Section 15 Unit Selection Procedures

How to select a *Posidyne*

The selection of a ***Posidyne*** Clutch/Brake, PMD-2000, Multi-Speed or Reversing Drive should include the following information:

A complete selection procedure is provided in the Engineering Section.

Step 1 Control Logic

The torque transmitted by ***Force Control*** clutch and brake products is proportional to and controlled by the clamping force exerted by the piston on the stack. Pressure can be applied to the piston by internal springs or by externally controlled air or hydraulic pressure. The springs are generally used to engage the brake stack or to center the piston. The actuation pressure is used to (1) overcome the pressure generated by the springs, (2) furnish controlled pressure to the piston or (3) furnish additional pressure to assist the spring pressure.

This interaction of internal springs and externally applied pressure determines how the drive unit reacts to control commands. This is called the **Torque Control Logic**.

As you can see, to exert pressure on the stack using the actuation pressure in many cases it is necessary to first overcome the internal spring pressure on the piston. The Control Logic is very important to the torque rating of the drive unit because of a maximum allowable actuation pressure, and must be known when calculating either actuation pressure for a required torque, or actual torque at a given actuation pressure.

Control Logic Availability

Logic	Description	X1	X2	X3	1.5	02	2.5	03	05	10	11	14	20	30
S	Air set clutch / Light spring set brake with air assist	X	X	X	X	X	X	X	X	X	X	X	X	X
SA	Air set clutch / Medium spring set brake with air assist	X	X	X	X	X	X	X	X	X	X	X	X	X
A	Air set clutch / Medium spring set brake	X	X	X	X	X	X	X	X	X	X	X	X	X
B	Air set clutch / Heavy spring set brake	----	----	----	X	X	X	X	X	X	X	X	X	X
C	Air set clutch / No brake	X	X	X	X	X	X	X	X	X	X	X	X	X
SCP	Spring centered piston / Air set clutch / Air set brake	----	----	----	----	----	X	X	X	X	X	X	X	X
P	Air set clutch / Air set brake / No springs	X	X	X	X	X	X	X	X	X	X	X	X	X

All clutches are air engaged. The Logic Selection is determined by the type of brake required.

S-Logic is for applications requiring a wide range of torque adjustment.
Example: Indexing conveyors typically require a very low brake torque. This is because there is a lot of friction, low back driving efficiency of worm gear reducers and the product may slip on the conveyor belt if stopped too quickly.

SA-Logic has the safety advantages of the A-Logic with an air assist to further increase brake torque if required.
Example: Lumber tilt hoist drive. The spring set brake holds the load if the air supply fails. Air assist allows operator to adjust desired operating torque.

A and B-Logic is for lifting devices or applications where adjustable brake torque is not required.

> Caution: B-Logic (Heavy spring set brake) may have too much brake torque and may damage connected equipment

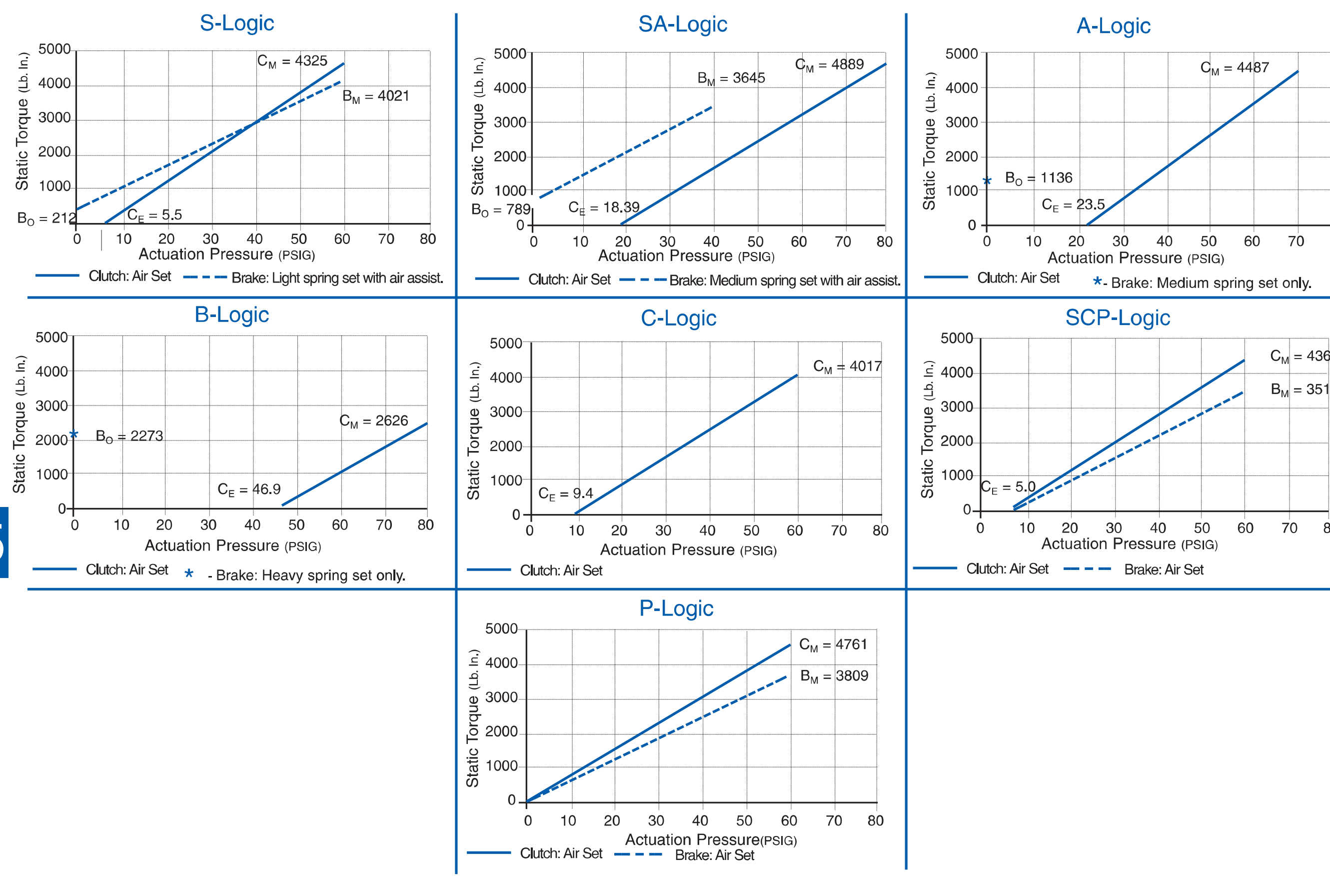
Example: Indexing cam operated dial table. The cam profile will stop the dial table. The ***Posidyne*** Brake only stops the drive train. A-Logic (Medium spring set brake) will not allow operator to increase brake torque which could damage the cam and cam followers. Check with cam manufacturer for maximum allowable brake torque.

C-Logic is used when no brake is required. If there is very little connected load, the output shaft may rotate when the clutch is released due to residual drag
Example: Inching drive. Motor and gear reducer connected to ***Posidyne*** input. ***Posidyne*** used for jogging machine. Separate ***Posidyne*** **Clutch/Brake** used as main drive on machine. (Consider ***Posidyne*** **Multi-Speed Drive** for this application.)

