Section 16 General Engineering Information

The following section is provided to furnish additional information that will increase the confidence level and understanding of the designer and engineer in applying the formulas published in previous sections. This section concentrates on working definitions and illustrations of the concepts governing the application of clutches and brakes.

There are three steps to successfully complete the selection of the proper size of clutch and brake.

1. **Calculate the Required Dynamic Torque Capacity for both the clutch and the brake.**
2. **Determine the Thermal Horsepower per engagement for both the clutch and the brake.**
3. **Determine the Required Average Thermal Horsepower Capacity for both the clutch and the brake.**

Dynamic Torque

Objective:
Determine the Dynamic Torque Capacity required at the Clutch/Brake Output Shaft.

To accurately determine the Torque Requirements during an acceleration or deceleration operation, the total inertia, component efficiency and total load torque must be determined and reflected back to the Clutch/Brake output shaft. A major consideration is the proper application of the inefficiencies to the individual drive components during an acceleration or deceleration period. The total inertial torque is the sum of all the individual torques associated with each drive component. The load torque is considered next and again the proper efficiency factor must be applied. The dynamic torque is then found by adding the total inertial torque and the load torque together.

Three distinct terms make up the equation for the dynamic torque.

1. Inertial Torque .... $\dfrac{WK^2 \times N}{307.2 \times t}$

2. Drive Efficiency .... E

3. Load Torque .... T_L

Terms #2 and #3 are always present in the system while the Inertial term #1 is only applicable during acceleration or deceleration.

Inertial Torque

Objective:
Determine the Reflected Inertia of the Drive System with Respect to the Output Shaft of the Clutch/Brake.

The first step in determining the dynamic torque is to determine the inertia in the system to be accelerated or decelerated. Inertia is the measure of resistance an object possesses to a change in its state of motion. The term used to quantify this property is $\mathbf{WK^2}$. The units for this term are **Lb. Ft.**^{2}. There are two primary activities required to obtain the $\mathbf{WK^2}$ in a form that can be used.

1. **Determine the inertia of each component in the system that will be cycled.**
2. **Reflect the inertia of each cycled component back to the clutch and brake**

1. Determining the Inertia of an Object

The rotational inertia an object has is a function of its mass and how that mass is distributed about the rotating axis. The effective radius is where the entire mass of the object can be thought to be concentrated. This effective radius is called the *Radius of Gyration* and is designated by the symbol **K**. The **WK²** is found by squaring **K** and multiplying it by the entire weight of the object.

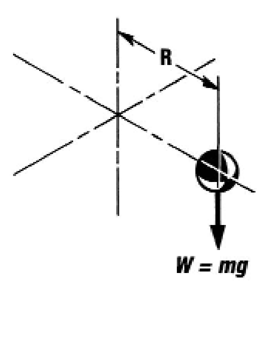
The steps to find the **WK²** of any object are:

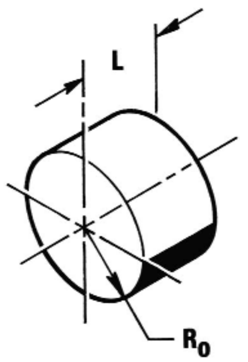
1. Determine the weight of the object.

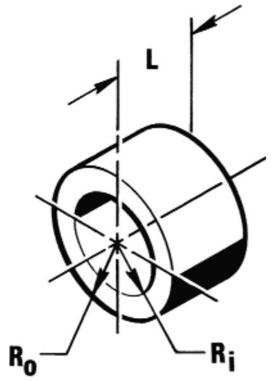
2. Determine K² from the geometry of the object.

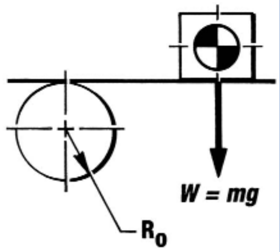
3. Multiply the two terms together.

The following formulas can be used to calculate the **Weight** and **K²** of various objects.









	Concentrated Weight	Uniform Disc	Hollow Cylinder	Translating Weight
Weight	$W = W$	$W = \frac{\pi D^2}{4} \times L \times Lb/In^3$	$W = \frac{\pi (D_o^2 - D_1^2)}{4} \times L \times Lb/In^3$	$W = W$
K^2	$K^2 = R^2$	$K^2 = \frac{R_0^2}{2}$	$K^2 = \frac{R_0^2 + R_1^2}{2}$	$K^2 = R_0^2$

2. Reflecting the Inertia of an Object back to the Clutch and Brake.

The objective is to obtain an equivalent **WK²** that can be attached to the output of the Clutch/Brake and represents the inertia of all cycled components in the system. The procedure to obtain the equivalent or reflected inertia is based on the principle that the total energy in the system is conserved. *(This means that the reflected inertia of an object is equal to the actual kinetic energy it possesses in the drive system.)* Since the kinetic energy varies with the square of the speed the reflected inertia is the object's actual inertia affected by the square of the ratio of the operating speed to the speed of the Clutch/ Brake. The total reflected inertia is the sum of the reflected inertia of each individual component. The following example, on the next page, will be used to help clarify selection of the correct Clutch/Brake.

Calculating the Inertia and Reflecting it to the Clutch/Brake

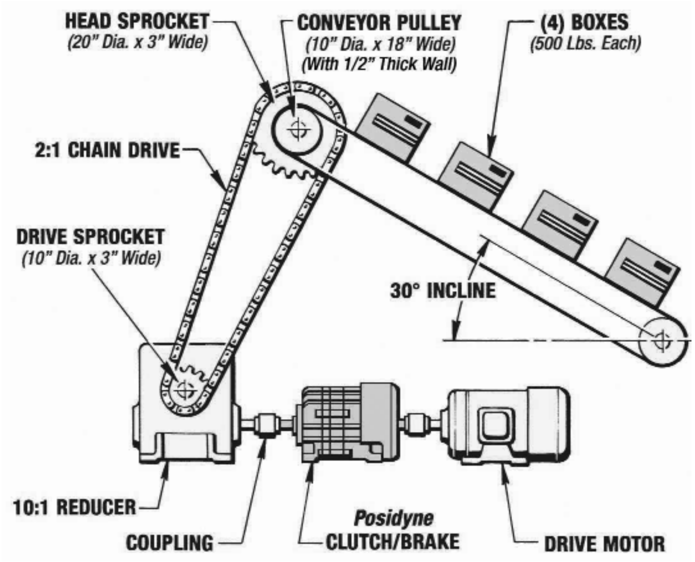
Example:

Required:

- Acceleration Time .... .4 Sec.
- Deceleration Time .... .13 Sec.
- Cycles per Minute 10
- Logic Type .... A
- Clutch/Brake Size 03

Posidyne

- Conveyor Efficiency 8
- Chain Drive Efficiency 9
- Reducer Efficiency 8
- Max. Pressure .... 60 PSIG



Boxes

Weight = 500 Lbs. ← Weight of each box

WK^2 = Weight x R_O^2 x No. of Boxes ← Total inertia of the 4 boxes

$$= 500 \times \left(\frac{5"}{12}\right)^2 \times 4 = 347.20 \text{ Lb. Ft.}^2$$

R_O = Radius of Conveyor Pulley

$$WK^2 \;@\; \textit{Posidyne} = WK^2 \times \left(\frac{1}{\text{Total Ratio}}\right)^2$$

← Inertia reflected thru the drive ratio from the box to the Clutch/Brake

$$= 347.20 \times \left(\frac{1}{10 \times 2}\right)^2 = \mathbf{.87\ Lb.\ Ft.^2}$$

Conveyor Pulleys

$$\text{Weight} = \pi \left(\frac{D_o^2}{4} - \frac{D_i^2}{4}\right) \times L \times .283\ (\textit{Lb.In.}^3)$$

← Total volume x .238 Lb.In.³ (Specific weight of steel)

$$= 3.1416 \left(\frac{10^2}{4} - \frac{9^2}{4}\right) \times 18 \times .283 = 76.02 \text{ Lbs.}$$

D_O = OD of Conveyor Pulley *(In.)*
D_i = ID of Conveyor Pulley *(In.)*
L = Length of Conveyor Pulley *(In.)*

$$WK^2 = \text{Weight} \times \left(\frac{R_o^2 + R_i^2}{12}\right) \times \text{No.}$$

$$= 76.02 \times {}^1\!/_2 \times \left[\left(\frac{5"}{12}\right)^2 + \left(\frac{4.5"}{12}\right)^2\right] \times 2 = 23.89 \text{ Lb.Ft.}^2$$

← Total inertia of both pulleys

No. = Number of Pulleys
R_O = Outside Radius *(Feet)*
R_i = Inside Radius *(Feet)*

$$WK^2 \;@\; \text{Clutch/Brake} = WK^2 \left(\frac{1}{\text{Total Ratio}}\right)^2$$

← Inertia reflected to the Clutch/Brake

$$= 23.89 \left(\frac{1}{10 \times 2}\right)^2 = \mathbf{.06\ Lb.Ft.^2}$$

20" Diameter Sprocket, 3" Wide

$$\text{Weight} = \pi \frac{Do^2}{4} \times L \times .283\ (\text{Lb.In.}^3)$$

$$= 3.1416 \times \frac{20^2}{4} \times 3 \times .283 = 266.72 \text{ Lbs.}$$

Weight calculated from total volume times .283 Lb. In.^{3} for steel.

D_O = OD of Sprocket *(In.)*

$$WK^2 = \frac{WR^2}{2} = \frac{W \times (R_O)^2}{2}$$

WK^2 calculated using $K^2 = \frac{R^2}{2}$

$$= \frac{266.72 \times (10''/12)^2}{2} = 92.61 \text{ Lb. Ft.}^2$$

R_O = Radius of Sprocket *(Feet)*

$$WK^2 \text{ @ Clutch/Brake} = WK^2 \left(\frac{1}{\text{Total Ratio}}\right)^2$$

$$= 92.61 \times \left(\frac{1}{10 \times 2}\right)^2 = \mathbf{.23\ Lb.Ft.^2}$$

WK^2 reflected to the Clutch/ Brake through the chain drive and reducer.

