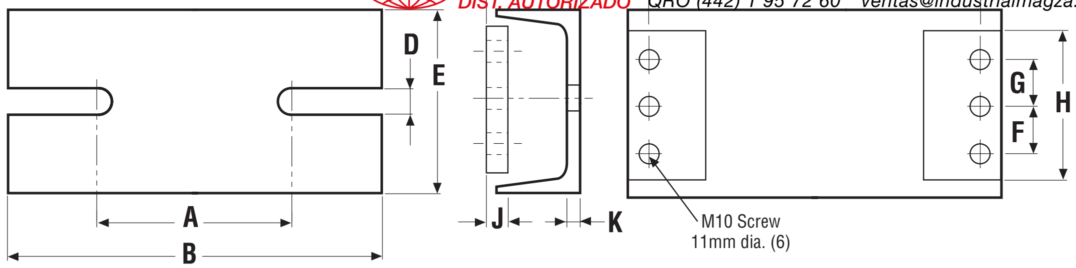
Note 3: Other sizes available if needed. Please consult factory.

Consult factory for more details

Dimensions

FM Series

Foot mounted load cells



inches/(mm)

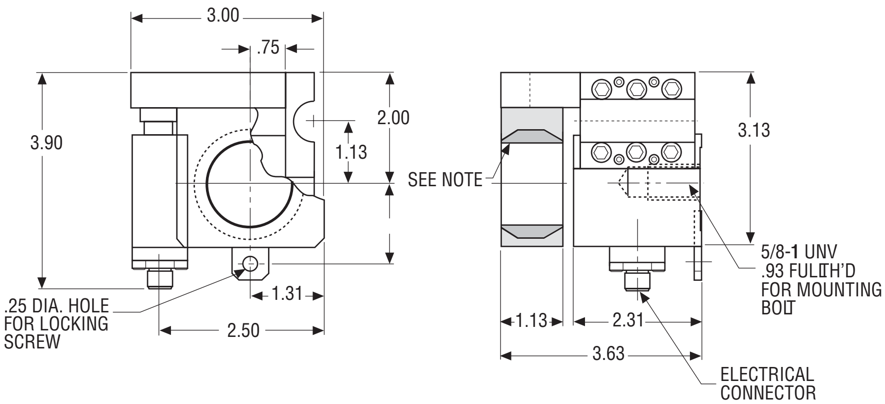
Size	Part Number	Load Ratings (lbs.)	A	B	C	D	E	F	G	H	I	J	K
	6910-840-100	**22**											
	6910-840-102	**56**											
1	6910-840-104	**112**	4.055	7.874	6.890	.512	4.016	.984	.984	3.150	2.047	.472	.236
	6910-840-106	**225**	(103)	(200)	(175)	(13)	(102)	(25)	(25)	(80)	(52)	(12)	(6)
	6910-840-108	**562**											
	6910-840-110	**1124**											
2	6910-840-112	**2248**	5.591 (142)	8.858 (225)	7.677 (195)	.669 (17)	5.00 (127)	.984 (25)	.984 (25)	3.937 (100)	2.165 (55)	.709 (18)	.236 (6)
	6910-101-089	Cable Assembly 16 ft.											

ES Series

End Shaft Mounted Load Cells

AC10

Dual Load Cell, Non-Rotating Shaft Load ratings 60 lbs., 170 lbs., 500 lbs.

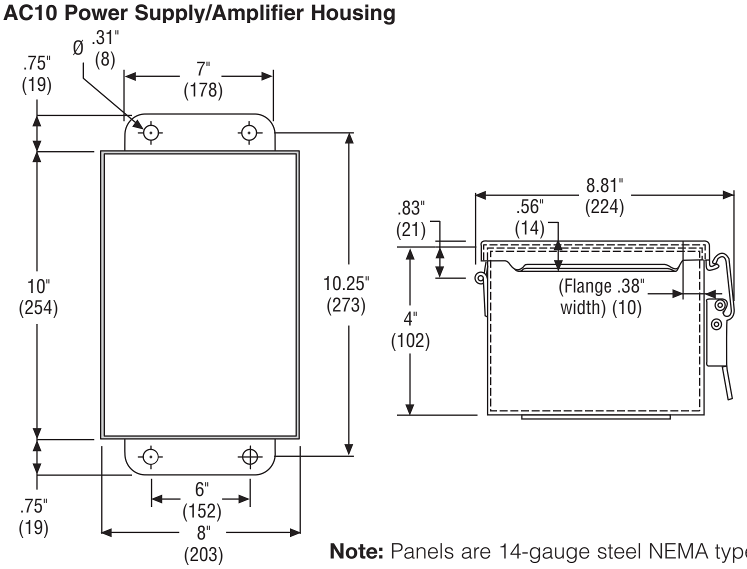


Cable Assembly
L1A30 30 ft. Cables

Note:
Stainless steel self-aligning bushing provided for shaft sizes 3/4", 1", 1-1/4" and 1-7/16" diameters. Other shaft diameters available on special order.