SCP-Logic is for applications that require a neutral position where neither the clutch or the brake is engaged.
Example: Lathe, where the chuck must be manually rotated to insert or remove the part.

P-Logic has an absence of spring bias. Both the clutch and brake are only engaged when air pressure is applied. This P Logic has the longest life since there are no springs that can fatigue and fail.
Example: Horizontal indexing belt conveyor and other applications where the torque control range is very low.

Sample Pressure vs. Torque Curves for 05 *Posidyne*.



Step 2 Static Torque

Static torque is the torque which can be transmitted by a clutch or brake without slipping. The static torque rating can be used to make a preliminary unit size selection using the **Quick Selection Table** following these four simple steps.

1. Determine Control Logic Type.

2. Determine Motor Horsepower.

3. Determine Speed (RPM) @ ***Posidyne***.

4. Select preliminary unit size under required logic type using Horsepower vs. RPM Selection Tables below.

EXAMPLE:

- 15 HP Drive Motor.
- 1800 RPM @ ***Posidyne*** input.
- Adjustable Torque Control for both the clutch and brake is required.

A preliminary selection can be made from the chart under "S" Logic, 15 Horsepower and 1800 RPM. A size 03 is found.

For applications under 900 RPM and over 1800 RPM input consult the Force Control factory.

Quick Reference *Posidyne* Selection Tables

S, C, P and SCP Logic

HP	Input RPM 900	1200	1500	1800
1/3	X1	X1	X1	X1
1/2	X1, X2	X1	X1	X1
3/4	X2, 1.5, 02	X1, X2	X1	X1
1	X3, 1.5, 02	X2,1.5, 02	X2,1.5,02	X1, X2
1 1/2	X3, 1.5, 02	X2,X3,02	X2,1.5,02	X2,1.5,02
2	X3, 1.5, 02	X3,1.5,02	X3,1.5,02	X2,1.5,02
3	2.5	X3,X4,2.5	X3,1.5,02	X3,1.5,02
5	2.5	X4,2.5	2.5	X3, 2.5
7 1/2	03	2.5	2.5	X4,2.5
10	03	03	2.5	X4,2.5
15	05	03	03	03
20	10	05	05	03
25	10	05	05	05
30	10	10	05	05
40	10	10	10	10
50	11	10	10	10
60	11	11	10	10
75	11	11	11*	10
100	14	11	11*	11*
125	20	14	11*	11*
150	30*	20	14*	11*
200	30*	30*	20	14*
250	30*	30*	---	20

* Requires external cooling system.

B Logic

HP	Input RPM 900	1200	1500	1800
1/3	---	---	---	---
1/2	---	---	---	---
3/4	1.5	---	---	---
1	1.5, 02	1.5, 02	1.5, 02	1.5, 02
1 1/2	1.5, 02	1.5, 02	1.5, 02	1.5, 02
2	2.5	1.5, 02	1.5, 02	1.5, 02
3	2.5	X4,2.5	2.5	1.5, 02
5	03	X4,2.5	2.5	2.5
7 1/2	03	03	2.5	X4,2.5
10	05	03	03	X4,03
15	10	05	05	03
20	10	10	05	05
25	11	10	10	10
30	11	10	10	10
40	14	11	11*	10
50	14	14	11*	11*
60	20	14	11*	11*
75	20	20	14*	14*
100	---	20	20	14*
125	---	---	20	20
150	---	---	---	20
200	---	---	---	---
250	---	---	---	---

* Requires external cooling system.

A and SA Logic

HP	Input RPM 900	1200	1500	1800
1/3	X1	X1	X1	X1
1/2	X1, X2	X1	X1	X1
3/4	X2, 1.5, 02	X1, X2	X1	X1
1	X3, 1.5, 02	X2,1.5,02	X2,1.5,02	X1, X2
1 1/2	X3, 1.5, 02	X2,X3,02	X2,1.5,02	X2,1.5,02
2	X3,1.5, 02	X3,1.5,02	X3,1.5,02	X2,1.5,02
3	2.5	X3,X42.5	X3,1.5,02	X3,1.5,02
5	2.5	X4,2.5	2.5	X3, 2.5
7 1/2	03	2.5	2.5	X4,2.5
10	03	03	2.5	X4,2.5
15	05	03	03	03
20	10	05	03	03
25	10	05	05	05
30	10	10	05	05
40	10	10	10	10
50	11	10	10	10
60	11	11	10	10
75	14	11	11*	10
100	20	14	11*	11*
125	30*	20	14*	14*
150	30*	20	20	14*
200	30*	30*	20	20
250	30*	30*	---	---

* Requires external cooling system.

Selections on the above Tables are based on static torque only and were calculated from the following Formula:

$$\textbf{Static Torque} = \frac{\textbf{HP x 63,000}}{\textbf{N}} \textbf{ x 3.00}$$

Where: Torque = In. Lbs.
HP = Rated motor horsepower.
N = RPM @ ***Posidyne*** input
3.00 = Approximate stall torque of motor.

Step 3 Dynamic Torque

Dynamic Torque is the torque required during engagement to accelerate or decelerate the rotating mass (Inertia) and overcome friction (Efficiency) and load torque within a specified time period. Each of these can have a positive or negative effect on the required dynamic torque capacity of the clutch or the brake and will not necessarily effect both in the same way. Therefore it is necessary to calculate both the ***Clutch Dynamic Torque*** and the ***Brake Dynamic Torque*** separately. For a complete explanation of Dynamic Torque, Load Torque and Enertia refer to Engineering Information on page 16.1.

Clutch

$$T_{dc} = \left\{ \frac{WK^2 \times N \times 12}{308 \times t_a} + T_L \right\} \times \frac{1}{E}$$

Example:
Calculate required dynamic torque for the clutch.

$$T_{dc} = \left\{ \frac{7.7 \times 1800 \times 12}{308 \times 0.75} + 330 \right\} \times \frac{1}{0.91} = \mathbf{1154\ lb\ in.}$$

Brake

$$T_{db} = \left\{ \frac{WK^2 \times N \times 12}{308 \times t_d} - T_L \right\} \times E$$

Example:
Calculate required dynamic torque for the brake.

$$T_{dc} = \left\{ \frac{7.7 \times 1800 \times 12}{308 \times 0.75} - 330 \right\} \times 0.91 = \mathbf{355\ lb\ in.}$$

Where:
T_{dc} = Clutch Dynamic Torque required (Lb. In.)
T_{db} = Brake Dynamic Torque required (Lb. In.)
WK^2 = Inertia. (Lb. Ft.^{2})
N = RPM @ ***Posidyne.***
t_a = Acceleration time. (Sec.)
t_d = Deceleration time. (Sec.)
E = Efficiency of drive train.
*T_L = Load torque. (Lb. In.)

Select a unit size with dynamic ratings exceeding the values calculated.