10" Diameter Sprocket, 3" Wide

$$\text{Weight} = \pi \frac{D_O{}^2}{4} \times L \times .283\ (\text{Lb.In.}^3)$$

$$= 3.1416\ \frac{10^2}{4} \times 3 \times .283 = 66.68 \text{ Lbs.}$$

Weight calculated from total volume times .283 Lb. In.^{3} for steel.

D_O = OD of Sprocket *(In.)*

$$WK^2 = \frac{WR^2}{2} = \frac{W \times (R_O)^2}{2}$$

WK^2 calculated using $K^2 = \frac{R^2}{2}$

R_O = Radius of Sprocket *(Feet)*

$$= \frac{66.68 \times (5''/12)^2}{2} = 5.79 \text{ Lb. Ft.}^2$$

$$WK^2 \text{ @ Clutch/Brake} = WK^2 \left(\frac{1}{\text{Total Ratio}}\right)^2$$

$$= 5.78 \times \left(\frac{1}{10}\right)^2 = \mathbf{.06\ Lb.Ft.^2}$$

WK^2 reflected to the Clutch/ Brake through the chain drive and reducer.

10:1 Reducer	Coupling	Posidyne Clutch/Brake
WK^2 = **.17 Lb. Ft.^{2}** (Information from Vendor)	WK^2 = **.78 Lb. Ft.^{2}** (Information from Vendor)	03 WK^2 = **.20 Lb. Ft.^{2}** (Information from Vendor)

Total System Reflected Inertial Torque

$$WK^2 = .87 + .06 + .23 + .06 + .17 + .78 + .20 = \mathbf{2.37\ Lb.\ Ft.^2}$$

16 Procedure to Determine the Required Dynamic Torque Capacity for the Clutch and the Brake.

To make a seemingly complex procedure easily understandable use the **Dynamic Torque Analysis Table** shown below. Using the previous example will help you follow the steps to proper Clutch and Brake Selection by filling in the Table on the next page.

Dynamic Torque Analysis Table

1	2	3	4	5	6	7	8	CLUTCH 9	CLUTCH 10	BRAKE 11	BRAKE 12
Component	Speed Ratio @ Input	Inertia WK^2 (Lb. Ft.^{2})	Component Efficiency	Accumulated Efficiency Factor %	Reflected Inertia WK^2 (Lb. Ft.^{2})	Load Torque (Lb. In.)	Reflected Load Torque T_L (Lb. In.)	Reflected Inertial Torque T_{ic} (Lb. In.)	Dynamic Torque T_{dc} (Lb. In.)	Reflected Inertial Torque T_{ib} (Lb. In.)	Dynamic Torque T_{db} (Lb. In.)
Posidyne	1	0.20	1.0	1.000	0.20			35.16		-108.17	
Coupling	1	0.78	1.0	1.000	0.78			137.11		-421.88	
10:1 Reducer	1	0.17	0.8	1.000	0.17			29.88		-91.95	
10” Dia. Sprocket	10	5.79	0.9	0.800	0.06			13.18		-25.96	
20” Dia. Sprocket	20	92.61	1.0	0.720	0.23			56.15		-89.57	
Conveyor Pulley	20	23.89	0.8	0.720	0.06			14.65		-23.37	
Boxes	20	347.20	1.0	0.576	0.87	6732.0	584.38	265.50		-271.04	
Summation					2.37	6732.0	584.38	551.63	1136.01	-1031.94	-447.56

Dynamic Torque (Clutch)

1. **List all of the Cycled Components** in Column 1 starting at the Clutch/Brake and proceeding to the Load.
2. **List the Input Gear Ratio for each Component** in Column 2. Notice that the 10:1 Reducer is assigned a ratio of 1 because the input shaft is connected directly to the clutch/brake and runs at 1800 RPM. The 10” Sprocket is assigned a ratio of 10 because it turns at 180 RPM. The 20” Sprocket is assigned a ratio of 20 because it turns at 90 RPM.,etc.
3. **List the Rotational Inertia for each component** in Column 3. *These values were calculated on pages 16.3 and 16.4.*
4. **List the Component Efficiency** in Column 4. *These values can be obtained from vender information or by using engineering judgement.*
5. **Determine the Efficiency Factor for each Component** and list it in Column 5. *The Efficiency Factor at the Posidyne Clutch/Brake is 1. The remaining efficiency factors are determined by multiplying all the Efficiency Values together that are listed in Column 4 above the component considered in the table.*

Posidyne: =1
Coupling: (1) = 1
Reducer: (1) x (1) = 1
10” Dia. Sprocket: (1) x (1) x (.8) = .8
20” Dia. Sprocket: (1) x (1) x (.8) x (.9) = .72
Conveyor Pulley: (1) x (1) x (.8) x (.9) x (1) = .72
Boxes: (1) x (1) x (.8) x (.9) x (1) x (.8) = .576

6. **Compute the Reflected Inertial Torque Requirements for each Component** and list it in Column 6 using the gear ratio and rotational inertia listed in Columns 2 and 3.
7. **Determine the Load Torque Requirement for the Drive** and list it in Column 7 for the component with which it is associated. *Load Torque is the torque required to maintain a system at constant velocity.* **This Torque can be found by solving for Static Equilibrium.** T_L is computed for the boxes on the next page. Enter 6732.0 in column 7 for the boxes.
8. **Apply the associated Efficiency Factor to T_L and reflect it back to the Clutch/Brake.** The Torque at the Clutch/Brake varies inversely to the speed reduction between the Clutch/Brake and the Conveyor Head Pulley. **Determine the Reflected Load Torque** and list it in column 8.

$$\text{Reflected Load Torque} = T_L = \frac{T_L \text{ (column 7)}}{\text{Eff. factor (column 5) x ratio}}$$

$$T_L = \frac{6732}{.576 \times 20} = 584.4 \text{ Lb. In.}$$

9. **Determine the Reflected Inertial Torque Requirement for each component** and list it in column 9. *The sum of this column is the Total Reflected Inertial Torque Requirement. The two conversion constants in the following equations are (12 Inches / foot) in the numerator and (307.2 foot revolution/minute second) in the denominator.*

$$\text{Reflected Inertial Torque} \quad T_i = \frac{WK^2 \textit{ (col. 6)} \times N \textit{ (change in speed)} \times 12}{307.2 \times t \textit{ (time)} \times Ef \textit{ (Efficiency factor col. 5)}}$$

$$\text{Boxes:} \quad T_i = \frac{(.87) \times (1800 - 0) \times 12}{307.2 \times (.4) \times .576} = 265.5 \text{ Lb. In.}$$

$$\text{Conveyor Pulley} \quad T_i = \frac{(.06) \times (1800 - 0) \times 12}{307.2 \times (.4) \times .72} = 14.65 \text{ Lb. In.}$$

Etc.....

10. The sum of the Total Reflected Inertial Torque and the Reflected Load Torque is the **Dynamic Clutch Torque Required.**

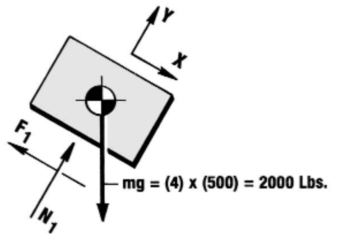
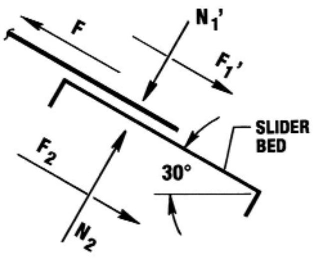
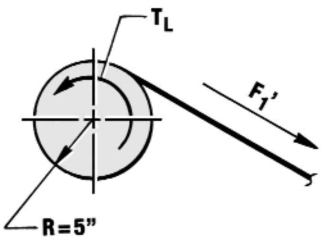
Required Dynamic Clutch Torque $= T_{dc} = T_L + T_{ic}$

$$T_{dc} = 584.38 + 551.63$$

$$T_{dc} = 1136.0 \text{ Lb. In.}$$

16

Solving for Static Equilibrium

Example - FBD #1 - Box	Example - FBD #2 - Slider Bed & Belt	Example - FBD #3 - Head Pulley
		

FBD #1 - Box

$\Sigma F_x = 0$

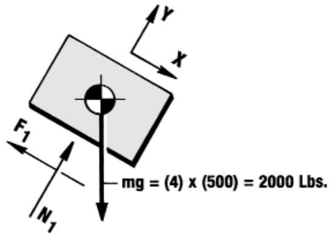
$0 = W \ x \ Sin\ 30° - F_1$

EQ. 11.1 → $F_1 = W \ x \ Sin\ 30°$ (Assuming no slippage between belt and boxes)

$\Sigma F_y = 0$

$0 = N_1 - W \ x \ Cos\ 30°$

EQ. 11.2 → $N_1 = W \ x \ Cos\ 30°$



FBD #2 - Slider Bed & Belt

EQ. 11.3 → $\Sigma F_x = 0$

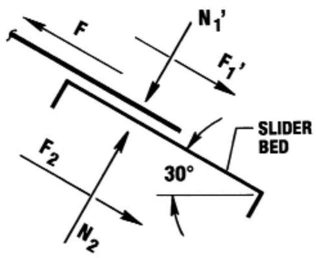
$F_1' + F_2 = F$

$\Sigma F_y = 0$

EQ. 11.4 → $N_1' = N_2$

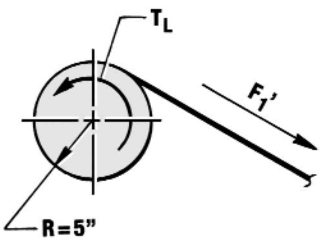
EQ. 11.5 → $F_2 = N_2\mu = N_1'\mu$

$\mu = .2$ (Between slider bed and belt)



FBD #3 - Head Pulley

EQ. 11.6 → $\Sigma M_o = T_L = F_1' \ x \ (5")$



Substitute EQ. 11.1 into 11.3 - Substitute EQ. 11.2 into 11.5 - Substitute EQ. 11.5 into 11.3

EQ. 11.7 → $F = W \ x \ Sin\ 30° + W \ x \ Cos\ 30° \ \mu$

$F_1 = 2000 \ x \ (.5) + 2000 \ x \ (.866) \ x \ (.2) = 1346.4$ Lbs.