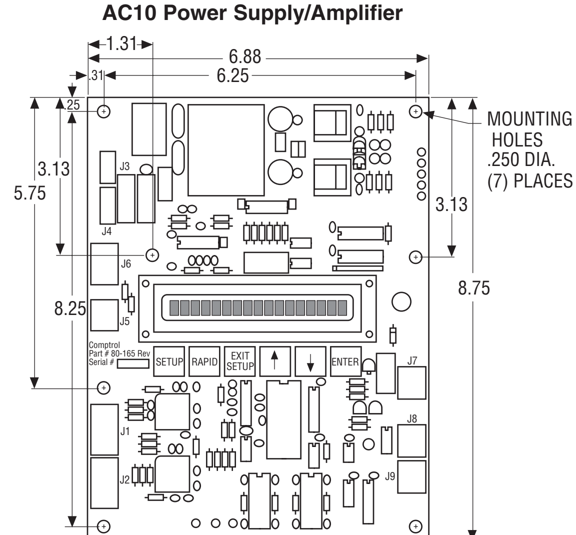
PSAC10-H

AC10 Power Supply/Amplifier Housing



PSAC10

AC10 Power Supply/Amplifier

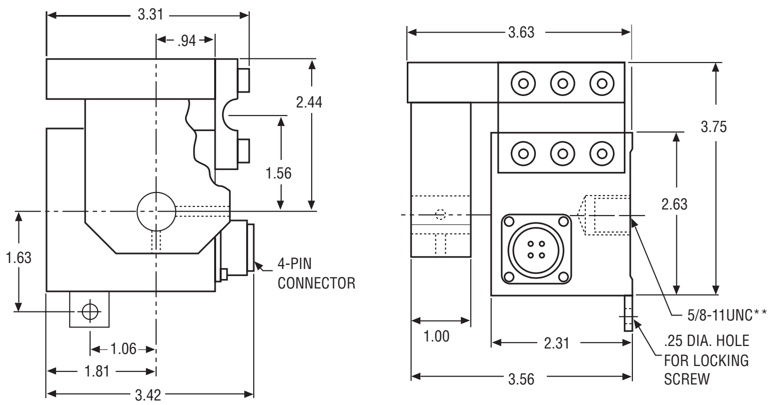


Note: Panels are 14-gauge steel NEMA type 12 and 13.

A30

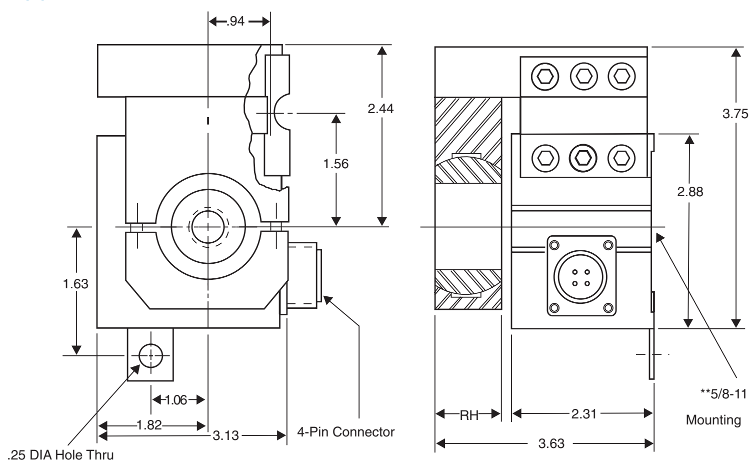
Single Load Cell, Non-Rotating Shaft

Sheave or pulley mounting with projection of 1 or 2 inches.



Load Ratings: 20 lbs., 50 lbs., 90 lbs.
Note: Other load ratings available - consult factory.

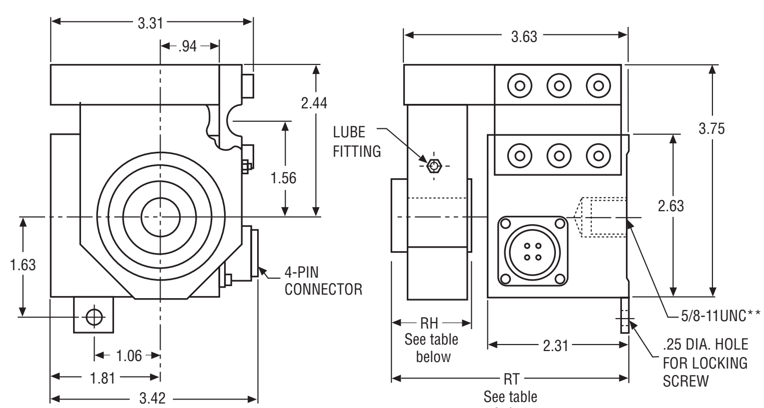
B30



Load Ratings: 20 lbs., 50 lbs., 90 lbs., 200 lbs., 500 lbs.
Note: Other load ratings available - consult factory.

C30

Dual Load Cell, Rotating Shaft



Load Ratings: 20 lbs., 50 lbs., 90 lbs., 200 lbs., 500 lbs.
Note: Other load ratings available - consult factory.

Cable Assemblies– For All 30 Series

L1A25	25 ft. with Connector
L1A99	99 ft. with Connector

RH and RT dimensions based on shaft diameter

Inches	3/4	1.0	1-1/4	1-7/16
Code	12	16	20	23
RH	1.31	1.38	1.69	
RT	3.88		4.13	

Standard Shaft Diameters

Shaft Diameter	Standard
0.75"	3/4"
1.00"	1"
1.25"	1-1/4"
1.4375"	1-7/16"

Other shaft sizes available on special order - consult factory