Step 4 Thermal Energy per Engagement

Thermal Energy per Engagement is the amount of energy to be dissipated by the Posidyne during each engagement and/ or brake. This thermal energy requirement may be calculated using the following formula only if the beginning RPM of the clutch and the ending RPM of the brake is zero (0) RPM. For additional information on all beginning and ending speeds see Engineering Information on page 16.8.

Clutch

$$TE_c = 1.7 \times WK^2 \times \left\{ \frac{N}{100} \right\}^2 \times \frac{T_{dc}}{T_{dc} - T_L}$$

Example:
Calculate total required energy per engagement capacity for the clutch.

$$TE_c = 1.7 \times 7.7 \times \left\{ \frac{1800}{100} \right\}^2 \times \frac{1144}{1144-330} = \mathbf{5{,}961\ ft\ lbs}$$

Brake

$$TE_b = 1.7 \times WK^2 \times \left\{ \frac{N}{100} \right\}^2 \times \frac{T_{db}}{T_{db} + T_L}$$

Example:
Calculate total required energy per engagement capacity for the brake.

$$TE_b = 1.7 \times 7.7 \times \left\{ \frac{1800}{100} \right\}^2 \times \frac{346}{346+330} = \mathbf{2{,}171\ ft\ lbs}$$

Where:
TE_c = Clutch Thermal Energy per Engagement (Ft. Lbs.)
TE_b = Brake Thermal Energy per Engagement (Ft. Lbs.)
1.7 = Constant
WK^2 = Inertia. (Lb. Ft.^{2})
N = RPM @ ***Posidyne.***
T_{dc} = Clutch Dynamic Torque (Lb. In.)
T_{db} = Brake Dynamic Torque (Lb. In.)
*T_L = Load torque. (Lb. In.)
(System friction, inclined or vertical loads, etc.)

NOTE: For additional information to calculate WK^2 see Engineering Information on Page 16.3.
***NOTE:** Be sure to use the proper sign (+ -). Usually it is a positive number for the clutch calculation and a negative number for the brake calculation.

Step 5 Thermal Horsepower

Posidynes are also rated on Average Thermal Horsepower capacity which is the amount of thermal energy the units can dissipate continually (1 THP = 42.2 BTU = 33,000 Ft. Lbs.) based on the type of cooling - basic, fan, water or forced lube. The average thermal horsepower rating required can be calculated using the following formula.

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

Example:

Calculate thermal horsepower capacity required.

$$THP = \frac{(5{,}961 + 2{,}171) \times 6}{33{,}000} = 1.48\ THP$$

Where: THP = Average Thermal Horsepower

TE_c = Thermal Energy per Engagement (Clutch)

TE_b = Thermal Energy per Engagement (Brake)

CPM = Cycles per Minute

33,000 = Constant

A cooling method must be specified that provides an Average Thermal Horsepower rating exceeding the values calculated.

Step 6 Overhung Load

Overhung Load is the load attribute to the pull of the chain or belt drive on the input or output shafts. Larger diameter pulleys or sprockets will decrease overhung load but will increase the WK^2 in the preceding calculations. The required Overhung Load capacity can be calculated from the following formula.

$$P = \frac{126{,}000 \times HP \times K}{N \times D}$$

Example:

Calculate overhung load capacity required for the input shaft.

$$P = \frac{126{,}000 \times 15 \times 1.25}{1800 \times 4} = 328 \text{ Lbs.}$$

Where: P = Overhung Load. (Lbs.)

HP = Horsepower.

N = RPM @ ***Posidyne*** Input.

D = Pitch diameter of the Pulley or Sprocket. (In.)

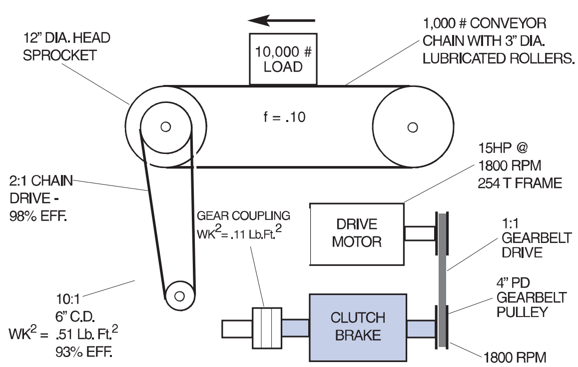
K = 1.1 for the Chain.

K = 1.25 for the gearbelt.

K = 1.5 for a V-Belt.

The unit size selected must have overhung load ratings exceeding the values calculated.

Example: Indexing Conveyor Drive



Data Required for Proper Selection...

Logic Type	=	S
CPM	=	6 Cycles per Minute
t_a (Accel. Time)	=	.75 Seconds
t_d (Decel. Time)	=	.75 Seconds
K (Gearbelt Drive)	=	1.25

Cyclic WK^2

$$\text{Reflected Load } WK^2 = \text{Load} \times (\text{Radius})^2 \times \left(\frac{1}{\text{Ratio}}\right)^2 \div 144$$

$$= 10{,}000 \times (6")^2 \times \left(\frac{1}{2 \times 10}\right)^2 \div 144 = 6.25 \text{ Lb. Ft.}^2$$

$$\text{Reflected Conveyor} = \text{Chain} \times (\text{Radius})^2 \times \left(\frac{1}{\text{Ratio}}\right)^2 \div 144$$

$$\text{Chain } WK^2 = 1{,}000 \times (6")^2 \times \left(\frac{1}{2 \times 10}\right)^2 \div 144 = 0.63 \text{ Lb. Ft.}^2$$

Reducer WK^2 = Vender Information = 0.51 Lb. Ft.^{2}

Coupling WK^2 = Vender Information = 0.11 Lb. Ft.^{2}

03 *Posidyne* WK^2 = From Table on Page 1-11 = 0.20 Lb. Ft.^{2}

Total Cyclic WK^2 @ *Posidyne* = 7.7

Efficiency of Drive Train

E = Chain Drive Efficiency x Reducer Efficiency = .98 x .93 = .91 Efficiency

Total Load Torque

$$T_L = [(10{,}000 \text{ Lbs.} + 1{,}000 \text{ Lbs.}) \times .1_f] \times 6" \text{ Rad.} \div 20{:}1 = \mathbf{330\ Lb.\ In.}$$

Selection

Referring to the ***Posidyne*** selection tables we find under S- logic that the size 03 (1864 Lb.In. @ 60 PSI) is required to meet the Dynamic Torque requirements of the clutch. The rating for the brake (1485 Lb. In. @ 60 PSI) exceeds the Dynamic Brake requirement. The average Thermal Horsepower rating indicates the need for fan cooling. A check of the Overhung Load rating for the 03 (1150 Lbs.) in the Overhung Load table on page 2.16 is satisfactory.

The proper ***Posidyne*** selection is a size 03, fan cooled and S-logic. Using the Ordering System Chart in Section 2 page 2.17 the Ordering Number is developed.