Substitute 11.7 into 11.6 $\quad T_L = 1346.4 \ x \ (5) = 6732$ Lb. In.

> *** NOTES:** T_{dc} is the torque required during acceleration.
> T_L is the torque required during constant velocity.
> N (Change in Speed) = $N_2 - N_1$
> Where N_2 = Final Speed
> N_1 = Initial Speed

16

Dynamic Torque (Brake)

11. **Determine the Reflected Inertial Torque Requirements for each component** and list them in Column 11. The sum of this column is the Total Reflected Inertial Braking Torque Requirement. The two conversion constants in the equations are (12 inches / foot) in the numerator and (307.2 foot revolution / minute second) in the denominator.

Reflected Inertial Torque $= T_i = \dfrac{WK^2 \text{ (col. 6) x N } (\textit{speed change}) \text{ x 12 x Eff. factor (col.5)}}{307.2 \text{ x t } (\textit{time})}$

Boxes $= T_i = \dfrac{(.87) \text{ x } (0 - 1800) \text{ x 12 x } (.576)}{307.2 \text{ x } .13} = -271.04$ Lb. In.

Conveyor Pulley $= Ti = \dfrac{(.06) \text{ x } (0 - 1800) \text{ x 12 x } (.72)}{307.2 \text{ x } .13} = -23.37$ Lb. In.

Etc.......

Dynamic Torque Analysis Table

1	2	3	4	5	6	7	8	CLUTCH 9	CLUTCH 10	BRAKE 11	BRAKE 12
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Coupling	1	0.78	1.0	1.000	0.78			137.11		-421.88	
10:1 Reducer	1	0.17	0.8	1.000	0.17			29.88		-91.95	
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Conveyor Pulley	20	23.89	0.8	0.720	0.06			14.65		-23.37	
Boxes	20	347.20	1.0	0.576	0.87	6732.0	584.38	265.50		-271.04	
Summation					2.37	6732.0	584.38	551.63	1136.01	-1031.94	-447.56

12. The total of columns 8 and 11 equals the Dynamic Brake Torque Required. ***NOTE:*** If this value has the same sign as the Clutch Torque it indicates that the system will decelerate in less time than assumed.

$$T_{db} = T_L + T_{ib}$$

$$T_{db} = 584.38 - 1031.94$$

$$T_{db} = -447.56 \text{ Lb. In.}$$

Holding Torque (Brake)

Dynamic Brake Torque is the torque during deceleration and is not necessarily the Holding Torque *(i.e. torque required for static equilibrium after the system has come to rest)*. First of all the inertial torque disappears after the system is stopped. Also the Reflected Load Torque is likely to change.

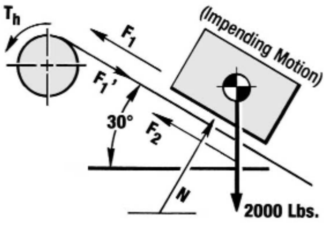
During deceleration the load torque is the same as for the acceleration phase. This is because friction always opposes the motion of the box. Once the box has stopped the friction force (F_2) acts in the other direction and therefore the Holding Torque will be different than the Load Torque. Compute the holding torque as shown below. The Holding Torque is not entered in the Dynamic Torque Analysis Table shown above.

FBD #4 (For Holding Torque)

$$F_1 = 2000 \times \sin 30° - 2000 \times \cos 30°\mu$$

$$F_1 = 653.6 \text{ Lbs.}$$

$$T_h = \frac{653.6 \times (5) \times .576}{20} = 94.1 \text{ Lb. In.}$$



Sign Convention:

The following is an explanation for the ***Sign Convention*** used in the **Dynamic Torque Analysis Procedure** presented in this Engineering Section. The procedure is set up so that the Inertial and Load Torques are considered separately. The Load Torque in this procedure is found from solving for **Static Equilibrium**. Since this is the case the sign for the Load Torque does not change sign during the analysis. In order not to arbitrarily change the sign on the Load Torque to obtain the correct Dynamic Torque the following convention has been used:

1. The direction of Torque required to accelerate the mass of the system is always considered positive.
2. Next, a Static **Free Body Diagram (FBD)** is generated to determine the Load Torque. If this torque acts in the direction of the acceleration or inertial torque then it is considered to be positive. If not, then the Load Torque is negative.
3. The Sign of the Deceleration Inertial Torque is opposite the Sign for the Acceleration Torque.

Equation 1

Dynamic Torque		Inertial Torque		Load Torque
T_d	=	T_i	+	T_L

During acceleration the **Clutch Inertial Torque** will be positive by definition. The **Load Torque** will generally be positive, especially if the load is predominately a friction or inertia load. It is possible for the **Load Torque** to be negative in some cases. This could happen if the weight of the load, or some other kind of stored energy like a compressed spring, is helping to accelerate the load. During acceleration the **Dynamic Torque** should be positive. If the **Clutch Dynamic Torque** is negative then it means that your load is capable of accelerating by itself faster than you are trying to accelerate it with the clutch.

During deceleration the **Brake Inertial Torque** will be negative by definition. The **Load Torque** will be the same as it was during acceleration unless something physically changed about the load during the process. The computed **Brake Dynamic Torque** should be negative. If the Brake Dynamic Torque turns out positive then it means that your load will stop by itself faster than the braking time you have used for the calculations.

Thermal Energy

The **Thermal Energy** required to be dissipated by the Clutch and Brake is the heat generated by the stack slipping during the engagement process. Heat transfer from the Clutch and Brake occurs by conduction through the drive plates and convection with the oil. The durability of the friction material is a function of the thermal load imposed on it. The wear rate of the material is low relatively independent of the heat generated up to a certain critical energy level, above which excessive wear will occur at a very rapid rate and limits the useful life of the drive.

The thermal load on the Clutch is different than the thermal load the Brake has to dissipate. The reason for this difference is that the Clutch is a mechanism which does work. *(i.e. the input torque acts through an angular displacement.)* The Brake on the other hand takes a torsional reaction but has no displacement. Therefore work is not done by the brake. **Thermal Energy** is also sometimes called **Kinetic Energy per Engagement.**

Clutch Kinetic Energy per Engagement

$$\text{Thermal Energy} = (.436)T_{dc} \times \left[\frac{N\ \textit{(Speed Change)}}{100}\right] \times T\ \textit{(Time)}\ \text{Ft. Lbs.}$$

$$TE_C = (.436) \times (1136.01) \times \left[\frac{1800}{100}\right] \times .4 = \mathbf{3566\ Ft.\ Lbs.}$$

Brake Kinetic Energy per Engagement

$$\text{Thermal Energy} = (.436)T_{db} \times \left[\frac{N\ \textit{(Speed Change)}}{100}\right] \times T\ \textit{(Time)}\ \text{Ft. Lbs.}$$

$$TE_B = (.436) \times (-447.56) \times \left[\frac{-1800}{100}\right] \times .13 = \mathbf{457\ Ft.\ Lbs.}$$

The units of the conversion factor are .436 (Ft.Min) / (In. Revolution Sec.)

Average Thermal Horsepower

The objective of considering the **Average Thermal Horsepower** is to determine if the steady state power level is approaching the dissipation capacity of the unit. If the **Average Thermal Horsepower** exceeds the capacity for free convection, additional cooling such as a Fan or Water Cooling is required.

$$\text{Average Thermal HP} = \frac{(TE_C + TE_B) \times CPM}{33{,}000}$$

$$THP = \frac{(3566 + 457) \times 10}{33{,}000} = \textbf{\textit{1.22 Thermal HP}}$$

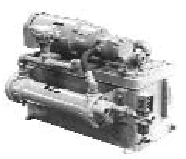
Cooling Options

Force Control provides the designer with four different **Cooling Options** to select from for most sizes of the ***Posidyne*** product group. The ***Posistop*** group features the basic or conduction cooling method as standard. The ***Positorq*** group is usually equipped with **Forced Oil Lubrication** for cooling under constant slip conditions.

Basic

Fan Cooled

Water Cooled



Forced Lube

Posidyne Clutch/Brake Selection

Use the Tables on Page 2.10 to select a fan cooled 03 *Posidyne* clutch/brake with "A" logic. The required dynamic clutch torque determines the selection in this example. In many cases, thermal horsepower will be the determining factor.

The max. dynamic clutch torque of the "A" Logic 03 *Posidyne* is 2,413 Lb. In. at the max. clutch air pressure of 80 psi. The required conveyor dynamic clutch torque of 1,136 Lb. In. is obtained by reducing the clutch air pressure with a regulator.

The dynamic brake torque of the "A" Logic *Posidyne* is 509 Lb. In. which nearly matches the calculated conveyor dynamic brake torque of 448 Lb. In. The conveyor in this example stops quickly by itself because of the 30° incline and friction. In this case it is possible to reduce cost and complexity by using "A" or "B" logic clutch/brakes which have spring set brakes. The .13 sec. stopping time produced by the "A" logic *Posidyne* is acceptable. If the stopping time needs to be controllable, then either "S" or "SA" logic would have been selected.

The static brake torque of the clutch/brake (602 Lb.In.) needs to exceed the calculated holding torque of the conveyor (94.1Lb. In.), and it does.

The max. RPM of the *Posidyne* (1800 RPM for fan cooled) matches the motor speed in the example.

The max. kinetic energy per engagement of the *Posidyne* (21,494 Ft. Lbs.) needs to exceed the value calculated for the conveyor. For the example the clutch KE per engagement is 3,566 Ft. Lbs. and the brake KE per engagement is 457 Ft. Lbs.

The average thermal horsepower of the clutch/brake needs to exceed the value calculated for the conveyor. For the example the average thermal horsepower is 1.22 HP, so the 03 *Posidyne* can be mounted either horizontally (2.8 HP) or vertically (1.4 HP). Horizontal mounting will let the clutch/brake run cooler and provide a greater margin of safety for extra boxes on the conveyor, etc.

Technical Data

English-Metric Conversion Factors

Multiply the Base Unit by the Factor shown to obtain the desired Conversion.