03-1S1-H-5

If a piggyback is required the Ordering Number becomes:

03-7S7-H-5 (254 T-Frame)

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Posidyne® Logic Specifications

Pressure vs. Static Torque Chart

(X Class Posidyne Clutch/Brake)

Size	Logic	Clutch C_M Max. Clutch Static Torque *(Lb.In.)*	Clutch C_M Max. Air Actuation Pressure *(PSIG)*	Clutch C_E Clutch Engmt.Air Pr. Req'd. *(PSIG)*	Clutch C_T Clutch Net Torque *(Lb. In./ PSIG)*	Brake B_O Spring Set Only-Torque w/o Air Assist *(Lb.In.)*	Brake B_M Max. Brake Static Torque *(Lb.In.)*	Brake B_M Max. Brake Actuation Pressure *(PSIG)*	Brake B_S Brake Spring Bias *(PSIG)*	Brake B_T Brake Net Torque *(Lb. In./ PSIG)*
X1	S	99	70	16	1.83	24	106	45	13.1	1.83
	SA	90	80	31		49	104	30	26.8	1.83
	A	90	80	31		49	------------------		----	-----
	C	99	70	16		----	------------------		-----	-----
	P	110	60	3		----	110	60	------	1.83
X2	S	198	70	16	3.86	48	213	45	13.1	3.66
	SA	179	80	31		98	208	30	26.8	3.66
	A	179	80	31		98	------------------		----	----
	C	198	70	16		----	------------------		----	----
	P	220	60	3		----	220	60	----	3.66
X3	S	468	80	16	7.32	93	531	60	12.6	7.32
	SA	359	80	31		189	480	40	25.6	7.72
	A	359	80	31		189	------------------		----	----
	C	468	80	16		----	------------------		----	----
	P	512	70	3		----	512	70	----	7.32

NOTES:
1. For Dynamic torque ratings multiply static torque ratings above by .846 for all X Class *Posidyne* Clutch/Brakes.
2. "S", "SA" and "C" logics are not standard.

Posidyne® Logic Specifications

Pressure vs. Static Torque Chart *(Sizes 1.5-05 Posidyne Clutch/Brake)*

		Clutch				Brake				
		C_M		C_E	C_T	B_O	B_M		B_S	B_T
Size	Logic	Max. Clutch Static Torque *(Lb.In.)*	Max. Air Actuation Pressure *(PSIG)*	Clutch Engmt.Air Pr. Req'd. *(PSIG)*	Clutch Net Torque *(Lb. In./ PSIG)*	Spring Set Only-Torque w/o Air Assist *(Lb.In.)*	Max. Brake Static Torque *(Lb.In.)*	Max. Brake Actuation Pressure *(PSIG)*	Brake Spring Bias *(PSIG)*	Brake Net Torque *(Lb. In./ PSIG)*
	S	427	60	4.2		32	484	60	3.3	7.6
	SA	387	70	18.8		110	492	70	15.0	7.6
1.5	A	387	70	18.8	7.6	110	------------------		----	-----
	B	240	70	37.6		220	------------------		----	-----
	C	427	60	4.2		----	------------------		-----	-----
	P	464	70	3.0		----	464	70	0	7.6
	S	518	60	7.30	9.8	48	553	60	5.7	8.4
	SA	542	80	24.9		164	501	40	19.5	8.4
02	A	503	80	20.0		126	------------------		----	----
	B	386	80	40.0	8.4	252	------------------		----	----
	C	335	60	20.0		----	------------------		----	----
	P	590	60	3.0	9.8	----	505	60	0	8.4
	S	1,331	60	6.6		113	1,396	60	5.3	21.4
	SA	1,482	80	21.9		512	1,663	40	18.0	28.5
	A	1,451	80	20.6		476	------------------		----	----
2.5	B	968	80	41.2	25	952	------------------		----	----
	C	1,270	60	9.1		----	------------------		----	----
	SCP	1,234	60	2.5		----	1,051	60	-3.5	21.4
	P	1,497	60	3.0		----	1,283	60	0	21.4
	S	2,574	60	6.0		144	2,049	60	4.5	31.7
	SA	2,790	80	21.4		651	2,238	40	16.4	39.7
	A	2,852	80	20.0		602	------------------		----	----
03	B	1,895	80	40.2	47.6	1203	------------------		----	----
	C	2,474	60	8.0		----	------------------		----	----
	SCP	2,668	60	4.0		----	1,833	60	-2.0	31.7
	P	2,857	60	3.0		----	1,905	60	0	31.7
	S	4,325	60	5.5		212	4,021	60	3.3	63.5
	SA	4,889	80	18.4		789	3,645	40	11.0	71.42
	A	4,487	80	23.5		1,196	------------------		----	----
05	B	2,626	80	46.9	79.4	2,273	------------------		----	----
	C	4,017	60	9.4		----	------------------		----	----
	SCP	4,362	60	5.0		----	3,518	60	-4.6	63.5
	P	4,761	60	3.0		----	3,809	60	0	63.5

NOTE: For Dynamic torque ratings multiply static torque ratings above by .846.

Pressure vs. Static Torque Chart

(Sizes 10-30 Posidyne Clutch/Brake)

Size	Logic	Clutch				Brake				
		C_M		C_E	C_T	B_O	B_M		B_S	B_T
		Max. Clutch Static Torque (Lb.In.)	Max. Air Actuation Pressure (PSIG)	Clutch Engmt.Air Pr. Req'd. (PSIG)	Clutch Net Torque (Lb. In./ PSIG)	Spring Set Only-Torque w/o Air Assist (Lb.In.)	Max. Brake Static Torque (Lb.In.)	Max. Brake Actuation Pressure (PSIG)	Brake Spring Bias (PSIG)	Brake Net Torque (Lb. In./ PSIG)
	S	9,832	60	7.3		691	10,489	60	4.2	163.3
	SA	9,471	80	29.2		2,766	9,297	40	16.9	163.3
	A	10,031	80	26.3		2,797	-----------------		----	----
10	B	5,097	80	52.7	186.6	5,593	-----------------		----	----
	C	9,228	60	10.5		----	-----------------		----	----
	SCP	9,936	60	6.7		----	8,621	60	-7.3	163.3
	P	11,197	60	3.0		----	9,797	60	0	163.3
	S	18,045	80	8.0		888	14,926	80	5.0	175.5
	SA	13,358	80	27.0		2,961	9,980	40	16.9	175.5
	A	14,036	80	24.0		2,661	-----------------		----	----
11	B	8,019	80	48.0	250.7	5,322	-----------------		----	----
	C	18,045	80	8.0		----	-----------------		----	----
	SCP	17,833	80	9.0		----	17,833	80	-4.6	175.5
	P	20,054	80	3.0		----	14,038	80	0	175.5
	S	22,989	80	9.4		1,681	23,737	80	6.0	275.7
	SA	16,484	80	29.4		5,237	16,264	40	19.0	275.7
	A	17,576	80	26.0		4,660	-----------------		----	----
14	B	10,783	80	47.0	325.8	8,352	-----------------		----	----
	C	23,453	80	8.0		----	-----------------		----	----
	SCP	26,066	80	8.8		0	20,793	80	-4.6	275.7
	P	24,279	80	4.3		0	22,056	80	----	275.7
	S	31,082	80	8.0		2,018	32,274	80	5.3	378.2
	SA	25,837	80	20.2		5,045	20,173	40	13.3	378.2
	A	26,332	80	19.0		4,759	-----------------		----	----
20	B	18,087	80	38.0	432.2	9,518	-----------------		----	----
	C	30,455	80	9.5		----	-----------------		----	----
	SCP	32,737	80	4.8		----	28,115	80	-6.2	378.2
	P	34,578	80	3.0		----	30,256	80	0	378.2
	S	78,857	50	7.9		8,010	72,185	40	5.0	1604.4
	SA	75,478	60	19.7		20,026	68,157	30	12.5	1604.4
30	A	75,478	60	19.7	1871.7	20,026	-----------------		----	----
	C	78,857	50	7.9		----	-----------------		----	----
	SCP	76,600	45	4.0		----	65,657	45	4.1	1604.4
	P	74,871	40	3.0		----	64,175	40	0	1604.4

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NOTE: For Dynamic torque ratings multiply static torque ratings above by .846 for all sizes.

To find Torque Developed at a given Actuation Pressure.

$$\text{Clutch Torque} = (\text{PSI} - C_E) \times C_T$$
$$\text{Brake Torque} = (\text{PSI} + B_S) \times B_T$$

To find Actuation Pressure needed for Req'd. Torque.

$$\text{Clutch PSI} = (T_R / C_T) + C_E$$
$$\text{Brake PSI} = (T_R / B_T) - B_S$$

How to Select the Correct Posistop

Selecting the correct ***Posistop*** **Brake** is very similar to the selection procedures used for the brake component selection of the ***Posidyne***. **Note:** One major difference is that Torque Ratings are in ***Lb. Ft.*** rather than ***Lb. Ins***.

Selection of the ***Posistop*** products vary slightly with the different types.

Motor Mounted

The **Motor Mounted *Posistop*** is spring set, air release. The initial selection for a motor brake should be made based on the frame size of the motor. It is important to check the shaft diameter, shaft length, pilot diameter and bolt circle to select the proper size ***Posistop.*** This selection should be checked against the Torque and Thermal Requirements. The formulas are provided for this purpose.

Flange Mounted

The **Flange Mounted *Posistop*** is spring set, air release and comes in the same Torque Ratings as the Motor Mounted Brakes. The flanges and hubs are typically designed to fit various machine faces and shaft extensions.

Foot Mounted

The **Foot Mounted *Posistop*** comes both as spring set, air release (Type A and B Logics) and as an adjustable unit with air set, spring release (Type S Logic). The Control Logic must be determined before a selection can be made.

The following formulas can be used to calculate the required Torque and Thermal Ratings.

Static Torque

The **Static Torque Ratings** of the ***Posistop*** units represent the **Holding Torque Capacity** with the actuation pressure noted in the "S" Logic units or when zero release pressure is supplied, i.e., multiple springs fully engaging the brake stack in the "A" or "B" Logic units. **Static Torque may be determined by the following formula:**

$$T_S = \frac{HP \times 5250 \times 2}{N}$$

Where: Ts = Torque (Lb. Ft.)
HP = Horse Power of Motor
N = RPM @ Brake
2 = Constant for moderate inertial loads.
(Consult factory for hoist type applications)

Select a motor brake with a Static Torque Rating equal to or greater than the Torque Value determined above.

Dynamic Torque

For applications requiring precise stopping action, it becomes necessary to consider the torque available to decelerate the load.

$$T_{db} = \text{Torque (Lb. Ft.)} = \left\{ \frac{WK^2 \times N}{308 \times t} + T_L \right\} \times E$$

WK^2 = Lb. Ft.^{2} (Total Cyclic Inertia)
N = RPM
t = Stopping Time required in seconds
308 = Constant
T_L = Load Torque (Lb. Ft.)
E = Efficiency

The Torque figure in the preceding formula is considered to be average torque available during deceleration. Select a Brake with a Dynamic Torque Rating equal to or greater than the Torque determined above.

Thermal Dissipation-Cyclic Drive

The considerations for Heat Dissipation requirements are based on the following formula:

$$TE_b = 1.7 \times WK^2 \times \left\{ \frac{N}{100} \right\}^2 \times \frac{T_{db}}{T_{db} - T_L}$$

TE_b = Thermal Energy per Engagement (Ft. Lbs.)
1.7 = Constant
WK^2 = Total Cyclic Inertia (Lb. Ft.2)
N = RPM @ Brake
T_{db} = Dynamic Torque (Lb. Ft.)
T_L = Load Torque (Lb. Ft.)

This formula gives the **Thermal Energy absorbed by the brake in any one engagement**, providing all factors remain constant for the application. **Force Control *Posistop*** units are rated thermally in terms of **Horsepower / Seconds per Minute**.

1 HP Sec. /Min. = 0.7 BTU = 550 Ft. Lbs./Min.

Horsepower Seconds per Minute is a continuous rating based on the cyclic rate of the application and the kinetic energy to be absorbed per stop.

$$\text{HP Sec. /Min.} = \frac{TE_b \times CPM}{500}$$

HP Sec./Min. = Thermal Load (Continuous)
CPM = Cycles per Minute
550 = Constant

Select a ***Posistop*** **Brake** with a Thermal Rating equal to or greater than the Thermal Load (Continuous) determined above.

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How to Select Your Positorq Absorber Brake

Dynamometer Application

In a **Dynamometer Application** normally a ***Positorq* Brake** is used to resist rotation of a shaft at some torque load. It can be used to absorb energy continuously as in product life testing, or for a short time for maximum load carrying capability. It can also be used to lock-up the shaft for destructive testing.

The ***Positorq* Brake** size is based primarily on torque and thermal horsepower. Sizing of the ***Positorq* Brake** should be done by application engineers at ***Force Control***, however the following information will be required for sizing and determining cooling systems.

1. The first step is to determine the **maximum continuous slip torque** required at any speed. This is the torque at which the ***Positorq* Brake** is required to slip, absorbing energy continuously.
2. The next step is to determine the **lock-up or holding torque** required. This could be used for destructive testing or maximum load carrying ability.
3. Next determine the **maximum heat load (Thermal Horsepower)** to be dissipated. This can usually be determined by the maximum horsepower of the input driver. It can also be calculated by using torque and speed.
4. Determine the **maximum speed** in RPM required at the ***Positorq* Brake.**
5. Determine the **minimum torque** required at the maximum speed. Due to residual drag in the ***Positorq* Brake**, zero torque is not available depending on speed and ***Positorq*** size. Minimum torque is affected by brake size, number of discs, RPM, fluid flow and temperature.

BASIC FORMULA
TORQUE (Lb. Ft.)

$$TORQUE = \frac{(HP)\ (5250)}{RPM}$$

$$THP = \frac{(Torque)\ (RPM)}{5250}$$

$$RPM = \frac{(THP)\ (5250)}{Torque}$$

HP = Prime mover horsepower less efficiency and work losses in system or device being tested.

RPM = Speed at brake shaft.

THP = Thermal horsepower to be dissipated. (Continuous)

Torque = Lb. Ft.

NOTE: Please consult factory for your particular application. There are many options available that are not shown in this catalog.