Measurement	Base Unit	Factor	Conversion
Length	Inch (In.)	25.4	Millimeter (mm)
	Millimeter (mm)	.03937	Inch (In.)
Torque	Pound-Feet (Lb. Ft.)	1.355818	Newton-Meter (Nm)
	Newton-Meter (Nm)	.73756	Pound-Feet (Lb. Ft.)
	Ounce-Inch (Oz. In.)	.007062	Newton-Meter (Nm)
	Newton-Meter (Nm)	141.603	Ounce-Inch (Oz. In.)
Moment of Inertia	Pound-Feet Squared (Lb. Ft.^{2})	.042	Kilogram-Meter Squared (kgm^2)
	Kilogram-Meter Squared (kgm^2)	23.81	Pound-Feet Squared (Lb. Ft.^{2})
Energy	Foot-Pound (Ft. Lb.)	1.355818	Joule (J)
	Joule (J)	.73756	Foot-Pound (Ft. Lb.)
Force	Pound (Lb.)	4.448222	Newton
	Newton	.224808	Pound (Lb.)
Power	Horsepower (HP)	.7457	Kilowatt (kW)
	Kilowatt (kW)	1.341	Horsepower (HP)
Thermal Capacity	Horsepower-Seconds per Minute (hp-sec./min.)	12.42854	Watts (W)
	Watts (W)	.08046	Horsepower-Seconds per Minute (hp-sec./min.)
Temperature	Degrees Fahrenheit (°F)	(°F-32) x 5/9	Degrees Celsius (°C)
	Degrees Celsius (°C	(°C x 9/5) + 32	Degrees Fahrenheit (°F)

Conversion Factors

Base Unit	Multiply by	To Obtain
Horsepower	60.0	hp-sec./min.
Ft.-Lb./Sec	.109	hp-sec./min.
Ft.-Lb./Min.	.0018	hp-sec./min.
In.-Lb./Sec.	.009	hp-sec./min.
In.-Lb./Min.	.00015	.hp-sec./min.

Torque & Rotary Inertia Conversion Factors

TORQUE CONVERSION TABLE

(Multiply by entry in table to convert from A to B)

A \ B BASE UNIT	dyne-cm	gm-cm	oz-in	Kg-cm	lb-in	Newton-m	lb-ft	Kg-m
dyne-cm	1	1.01972×10^{-3}	1.41612×10^{-5}	1.01972×10^{-6}	8.85073×10^{-7}	10^{-7}	7.37561×10^{-8}	1.01972×10^{-8}
gm-cm	980.665	1	1.38874×10^{-2}	10^{-3}	8.67960×10^{-4}	9.80665×10^{-5}	7.23300×10^{-5}	10^{-5}
oz-in	7.06157×10^{4}	72.0079	1	7.20079×10^{-2}	6.25×10^{-2}	7.06157×10^{-3}	5.20833×10^{-3}	7.20079×10^{-4}
Kg-cm	9.80665×10^{5}	1000	13.8874	1	0.867960	9.80665×10^{-2}	7.23300×10^{-2}	10^{-2}
lb-in	1.12985×10^{6}	1.15213×10^{3}	16	1.15213	1	0.112985	8.33333×10^{-2}	1.15213×10^{-2}
Newton-m	10^{7}	1.01972×10^{4}	141.612	10.1972	8.85073	1	0.737561	0.101972
lb-ft	1.35582×10^{7}	1.38255×10^{4}	192	13.8255	12	1.35582	1	0.138255
Kg-m	9.80665×10^{7}	10^{5}	1.38874×10^{3}	100	86.7960	9.80665	7.23300	1

ROTARY INERTIA CONVERSION TABLE

(Multiply by entry in table to convert from A to B)

A \ B BASE UNIT	gm-cm^2	oz-in^2	gm-cm-sec^2	Kg-cm^2	lb-in^2	oz-in-sec^2	lb-ft^2	Kg-cm-sec^2	lb-in-sec^2	lb-ft-sec^2 or slug-ft^2	Kg-m^2
gm-cm^2	1	5.46745×10^{-3}	1.01972×10^{-3}	10^{-3}	3.41716×10^{-4}	1.41612×10^{-5}	2.37303×10^{-6}	1.01972×10^{-6}	8.85073×10^{-7}	7.37561×10^{-8}	10^{-7}
oz-in^2	182.901	1	0.186507	0.182901	0.0625	2.59009×10^{-3}	4.34028×10^{-4}	1.86507×10^{-4}	1.61880×10^{-4}	1.34900×10^{-5}	1.82901×10^{-5}
gm-cm-sec^2	980.665	5.36174	1	0.980665	0.335109	1.38874×10^{-2}	2.32714×10^{-3}	10^{-3}	8.67960×10^{-4}	7.23300×10^{-5}	9.80665×10^{-5}
Kg-cm^2	1000	5.46745	1.01972	1	0.341716	1.41612×10^{-2}	2.37303×10^{-3}	1.01972×10^{-3}	8.85073×10^{-4}	7.37561×10^{-5}	10^{-4}
lb-in^2	2.92641×10^{3}	16	2.98411	2.92641	1	4.14414×10^{-2}	6.94444×10^{-3}	2.98411×10^{-3}	2.59009×10^{-3}	2.15840×10^{-4}	2.92641×10^{-4}
oz-in-sec^2	7.06157×10^{4}	386.088	72.0079	70.6155	24.1305	1	0.167573	7.20079×10^{-2}	6.25×10^{-2}	5.20833×10^{-3}	7.06155×10^{-3}
lb-ft^2	4.21403×10^{5}	2304	429.711	421.401	144	5.96756	1	0.429711	0.372972	3.10810×10^{-2}	4.21401×10^{-2}
Kg-cm-sec^2	9.80665×10^{5}	5.36174×10^{3}	1000	980.665	335.109	13.8874	2.32714	1	0.867960	7.23300×10^{-2}	9.80665×10^{-2}
lb-in-sec^2	1.12985×10^{6}	6.17740×10^{3}	1.15213×10^{3}	1.12985×10^{3}	386.088	16	2.68117	1.15213	1	8.33333×10^{-2}	.112985
lb-ft-sec^2 or slug-ft^2	1.35582×10^{7}	7.41289×10^{4}	1.38255×10^{4}	1.35582×10^{4}	4.63305×10^{3}	192	32.1740	13.8255	12	1	1.35582
Kg-m^2	10^{7}	5.46748×10^{4}	1.01972×10^{4}	10^{4}	3.41716×10^{3}	141.612	23.7304	10.1972	8.85073	.737561	1

Useful Formulas

Torque

$$T = \text{Force} \times \text{Radius}$$

$$T\ (\text{Lb. In.}) = HP \times \frac{63000}{N}$$

HP = Horsepower
N = Revolutions/Minute

$$T\ (\text{Lb. Ft.}) = HP \times \frac{5250}{N}$$

HP = Horsepower
N = Revolutions/Minute

Dynamic Torque (Lb. Ins.)

$$\text{Clutch} = \left[\frac{WK^2 \times N \times 12}{307.2\ ta} + T_L\right] \times \frac{1}{E}$$

$$\text{Brake} = \left[\frac{WK^2 \times N \times 12}{307.2\ td}\right] \times E + \frac{T_L}{E}$$

WK^2 = Inertia (Lb. Ft.^{2})
N = Change in RPM
ta = Accel. Time (Sec.)
td = Decel. Time (Sec.)
T_L = Load Torque (Lb. In.)
E = Efficiency

Conversion Factor = $307.2 \left(\frac{\text{Ft Rev.}}{\text{Min. Sec}}\right)$

Power

$$HP = \frac{T \times N}{63{,}000}$$

T = Torque (Lb. In.)
N = Revolutions/Minute
HP = Horsepower

$$HP = \frac{T \times N}{5250}$$

T = Torque (Lb. Ft.)
N = Revolutions/Minute
HP = Horsepower

Thermal Energy/Engagement

$$\text{Clutch: } TE_c\ (\text{Ft. Lbs.}) = (.43633) \times T_{dc} \times \left(\frac{\Delta N}{100}\right) \times t$$

$$\text{Brake: } TE_b\ (\text{Ft. Lbs.}) = (.43633) \times T_{db} \times \left(\frac{\Delta N}{100}\right) \times t$$

ΔN = Speed Change (RPM)
T_{dc} = Dynamic Clutch Torque (Lb. In.)
T_{db} = Dynamic Brake Torque (Lb. In.)
t = Time (Seconds)

Conversion Constant = $.43633 \left(\frac{\text{Ft Min.}}{\text{In. Rev. Sec.}}\right)$

Average Thermal Horsepower

$$THP = \frac{[TE_c + TE_b] \times CPM}{33{,}000}$$

TE_c = Thermal Energy (Clutch)
TE_b = Thermal Energy (Brake)
CPM = Cycles/Minute

Horsepower Sec./Min.

$$HP\ \text{Sec./Min.} = \frac{TE_b \times CPM}{550}$$

TE_b = Thermal Energy (Brake)
CPM = Cycles/Minute

Useful Formulas (Continued)

WK^2 (Inertia)

Concentrated Weight $WK^2 = WR^2$

Translating Weight $WK^2 = WR^2$

W = Weight (Lbs.)
R = Radius (Inches)

Uniform Disc $WK^2 = \left[\frac{\pi D^2}{4} \times L \times Lb./In.^3 \right] \times \frac{R^2}{2}$

D = Diameter (Inches)
L = Length (Inches)
R = Radius (Inches)

Hollow Cylinder $WK^2 = \left[\pi \frac{(Do^2 - Di^2)}{4} \times L \times Lb./In.^3 \right] \times \frac{Ro^2 + Ri^2}{2}$

Do = Outside Diameter (Inches)
Di = Inside Diameter (Inches)
Ro = Outside Radius (Inches)
Ri = Inside Radius (Inches)
L = Length (Inches)

Reflected $WK^2 = WK^2 \times \left(\frac{1}{Ratio} \right)^2$

WK^2 = Inertia

Weight of Cylinder $= \frac{\pi D^2}{4} \times L \times Lb./In.^3$

D = Diameter (Inches)
L = Length (Inches)

Specific Weight $Lb./Ft^3$

Steel 487
Cast iron 442
Aluminum............................... 169
Bronze 546

Specific Weight $Lb./In.^3$

Steel .... .282
Cast iron .... .256
Aluminum............................... .098
Bronze .... .316

Inertia Table (WK^2 of Steel Shafting and Discs)

To determine the **WK^2** of a given shaft or disc multiply the **WK^2** given below, by the length of the shaft or thickness of disc, in inches. To determine inertia of solids of greater diameter than shown below multiply the tenth of the diameter by 10^4 or move the decimal point 4 places to the right and multiply the length as above. For hollow shafts, subtract **WK^2** of the inside diameter from the **WK^2** of the outside diameter and again multiply by the length.