How to Select Your *Positorq* Absorber Brake

Tension Application

The ***Positorq*** can be used for two major types of applications:

(1) To supply constant tension for unwind applications in industries such as paper, foil, steel, coating, plating, etc.

(2) As an energy absorber (Dynamometer) to create a known controlled load on a system, usually for testing purposes.

The selection of a ***Positorq*** for a tension control application will vary depending on the type of tension application. The most common types of applications are:

(1) Constant Rewind - The parent roll is continuously unwound as for plating, coating, laminating, etc.

(2) Rewinding in Sets - The parent roll is wound onto several smaller rolls. In this application the parent roll will need to be stopped several times during the operation to change the smaller rolls.

For a Constant Rewind application the two Torque Requirements to be considered are:

(1) The Torque to maintain constant tension.

(2) The Torque to stop the enertia of the roll for a **panic stop**.

When Rewinding in Sets, there are three Torque Requirements to be considered.

(1) The Torque to maintain constant tension.

(2) The Torque to stop the roll Inertia only, (E-Stop).

(3) The Total Torque to stop the roll Inertia and overcome tension at the first set stop.

The Total Torque Required when stopping at the first set stop is: (1) the total of the torque to maintain tension plus **(2)** the torque to stop the roll.

The larger of (1) The Total Torque required at set stop or (2) The Torque required for a Panic Stop, should be used for selection purposes.

Determining Torque Capacity...

The following formulas are used to find the required torque capacity. Calculating the torque of a full roll and that at set stop is the same except for the difference in WK^2 and RPM.

(1) Tension Torque

1. Torque required to maintain constant tension.

$$Tt = \frac{D \times W \times (PLI) \text{ or } (PSI \times t)}{2 \times 12}$$

Tt = Torque (Lb.Ft.) to maintain tension.
D = Dia. of Roll. (Inches)
W = Width of Roll. (Inches)
PLI = Tension (Lb./Linear Inch)
PSI = Tension (Lb./Sq. Inch)
t = Thickness

(2) Stopping Torque

2. Torque required to stop roll.

$$Ts = \frac{WK^2 \times RPM}{308 \times t}$$

Ts = Torque for panic or set stop. (Lb.Ft.)
WK^2 = Inertia of roll when stopping.
RPM = Speed of roll when stopping.
308 = Constant
t = Time to decelerate.

(3) Total Torque

3. Total Torque required at set stop.

$$TT = Tt + Ts$$

TT = Total Torque at set stop. (Lb.Ft.)
Tt = Constant Tension Torque at set stop. (Lb.Ft.)
Ts = Stopping Torque at set stop. (Lb.Ft.)

Thermal Horsepower to maintain tension

$$THP_t = \frac{W \times PLI \times FPM}{33{,}000}$$

THP_t = Thermal Horsepower (Tension)
W = Width (Inches)
PLI = Tension (Lb. per Linear Inch)
FPM = Feet per Minute
33,000 = Constant

The correct ***Positorq*** unit is then selected based on the highest torque requirement and thermal horsepower rating. The Cooling Unit is selected based on Thermal Horsepower.

Thermal Horsepower relates to the amount of energy that must be absorbed, and is used to size the cooling system. Usually the thermal horsepower absorbed to maintain tension is satisfactory for calculating **Cooling System Capacity**, however in some cases where a very small PLI is required the thermal energy to stop the roll may be the limiting factor. The following formula can be used to determine the Thermal Horsepower. Refer to the Engineering Section for further information if selection of Stopping Thermal Horsepower is required.

***
Tension Brake Selection for Unwind Applications

The primary function of the ***Positorq* unit** in **Unwind Applications** is to provide a controlled resistance to the parent roll so the web tension remains constant as the roll diameter changes. Constant web tension is required to produce a satisfactory roll that has uniform hardness. For this to occur the torque reaction of the Brake will vary proportionally to the change in roll size as it unwinds. The angular velocity of the roll, however, varies inversely to the change in its size when web speed is held constant. These requirements produce a constant horsepower condition. In other words, as the torque requirement for the brake decreases, the speed of the roll increases such that the product of the torque and speed is a constant.

Unwind Applications can be broken down into two typical categories.

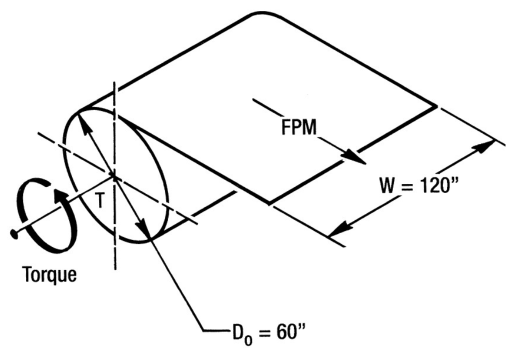
1. Constant Rewind Applications

When the Parent Roll is being rewound for processing or storage and the process is not interrupted. This condition is common to applications found in the steel industry

2. Rewind Applications with Multiple Sets

When a parent roll is being rewound into several smaller rolls for shipment or storage. This condition is common to applications found in the paper industry.

There are small but important differences between the applications. The following example is used to illustrate them. **The two parameters on which the proper selection is made in both applications is torque and horsepower requirements**



Unwind Example

Required Information

1. Web Speed = FPM = 6000
2. Web Tension = PLI = 5
3. Web Width = W = 120 Inches
4. * Specific Weight of Paper = 45 Lbs./Ft.^{3} (80 Lb. paper x .007” thick)
5. Max. Diameter = D_0 = 60 Inches
6. ** Number of Set = 4
7. Deceleration Time between Sets = t_s = 30 Sec.
8. Panic Stop Time = t_p = 15 sec.
9. Core Diameter = D_c = 10 Inches

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NOTES:

* Paper is normally specified by a “base weight” (i.e. 500 sheets having an area of 6 Ft.^{2} equals a certain weight.) In order to determine the specific weight of the paper the thickness or caliper of the paper has to be obtained

Example: 80 Lb. paper that is .007” Thick

$$\text{Specific Weight} = \frac{80 \text{ Lbs.}}{[3000] \times [.007/12]} = 45 \text{ Lbs. Ft.}^3$$

** Sets - Number of rolls that the parent roll will be wound.

*** The above selection procedure is valid for applications of web speeds no greater than 6000 FPM and with web tension of 5 PLI or greater.

Consult our Factory Sales Engineering Department for applications outside these limits.

Torque Requirements...Tension Brake (Positorq) Selection

The Maximum Torque Requirement for an Unwind Tension Application can be determined by evaluating the following (3) categories:

1. Torque Required at Constant Speed Operation

For **Steady State Operation** this value is simply found by using the following equation:

$$T_C = \frac{D \times W \times PLI}{2} \times \frac{1}{12}$$

Using the example on the previous page.

$$T_C = \frac{60 \times 120 \times 5}{2} \times \frac{1}{12} = \mathbf{1500\ Ft.\ Lbs.}$$

A torque reaction of 1500 Ft. Lbs. is required for Steady State Operation at full roll.