Per Inch of Length or Thickness

Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)	Dia. (Ins.)	WK^2 ($Lb.Ft.^2$)
0.75	0.00006	4.75	0.098	8.75	1.13	12.75	5.08	25.00	75.06	41.00	542.9	69.00	4355.3
1.00	0.0002	5.00	0.120	9.00	1.26	13.00	5.49	26.00	87.80	42.00	597.9	72.00	5163.6
1.25	0.0005	5.25	0.146	9.25	1.41	13.25	5.92	27.00	102.11	43.00	656.9	75.00	6079.5
1.50	0.001	5.50	0.176	9.50	1.57	13.50	6.38	28.00	118.10	44.00	720.2	78.00	7112.2
1.75	0.002	5.75	0.210	9.75	1.74	13.75	6.87	29.00	135.90	45.00	787.9	81.00	8271.1
2.00	0.003	6.00	0.249	10.00	1.92	14.00	7.38	30.00	155.64	46.00	860.3	84.00	9566.2
2.25	0.005	6.25	0.293	10.25	2.12	15.00	9.73	31.00	177.45	47.00	937.6	87.00	11007.8
2.50	0.008	6.50	0.343	10.50	2.34	16.00	12.59	32.00	201.48	48.00	1020.0	90.00	12606.5
2.75	0.011	6.75	0.399	10.75	2.57	17.00	16.05	33.00	227.87	49.00	1107.7	93.00	14373.2
3.00	0.016	7.00	0.461	11.00	2.81	18.00	20.17	34.00	256.77	50.00	1200.9	96.00	16319.5
3.25	0.021	7.25	0.531	11.25	3.08	19.00	25.04	35.00	288.33	51.00	1299.9	99.00	18457.1
3.50	0.029	7.50	0.608	11.50	3.36	20.00	30.74	36.00	322.73	54.00	1633.8	102.00	20798.1
3.75	0.038	7.75	0.693	11.75	3.66	21.00	37.37	37.00	360.11	57.00	2028.3	105.00	23355.0
4.00	0.049	8.00	0.787	12.00	3.98	22.00	45.01	38.00	400.64	60.00	2490.2	108.00	26140.7
4.25	0.063	8.25	0.890	12.25	4.33	23.00	53.77	39.00	444.51	63.00	3026.8	111.00	29168.5
4.50	0.079	8.50	1.00	12.50	4.69	24.00	63.75	40.00	491.88	66.00	3645.8	114.00	32452.0

WK^2 is given in $Lb.Ft.^2$. Multiply by 144 to get $Lb. In.^2$. Moving the decimal point one place in diameter shifts the decimal point in **WK^2** value 4 places in the same direction. Table is based on steel at 487 Lbs. per Cu.Ft. For materials other than steel, divide **WK^2** in table 487, and multiply by: Magnesium-109; Aluminum-169; Cast Iron-442; Brass-527; Bronze-546; Copper-555.

Motor Formulas

	3 Phase	1 Phase	Direct Current
Amps =	$\frac{HP \times 746}{1.73 \times V \times Eff \times pf}$	$\frac{HP \times 746}{V \times Eff \times pf}$	$\frac{HP \times 746}{V \times Eff}$
HP =	$\frac{1.73 \times A \times V \times Eff \times pf}{746}$	$\frac{A \times V \times Eff \times pf}{746}$	$\frac{A \times V \times Eff}{746}$

HP = Horsepower
V = Volts
Eff = Efficiency
pf = Power Factor
A = Amps

The Power Factor will be 80% and the Efficiency 80-90% for most motors.

Motor Information

Approximate Full Load Amps

HP	Three Phase 230 Volts	Three Phase 460 Volts	Single Phase 230 Volts	Direct Current 240 Volts
1/2	2	1	4.9	2.7
1	3.3	1.8	8	4.8
1-1/2	4.8	2.6	10	6.6
2	6.2	3.4	12	8.5
3	8.6	4.8	17	12.5
5	14.4	7.6	28	20
7-1/2	21	11	40	29
10	26	14	50	38
15	38	21	---	56
20	50	27	---	74
25	60	34	---	92
30	75	40	---	110
40	100	52	---	146
50	120	65	---	180
60	150	77	---	215
75	180	96	---	268
100	240	124	---	355
125	300	156	---	433
150	360	180	---	534
200	480	240	---	712

To determine Amps @ other voltages

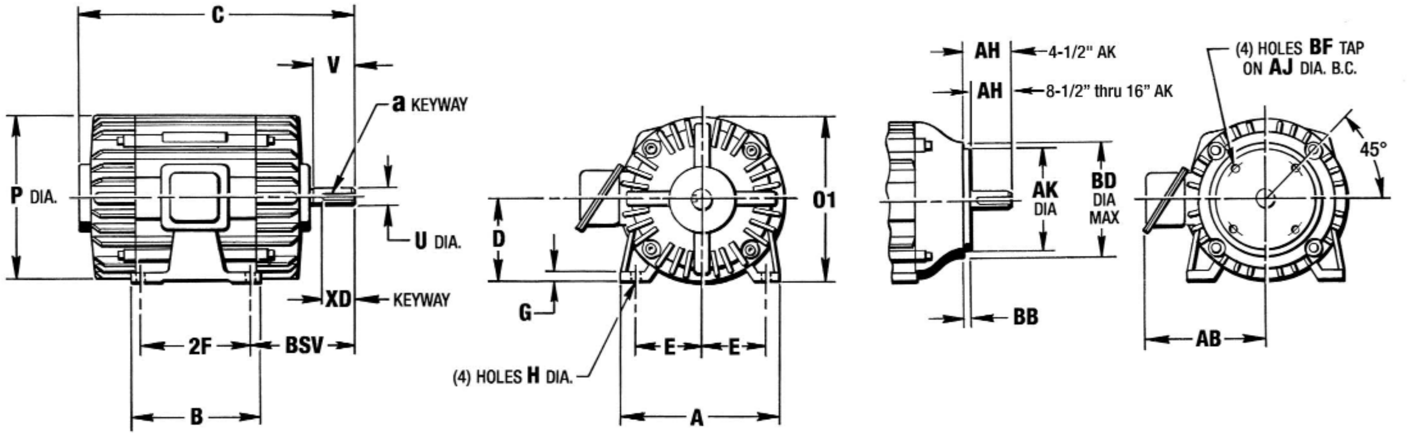
$$V = \frac{\text{Volts} \times \text{Table}}{X}$$

(X = Required Voltage)

U-Frame			T-Frame		
HP	RPM	Frame Size	HP	RPM	Frame Size
1	1800	182	1	1800	143T
1	1200	184	1	1200	145T
1.5	1800	184	1.5	1800	145T
1.5	1200	184	1.5	1200	182T
2	1800	184	2	1800	145T
2	1200	213	2	1200	184T
3	1800	213	3	1800	182T
3	1200	215	3	1200	213T
5	1800	215	5	1800	184T
5	1200	254U	5	1200	215T
7.5	1800	254U	7.5	1800	213T
7.5	1200	256U	7.5	1200	254T
10	1800	256U	10	1800	215T
10	1200	284U	10	1200	256T
15	1800	284U	15	1800	254T
15	1200	324U	15	1200	284T
20	1800	286U	20	1800	256T
20	1200	326U	20	1200	286T
25	1800	324U	25	1800	284T
25	1200	364U	25	1200	324T
30	1800	326U	30	1800	286T
30	1200	365U	30	1200	326T
40	1800	364U	40	1800	324T
40	1200	404U	40	1200	364T
50	1800	365U	50	1800	326T
50	1200	405U	50	1200	365T

Motor Dimensions

C-Face Dimensions



Frame Size	Overall Dimensions					Foot Mounting Dimensions							Shaft Extension Dim's.				C-Face Dimensions						Weight
	AB	BSV	C	O1 Max	P	A Max	B Max	D **	E	2F	G	H	a	U	V Min.	XD	AH	AJ	AK	BB	BD Max.	BF	Lbs.
56	4.91	4.63	12.50	7.50	7.44	6.50	4.00	3.50	2.44	3.00	.16	.34*	3/16 x 3/32	5/8	1.94	1.38	2.06	5.88	4.50	.16	6.50	3/8-16	25
143T	6.69	4.50	12.69	7.50	7.69	7.00	6.00	3.50	2.75	4.00	.38	.34	3/16 x 3/32	7/8	2.00	1.38	2.13	5.88	4.50	.16	6.50	3/8-16	65
145T										5.00													70
182	8.28	5.00	14.09	9.38	9.00	9.00	6.50	4.50	3.75	4.50	.44	.41	3/16 x 3/32	7/8	2.00	1.38	2.13	5.88	4.50	.16	6.50	3/8-16	70
184			15.00				7.50			5.50													70
182T	7.63	5.50	15.19	9.63			6.50			4.50			1/4 x 1/8	1-1/8	2.50	1.75	2.63	7.25	8.50	.25	8.88	1/2-13	100
184T			16.19				7.50			5.50													110
213	9.22	6.50	18.44	10.94	10.50	10.50	7.50	5.25	4.25	5.50	.50	.41	1/4 x 1/8	1-1/8	2.75	2.00	2.75	7.25	8.50	.25	9.00	1/2-13	135
215			19.94				9.00			7.00													140
213T	8.94	6.88	18.56	11.00			7.50			5.50			5/16 x 5/32	1-3/8	3.13	2.38	3.13						160
215T			20.06				9.00			7.00													175
254U	11.25	8.00	23.50	12.90	12.62	12.50	10.75	6.25	5.00	8.25	.69	.53	5/16 x 5/32	1-3/8	3.50	2.75	3.50	7.25	8.50	.25	9.00	1/2-13	240
256U			25.25				12.50			10.00													265
254T	11.38	8.25	23.25	13.00			10.75			8.25			3/8 x 3/16	1-5/8	3.75	2.38	3.75						300
256T			25.00				12.50			10.00													340
284U	11.84	9.62	26.88	14.00	14.00	14.00	12.50	7.00	5.50	9.50	.75	.53	3/8 x 3/16	1-5/8	4.63	3.75	4.63	9.00	10.50	.25	10.81	1/2-13	317
286U			27.88				14.00			11.00													372
284T	12.06	9.38	26.13	14.25			12.50			9.50			1/2 x 1/4	1-7/8	4.38	3.25	4.38						380
286T			27.69				14.00			11.00													410
324U	14.31	10.88	30.06	16.19	16.00	16.00	14.00	8.00	6.25	10.50	.88	.66	1/2 x 1/4	1-7/8	5.38	4.25	5.38	11.00	12.50	.25	12.81	5/8-11	470
326U			31.56				15.50			12.00													530
324T	14.25	10.50	29.69	16.38			14.00			10.50			1/2 x 1/4	2-1/8	5.00	3.88	5.00						600
326T			31.19				15.50			12.00													625
364U	16.44	12.25	32.63	18.09	18.00	18.00	15.25	9.00	7.00	11.25	1.00	.66	1/2 x 1/4	2-1/8	6.13	5.00	6.13	11.00	12.50	.25	13.94	5/8-11	745
365U			33.63				16.25			12.25													815

Above Dimensions are given in Inches and are for reference only and will vary by manufacturer.

* - The mounting holes on the 56 Frame Motors are .34" slots.

** - Dimension "D" will never be greater than the above values, but it may be less so that shims are usually required for coupled or geared machines. When the exact dimension is required, shims up to 1/32" may be necessary on frame sizes where "D" dimension is 8" and less, and on larger frames shims up to 1/16" may be necessary.

Unit Weights

Posidyne Clutch/Brakes

Size	Basic Weight (Lbs.)	Add Lbs. For Options					
		Fan Cooled	Water Cooled	C-Face Input	C-Face Output	Manifold Mntd. Valve	Optical Encoder
X1	42	----	----	----	----	1	2
X2	42	----	----	----	----	1	2
X3	57	----	----	----	----	1	2
1.5	32	3	----	----	----	4	2
02	84	4	22	6	5	4	2
2.5	140	5	3	8	6	4	2
03	150	8	3	10	8	4	2
05	208	10	4	----	----	4	2
10	359	15	7	----	----	4	3
11	393	Std.	----	----	----	6	3
14	412	Std.	----	----	----	6	3
20	858	26	13	----	----	6	3
30	2156	----	----	----	----	12	3

Posistop Motor Brakes

Size	**XB1**	**XB2**	**XB3**	**XB5**	**XB6**	**MB-056**	**MB-180**	**MB-210**	**MB-250**	**MB-280**	**MB-320**	**MB-440**
Weight *(Lbs.)*	31	31	45	140	150	15	40	45	100	108	160	390

Posistop Coupler Brakes

Size	XB1	XB2	XB3	XB5	XB6	056	210	250	280	320
Weight *(Lbs.)*	31	31	45	140	150	15	45	100	108	160

MagnaShear Motor Brakes

Brake Size	MSB2	MSB4	MSB6	MSB8	MSB9	MSB10	MSB12
Weight (Lbs.)	21	50	65	141	250	270	600

Unit Weights (Continued)

Assembled Brake Motor (ABM)

Brake Size	Motor Frame	Weight (Lbs.)
MB-056	56	35
	143T	55
	145T	60
	182T	94
	182U	85
	184U	85
MB-180	143T	80
	145T	85
	182T	119
	182U	110
	184U	110
MB-210	182T	124
	184T	138
	213T	180
	213U	188
	215U	203
MB-210L	213T	188
	215T	203
	254T	305
	254U	285
	256U	310

Brake Size	Motor Frame	Weight (Lbs.)
MB-250	213T	243
	215T	258
	254T	360
	254U	340
	256U	365
MB-280	254T	368
	256T	413
	284U	425
	286U	480
MB-320	284T	520
	286T	546
	324T	632
	326T	686
	324U	630
	326U	690
	364U	905
	365U	975

Electronic Controls

CONTROL	WEIGHT (Lbs.)
CLPC-LC	3.5

Foot Mounted Posistop Brakes

Size	03		05		10		11		14		20	
Type	S	T	S	T	S	T	S	T	S	T	S	T
Weight (Lbs.)	125	132	174	183	305	321	349	367	CF	CF	767	808

S - denotes a Single Unit. **T** - denotes a Tandem Unit.

E-Stop Brakes

BRAKE	WEIGHT (Lbs.)
ES-C	110
ES-D	128
ES-S	134
ES-L	160
ES-T	460
ES-M	500
ES-X	CF

CF = Consult Factory

Foot Mounted Positorq Absorber Brakes

Size	TB-03		TB-05		TB-10		TB-11		TB-14		TB-20	
Type	S	T	S	T	S	T	S	T	S	T	S	T
Weight (Lbs.)	125	132	174	183	305	321	349	367	CF	CF	767	808

S - denotes a Single Unit. **T** - denotes a Tandem Unit.

Designer's Toolbox

For over 30 years, Force Control Industries has been helping customers redesign their machinery to be more reliable, more accurate, faster and last longer. The Designer's Toolbox was developed to let you reap the benefits of our engineering efforts. We have over 100 Industry Application Notes documenting modifications to specific machinery and the benefits achieved. Visit our website at www.forcecontrol.com to see all the application bulletins.

Need Help with your application? Our Application Engineering group is ready to assist you. Our ability can be best illustrated by the wealth of applications and the loyal customers that continually comeback with more problems and applications for use to solve.

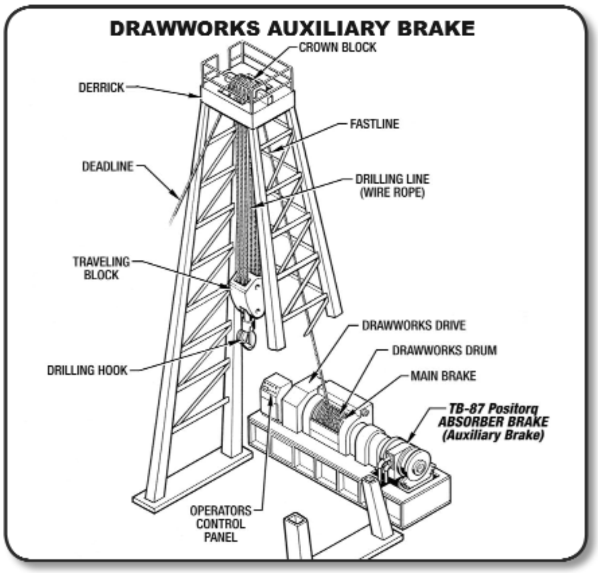
Oil Industry

APPLICATION BULLETIN

APPLICATION: Drawworks Auxiliary Brake

INDUSTRY: Oil and Gas Exploration

PRODUCT: Positorq Absorber Brake



AB-09-01 - 3/5/2005

AB-09-01 - 3/5/2005

DRAWWORKS AUXILIARY BRAKE

WHERE THEY ARE USED: A Drawworks is used on all offshore drilling rigs and many land-based drilling rigs. The Drawworks is an integral part of the hoisting system, which raises and lowers the drill string, top drive and swivel.

HOW THEY WORK: The hoisting system consists of the Drawworks, crown block, traveling block, wire rope (drilling line) and derrick. The Drawworks consists of a revolving drum around which the drilling line is spooled and unspooled - raising or lowering the hoist. When heavy loads are raised or lowered, the Drawworks brakes absorb the momentum created. The Drawworks has two different brakes - the main brake, typically band brakes or disc brakes, and the auxiliary brake, typically eddy-current or water brakes, used primarily during tripping operations. The main brake engages directing on the rim of the drum to stop the drum from hoisting or letting the drilling line out when making up or breaking out drill pipe and to control the Weight on Bit (WOB) while drilling. The auxiliary brake is used during tripping operations to control the decent speed of the drill string and to absorb the energy associated with the free fall of the drill string.

PROBLEM SOLVED: The primary brake has a tendency to grab and create a modulation thru intermitting starting and stopping of the brake. This modulation migrates down the drill string and causes a condition at the bit, called "Bit Bounce" resulting in premature wear on the drill bit. The auxiliary brakes are applied to try to correct or smooth this modulation with little success; they can't respond quick enough to changing conditions and provide little or no braking capability below 50 RPM - limiting the capability and performance of the Drawworks and control systems like an Autodriller systems.

The Force Control ***Positorq*** brake is hydraulic set for smooth continuous braking at all speeds and has the capability of full rated load at ZERO "0" RPM. The oil inside the brake shears to transmit torque while at the same time removing heat from the brake. The oil also lubricates all moving parts to prevent wear and corrosion.

IMPORTANT FEATURES:

The Positorq brake will improve rig performance and drilling capability.

- Adaptable to existing Drawworks 500 thru 5000 HP
- No battery or switch gear - freeing up space and eliminating maintenance
- Smooth continuous braking at full range of speeds extends bit life by eliminating "bit bounce" and improve the overall rate of penetration
- Full braking capability down to 0 RPM
- Net loss of over 20,000 lbs. enabling greater set back weight capabilities
- A great companion to an Autodriller system to achieve its full capability

FORCE CONTROL INDUSTRIES, INC.