2. Torque Required for Panic Stop

The **Required Torque for a Panic Stop** is a maximum when the roll is at its largest diameter. The following equations are used to determine this required torque:

A. Determine Weight of Roll

W_t = Volume x Specific Weight

$$W_t = \left[\frac{\pi D_2}{4} \times L\right] \times \left[\frac{1}{1728}\right] \times \text{Specific Weight}$$

$$W_t = \left[\frac{\pi\, 60^2}{4} \times 120\right] \times \left[\frac{1}{1728}\right] \times 45\ \text{Lbs./Ft.}^3 = \mathbf{8835\ Lbs.}$$

B. Determine WK² of Roll @ Max. Dia.

$$WK^2 = \frac{WR^2}{2}\left[\frac{1}{144}\right]\left[\text{Lb. Ft.}^2\right]$$

$$WK^2 = \frac{8835\,[30]^2}{2}\left[\frac{1}{144}\right] = \mathbf{27{,}609\ Lb.\ Ft.^2}$$

C. Determine RPM @ Max. Dia.

$$RPM = \frac{FPM \times 12}{\pi \times D}$$

$$RPM = \frac{[6000] \times [12]}{\pi \times [60]} = \mathbf{382\ RPM}$$

D. Torque Requirement

$$T_p = \frac{WK^2 \times RPM}{308 \times t_p}$$

$$T_p = \frac{27{,}609 \times 382}{308 \times 15} = \mathbf{2283\ Lb.\ Ft.}$$

3. Torque Required Between Sets

The **Maximum Torque Required between Sets** is the first set. This torque is determined as follows:

A. Determine Weight of Roll @ First Set

$$W_{t1} = W_o\left[1 - \frac{1}{N}\right] \textit{ where N = Number of Sets}$$

$$W_{t1} = 8835\left[1 - \frac{1}{4}\right] = 6{,}626.25\ \text{Lbs.}$$

B. Determine Dia. of Roll @ First Set

$$D_1 = \sqrt{\left[D_O^2 - D_C^2\right]\left[1 - {}^1/_N\right]}$$

$$D_1 = \sqrt{\left[60^2 - 10^2\right]\left[1 - {}^1/_4\right]} = \mathbf{51.2\ Inches}$$

C. Determine WK² @ First Set

$$WK^2 = \frac{WR^2}{2}\left[\frac{1}{144}\right] = \text{Lb. Ft.}^2$$

$$WK^2 = \frac{[6626]\,[25.6]^2}{2}\left[\frac{1}{144}\right] = \mathbf{15{,}078\ Lb.\ Ft.^2}$$

D. Determine RPM @ First Set Dia.

$$RPM = \frac{FPM \times 12}{\pi \times D}$$

$$RPM = \frac{[6000] \times [12]}{\pi \times 51.2} = 448\ RPM$$

E. Determine Stopping Torque (Inertia Only)

$$T_s = \frac{WK^2 \times RPM}{308 \times t_s}$$

$$T_s = \frac{15{,}078 \times 448}{308 \times 30} = 731\ \text{Lb. Ft.}$$

F. Total Torque Requirement

$$T_T = \left(T_c \times \frac{D_1}{D_0}\right) + T_s$$

$$T_T = \left(1500 \times \frac{51.2}{60}\right) + 731 = 2011\ \text{Lb. Ft.}$$

From this application the torque requirement at constant speed is the limiting requirement. Model 8245 Positorq is tentatively selected based on maximum torque. See next page for sizing Thermal Horsepower Requirements.

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Thermal Horsepower...Tension Brake (Positorq) Selection

The next parameter to evaluate in selecting the correct Positorq is the Thermal Horsepower Requirement. Each of the three modes of operation analyzed under Torque Requirements will have a different thermal demand. To make a proper selection each mode needs to be evaluated.

1. Thermal Power - Constant Speed

The Thermal Horsepower Requirement during this mode of operation can be determined from the following equation:

$$THP = \frac{W \times PLI \times FPM}{33{,}000}$$

$$THP = \frac{(120)\ (5)\ (6000)}{33{,}000} = 109\ THP$$

2. Thermal Power - Panic Stop

$$THP_p = \frac{T_p}{550}\left[(.1047)\ RPM - \frac{T_p}{WK^2}(t)\right]$$

$$THP_p = \frac{2283}{550}\left[(.1047 \times 382) - \frac{2283}{27{,}609}(0)\right] = 166\ THP$$

T_p = Panic Stopping Torque = 2283 Ft. Lbs.

WK^2 = @ Maximum Diameter = 27,609 Lb. Ft.^{2}

RPM = @ Maximum Diameter = 382

t = Time

NOTE: Thermal load is maximum at **t = 0** and decreases linearly to zero when the system has come to rest.

3. Thermal Power - Set Stop

$$THP = \frac{T_T}{550}\left[(.1047)\ RPM - \frac{T_T - T_C}{WK^2}(t)\right]$$

$$THP = \frac{2011}{550}\left[(.1047 \times 448) - \frac{2011\text{-}1500}{15{,}078}(0)\right] = 171\ THP$$

T_T = Total Torque between Sets = 2011 Lb. Ft.

WK^2 = First Set Diameter = 15,078 Lb. Ft.^{2}

RPM = @ First Set Diameter = 448

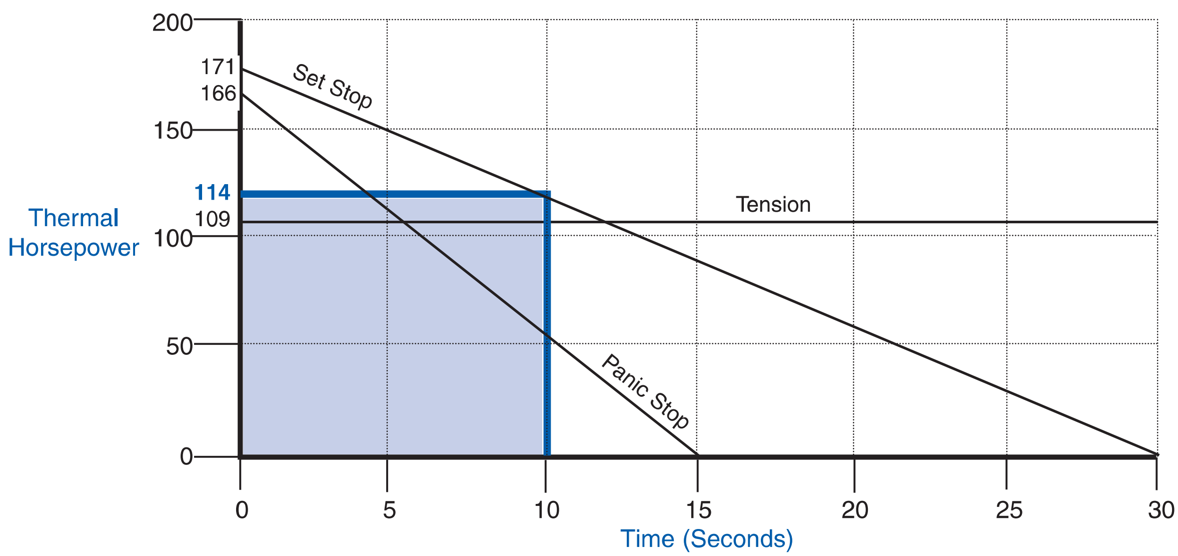
t = Time

NOTE: Once again the Maximum Thermal Load is at the beginning of deceleration and goes to zero as the system comes to rest.