3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513 868 0900 *Fax:* 513 868 2105

Mining Industry

APPLICATION BULLETIN

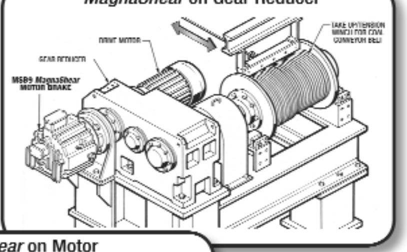
APPLICATION: Conveyor Constant Tension Winch

INDUSTRY: Mining, Tunneling

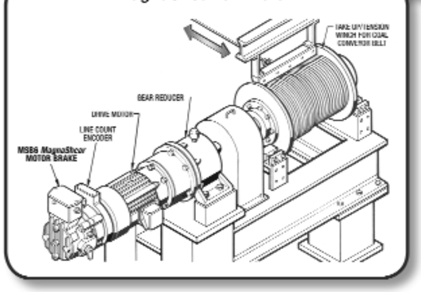
PRODUCT: *MagnaShear* Motor Brake

Conveyor Constant Tension Winch

MagnaShear on Gear Reducer



MagnaShear on Motor



AB-10-02 - 1/24/2006

AB-10-02 - 1/24/2006

CONVEYOR CONSTANT TENSION WINCH

WHERE THEY ARE USED: The Conveyor Constant Tension Winch is used extensively around the world in underground mines, underground tunnelling, and overland belt conveyors. The belt conveyors can be up to several thousand feet long and transport bulk material like coal, ore and soda.

HOW THEY WORK: The Conveyor Constant Tension Winch is used to tension the belt during startup, continuous operation, and stopping by moving an idler pulley on the slack side of the belt. The winch drive motor is controlled by an AC flux vector drive to maintain the proper belt tension. The *MagnaShear* Brake is released during normal operation, but is engaged when the conveyor is stopped and during the critical event of an emergency stop.

PROBLEMS SOLVED: Dry friction brakes are effected by the dust, dirt, water, oil, and other substances inherent to the underground and outdoor environments making them very susceptible to failure. The large coil of a dry friction brake is slow to respond. During an emergency stop these problems can cause the winch to go into a dangerous overspeed condition which can damage the winch, brake friction pads, and idler carriage. If this happens the idler no longer provides tension to the conveyor belt and the conveyor drive or backstop can't stop the load which then is dumped at the end of the conveyor.

The Force Control *MagnaShear* Brake provides smooth, reliable and quick braking under all conditions to stop the winch without losing tension in the conveyor belt during an emergency stop.

IMPORTANT FEATURES:

- Totally enclosed and sealed from outside contaminants.
- Quick response to keep winch from running away.
- Superior heat dissipation and long life friction material.
- Smooth "cushioned" stop for reduced shock to the drive system.
- Oil Shear design to provide maximum service life, with little maintenance and no adjustments.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com

Packaging Industry

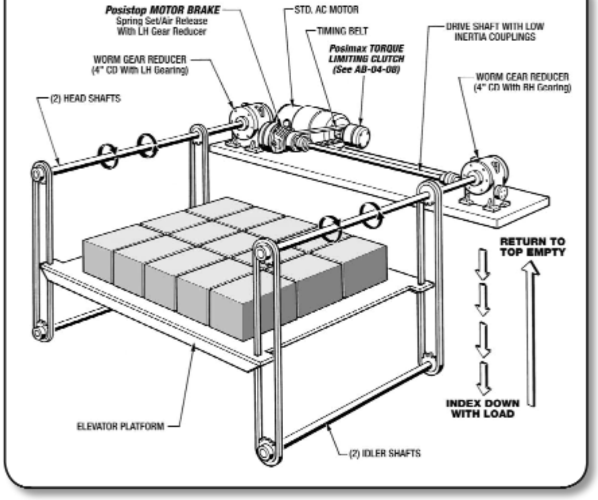
APPLICATION BULLETIN

APPLICATION: Bulk Palletizer Hoist Drive

INDUSTRY: Food, Beverage, Medical, Container and other Bulk Produced Products & Materials Palletized for Shipment

PRODUCT: *Posistop* Drive System - Includes *Posistop* Motor Brake & (2) Hollow Shaft Worm Gear Reducers plus *Posimax* Torque Limiting Clutch *(See AB-04-08)*

BULK PALLETIZER HOIST DRIVE



AB-04-07 - 2/24/2005

AB-04-07 - 2/24/2005

BULK PALLETIZER HOIST DRIVE

WHERE THEY ARE USED: Bulk material handling palletizers are used in manufacturing plants which produce soft drinks, beer, cereal, pet foods, bleach, detergent, motor oil, juice, candy, medical supplies, sugar, and other products shipped on pallets. De-palletizers are used to feed or break down the layers of palletized material. Glass, metal, or plastic containers are usually fed into a filling line or process.

HOW THEY WORK: Material, usually packed in cases, is positioned in layers on retractable slide plates just above an empty pallet on the raised hoist platform. The slide plates retract and the product drops approximately one inch onto the pallet. The hoist drive indexes down a distance equal to the height of the product, and the slide plates close. The process is repeated until a full height of layers is obtained. The full pallet is removed by other automation. An empty pallet is placed on the platform. The hoist drive raises the empty pallet and platform to the top position to start forming another full pallet.

PROBLEMS SOLVED: The ***Posistop* Drive System** combines Force Control's ***Oil Shear*** technology with a careful balance of high shock capacity worm gear geometry to produce a remarkably smooth and accurate hoist positioning drive.

1. The effect of increased stopping distance with each increase in product weight per additional layer is virtually eliminated by the improved thermal capacity & dynamic torque characteristics of the ***Posistop Oil Shear* Brake**.
2. Lipping of shallow-tray products is no longer a common problem.
3. Over-stress of lift chains due to high shock engagements of dry friction brakes is eliminated by the cushioned engagements of the ***Posistop Oil Shear* Brake**.
4. Placement of the drive motor eliminates excessive loading of the high speed drive shaft between the gear reducers.
5. Worm gear sets are cut with a carefully selected helix angle to balance load back driving forces with locking angle effects. This prevents harsh stops while permitting the oil shear brake to provide a controlled and repeatable stop for each product layer.
6. Direct mechanical connection of the spring set brake to the worm reducer input shaft minimizes dependence on other power transmission components to hold the hoist load.
7. The ***Posimax* Torque Limiting Clutch** provides jam protection for the motor, drive, and lift system. *See Application Bulletin AB-04-08 for details.*

IMPORTANT FEATURES:

- Multiple-disc, spring-set, ***Oil Shear* Motor Brake** for long service life.
- Advanced friction material and ***Oil Shear*** design provide consistent, reliable stops.
- Worm gear geometry combined with ***Oil Shear* Brake** provides smooth, controlled, & accurate positioning at each layer stop.
- Rugged and heavy construction for long service life.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com

OSB Industry

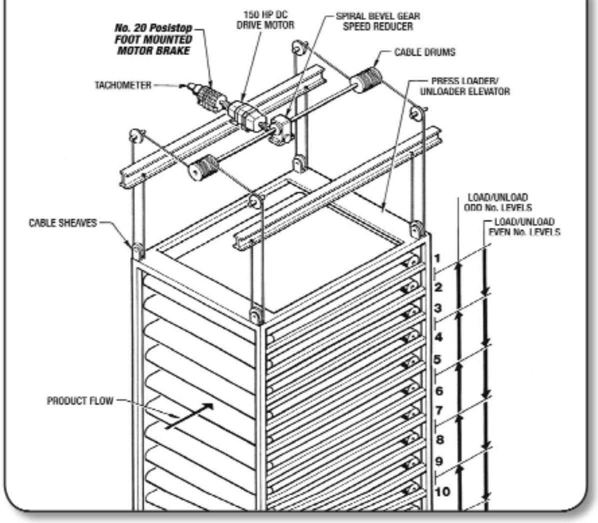
APPLICATION BULLETIN

APPLICATION: Press Loader & Unloader Elevator Drive

INDUSTRY: (OSB) Oriented Strand Board Plants

PRODUCT: Oil Shear *Posistop* Motor Brake with Tachometer

PRESS LOADER & UNLOADER ELEVATOR DRIVE



AB-02C-06 - 2/1/2005

AB-02C-06 - 2/1/2005

PRESS LOADER & UNLOADER ELEVATOR DRIVE

WHERE THEY ARE USED: The ***Press Loader Lug Chain Conveyor*** and the ***Press Lug Chain Drive*** are located in the Press area of the OSB plants. The Press Loader Lug Chain Conveyor is the first conveyor of the Press section, followed by the Press Lug Chain Drive.

HOW THEY WORK: The conveyors are driven by a DC motor with a brake and tachometer connected to auxiliary end of the motor.

Press Loader Lug Chain Conveyor: Is used to load the screens onto the Press Loader. Each time a screen exits the Pre-Load conveyor onto the Press Loader, the Press Loader Lug Chain Conveyor pulls the screen into the Loader. Then the Loader indexes to the next level to accept the next screen. There is one Press Loader Lug Chain Conveyor Drive on either side of the Press Loader. One is for the "even" levels, and one is for the "odd" levels. Since the drives are mounted to the foundation, and the loader moves up and down, the drive is not directly attached to the Loader. They transfer the torque via a crank type mechanism that is engaged each time the Loader is indexed up or down.

Press Lug Chain Drive: Is used to Load the screens into the press. This only takes place after all levels of the loader are full. When the press opens up, the Press Lug Chain Drive pulls all of the screens into the press and at the same time the Un-loader Boom pulls out all the pressed boards from the press.

PROBLEMS SOLVED: The main problem with the dry friction brakes is they mechanically fail frequently. The atmosphere around the forming line and throughout the plant is damp. The repetitive cycling of the brakes causes frequent failures. The brake and tachometer combination makes repair of the brakes difficult.

The Force Control ***Posistop*** Motor Brake puts an end to the monthly maintenance of the motor brakes. The ***Posistop*** Brakes are drop in replacement for many of the dry friction brakes that are typically used throughout the industry. The totally enclosed oil shear design of the ***Posistop*** Brake provides a totally enclosed brake that is immune to the damp atmosphere and there are no flimsy mechanical linkages to fail due to repetitive use.

The end result is a brake that will easily install to replace the typical dry friction electric brake to provide a very reliable, long life, brake with no maintenance other than an annual oil change.

IMPORTANT FEATURES:

- ***Oil Shear Technology*** gives the ***Posistop*** motor brake extremely long life, as well as consistent stopping.
- The totally enclosed design provides a brake that is not effected by harsh environments.
- Special model brakes provide drop in replacements for several of the commonly used dry friction brakes. See model FB-20-709.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com

Dimensional Lumber Industry

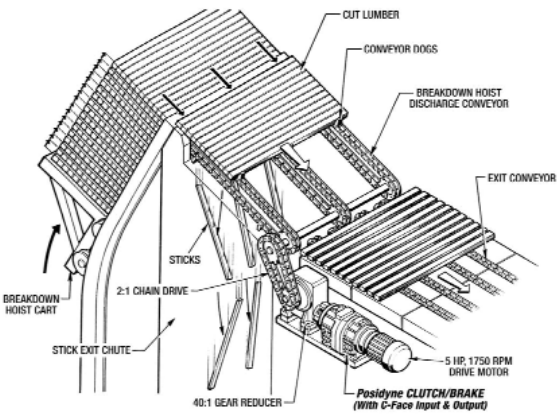
APPLICATION BULLETIN

APPLICATION: Breakdown Hoist Discharge Conveyor

INDUSTRY: Dimension Lumber Mills

PRODUCT: Oil Shear *Posidyne* Clutch/Brake

BREAKDOWN HOIST DISCHARGE CONVEYOR



AB-02A-05 - 1/26/2005

AB-02A-05 - 1/26/2005

BREAKDOWN HOIST DISCHARGE CONVEYOR

WHERE THEY ARE USED: The Breakdown Hoist Discharge Conveyor is found in dimensional lumber sawmills. It is used as an integral part of the breakdown hoist to unstack lumber a layer at a time either to be sorted or to be fed into the planer infeed system.

HOW THEY WORK: The breakdown hoist indexes up until the top layer of lumber begins to slide off on to the discharge conveyor. In this fully automated arrangement, the discharge conveyor catches and controls the action of the sliding lumber to maintain a smooth orderly descent to the take away conveyors.

Unwanted piling and jamb-ups are virtually eliminated. The sticks that separate the lumber layers automatically fall to the stick exit chute below.

PROBLEMS SOLVED:

Longevity
The breakdown hoist discharge conveyor is in-line and therefore a critical part of keeping the mill running. This chain and dog style conveyor starts and stops with each new layer of lumber to be processed.

Employing a standard motor that is allowed to run constantly and a ***Posidyne*** Clutch/Brake to provide a smooth controlled drive engagement is a key strategy to ensure long maintenance free life in all high cycle drive components. The ***Posidyne*** Clutch/Brake's totally enclosed housing and patented oil cooling techniques ensure reliable service in hot, dirty, wet and generally hostile environments.

Consistent Accuracy
Consistent timing is essential and must be maintained. Catching the lumber in a timed, orderly fashion, ensures less piling or jamb-ups. The ***Posidyne*** exhibits negligible torque changes throughout its life, or during cold start to hot run phase shift. The result of this is consistently accurate stops and starts with no adjustments required.

IMPORTANT FEATURES:

- Totally enclosed, oil cooled unit for long service life with low maintenance in the harshest environments.
- Oil Shear technology and innovative friction material provide smooth controlled torque for quick, smooth acceleration.
- Consistently accurate starts and stops with no adjustment required.
- Continuously running standard motor for long service life and lower energy consumption.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com

Metal Processing Industry

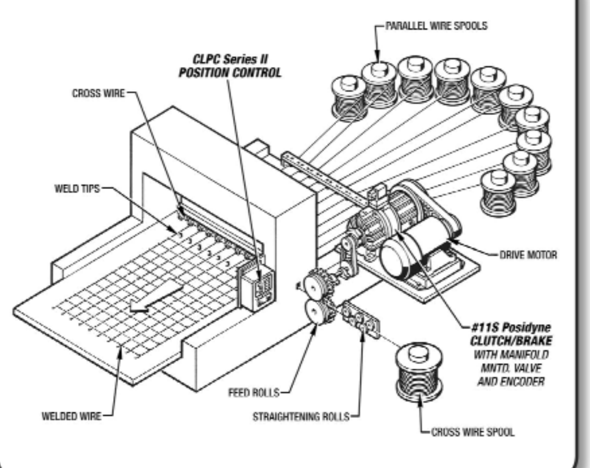
APPLICATION BULLETIN

APPLICATION: Cross Wire Feed Drive

INDUSTRY: Wire Weaving and Forming

PRODUCT: ***Posidyne*** Clutch/Brake With ***CLPC II*** Control

CROSS WIRE FEED DRIVE



AB-01-08 - 3/7/2005

AB-01-08 - 3/7/2005

CROSS WIRE FEED DRIVE

WHERE THEY ARE USED: A Cross Wire Feed Drive is used on wire weaving machines to shoot a wire across a number of parallel wires to be welded or woven.

HOW THEY WORK: A set of pinch rolls is used to index the wire strand across the incoming parallel wires. The rolls must index the wire the exact distance and stop. A ***Posidyne*** Clutch/Brake is used to accelerate the pinch rolls to full speed, and decelerate to a position stop. The ***CLPC Series II*** Closed Loop Positioning Control, by reading the encoder on the ***Posidyne*** Clutch/Brake, is used to set the length and position the stop consistently. The index length is easily set by setting the number of counts respective to the length on the front panel.

PROBLEMS SOLVED: A normal system uses an adjustable crank assembly pushing a rack, which turns a pinion, in turn rotating the pinch rolls. As the rack will oscillate forward and back two methods of release are used. One, the pinch rolls are set on a pneumatic or hydraulic lift, which separates the rolls from the wire on the reverse stroke. The other uses a single revolution cam clutch to overrun on the reverse stroke. The index length is adjusted by changing the length of the crank arm.

This is a very inefficient system in addition to having many mechanical parts, which wear, becoming loose causing length errors etc. Also mechanical changes must be changed to adjust the width of product.

The drive using the ***Posidyne*** Clutch/Brake and the ***CLPC Series II*** Closed Loop Positioning Control offers many advantages.

The ***Posidyne*** Clutch/Brake eliminates the reversing motion associated with the crank, starts and stops smoothly and accurately, eliminates many of the high wear items reducing maintenance and slop, and reduces noise considerably. The ***CLPC II*** Control continually monitors the stop position to adjust for changes in the machine and Clutch/Brake for accurate cut length. The length is easily entered on the front panel for quick changes in product width.

Improved wire length consistency permits reduction of trim cut length, reducing scrap produced.

IMPORTANT FEATURES:

- ***Oil Shear*** design provides high thermal and torque capacity for the heavy loads and high cycle rates required.
- Lubricated and cooled friction surfaces in a totally enclosed seal housing provide long service life.
- The ***CLPC Series II*** Closed Loop Positioning Control allows easy entry of wire length, and controls the ***Posidyne*** Clutch/Brake for accurate stop position.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com

Metal Forming Industry

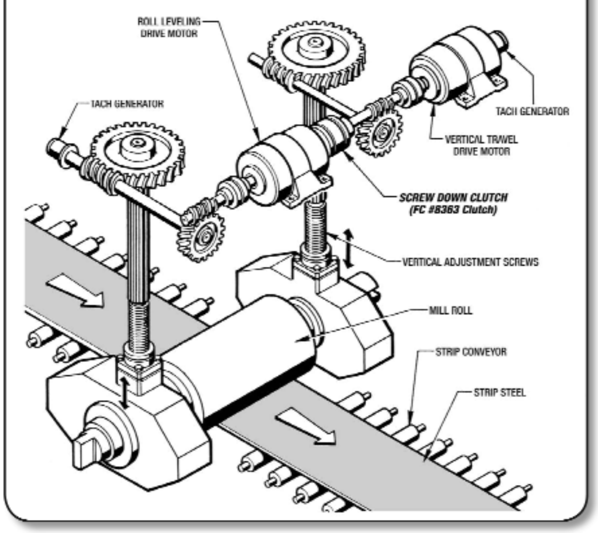
APPLICATION BULLETIN

APPLICATION: Screw Down Clutch

INDUSTRY: Hot Strip Steel Mill

PRODUCT: Oil Shear ***Posidyne*** Clutch/Brake

SCREW DOWN CLUTCH



AB-07-04 - 2/9/2005

AB-07-04 - 2/9/2005

SCREW DOWN CLUTCH

WHERE THEY ARE USED: All steel industry strip mills consist of many rough and finish rolling mills. These mills usually have two large steel rolls that are used to flatten or form the hot steel. Depending on the location and use of the mill, it may be necessary to adjust or level the large rolls. This procedure is done by using the screw down assembly located above the mill. The ***Screw Down Clutch*** is located in this assembly. The clutch can be engaged to drive both sides of the roll as one common unit, or disengaged if it is necessary to lift or lower one independent side of the roll.

HOW THEY WORK: As stated above, the clutch can be either engaged so the screw down assembly works as one unit, or disengaged if necessary to move only one side to perform the leveling procedure. The Force Control ***Screw Down Clutch*** is a thru shaft unit mounted on the common drive shaft between the two screw drive packages. With the use of a simple rotary air union, the air can be applied to release the clutch if the leveling procedure requires separate roll side adjustments.

PROBLEM SOLVED: Several different problems were solved by changing to the Force Control ***Screw Down Clutch***. The first problem was a maintenance problem. The Force Control Clutch outlasted any other type of unit by a considerable length of time. The second problem of accuracy was solved by the ***Oil Shear*** design of the Force Control Clutch. With the unique ***Oil Shear*** design the clutch gave more accurate adjustment of the rolls throughout the life of the Force Control unit. The final problem was solved due to the unique enclosed design of the Force Control unit. Because of the design, the holding torque of the unit was not affected by the outside dirty, wet, and oily environment.

IMPORTANT FEATURES:

- ***Oil Shear*** technology provides smooth and accurate engagements throughout the lifetime of the clutch.
- The totally enclosed, sealed design protects the unit from outside contaminants.
- The ***Oil Shear*** design provides the user with a long service life with very little maintenance required.

FORCE CONTROL INDUSTRIES, INC.
3660 Dixie Highway Fairfield, Ohio 45014
Phone: 513-868-0900 *Fax:* 513-868-2105
E-Mail: info@forcecontrol.com *Web:* www.forcecontrol.com