Since the thermal demand deceleration between sets is relatively transient, it would not be necessary to purchase a 200 THP Cooler. The optimum selection for the unit under consideration would be determined by finding the Thermal Capacity at which the transient load would not exceed the unit rating for more than 10 seconds.

$$THP = THP\ Peak\left[1 - \frac{10}{Decel\ Time}\right]$$

$$= 171\left[1 - \frac{10}{30}\right] = 114\ THP$$



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.

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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 **WK^2.** The units for this term are **Lb. Ft.^{2}**. There are two primary activities required to obtain the **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.

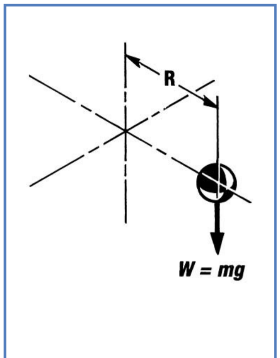
The steps to find the **WK²** of any object are:

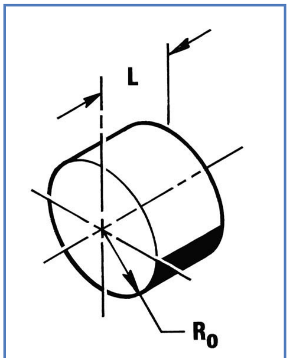
1. Determine the weight of the object.

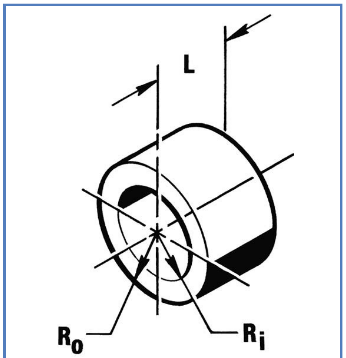
2. Determine K² from the geometry of the object.

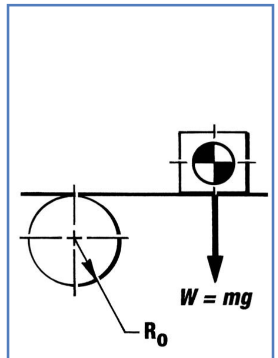
3. Multiply the two terms together.

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	$W = \frac{\pi D^2}{4} \times L \times Lb/In^3$	$W = \frac{\pi (D_0^2 - D_1^2)}{4} \times L \times Lb/In^3$	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

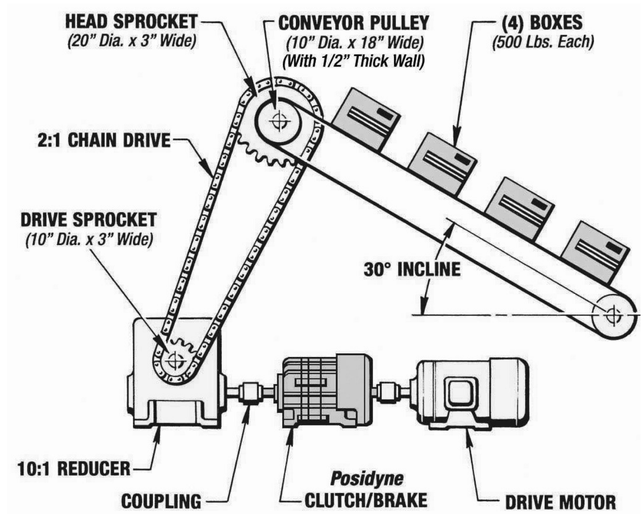
Example:

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

$WK^2 = \text{Weight} \times R_O^2 \times \text{No. of Boxes}$ — 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 @ *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

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

Conveyor Pulleys

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

$WK^2 = \text{Weight} \times \left(\frac{R_O^2 + R_i^2}{12}\right) \times \text{No.}$ — Total inertia of both pulleys

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

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

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

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

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}**	WK^2 = **.78 Lb. Ft.^{2}**	03 WK^2 = **.20 Lb. Ft.^{2}**
(Information from Vendor)	(Information from Vendor)	(Information from Vendor)

Total System Reflected Inertial Torque

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

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.**

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

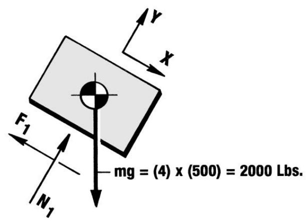
$$T_{dc} = 584.38 + 551.63$$

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

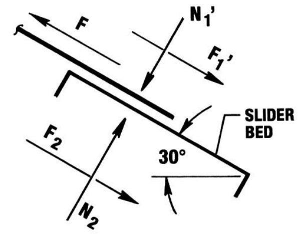
16

Solving for Static Equilibrium

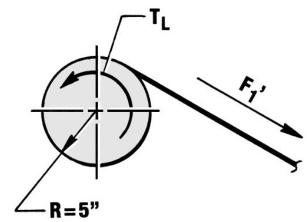
Example - FBD #1 - Box



Example - FBD #2 - Slider Bed & Belt



Example - FBD #3 - Head Pulley



FBD #1 - Box

$\Sigma F_x = 0$

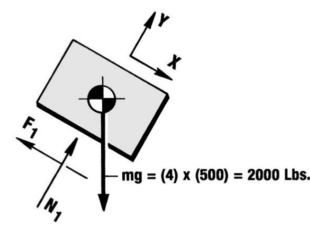
$0 = W \times \sin 30° - F_1$

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

$\Sigma F_y = 0$

$0 = N_1 - W \times \cos 30°$

EQ. 11.2 → $N_1 = W \times \cos 30°$



FBD #2 - Slider Bed & Belt

EQ. 11.3 → $\Sigma F_x = 0$

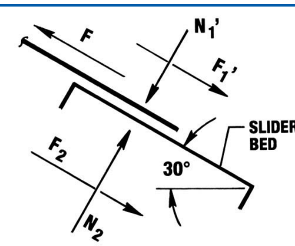
$F_1' + F_2 = F$

$\Sigma F_y = 0$

EQ. 11.4 → $N_1' = N_2$

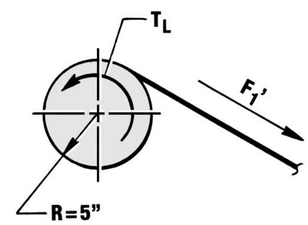
EQ. 11.5 → $F_2 = N_2\mu = N_1'\mu$

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



FBD #3 - Head Pulley

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



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

EQ. 11.7 → $F = W \times \sin 30° + W \times \cos 30° \mu$

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

Substitute 11.7 into 11.6 $T_L = 1346.4 \times (5) = 6732$ 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 $= T_i = \dfrac{WK^2 \text{ (col. 6)} \times N \text{ (\textit{speed change})} \times 12 \times \text{Eff. factor (col.5)}}{307.2 \times t \text{ (\textit{time})}}$

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

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

Etc.......

Dynamic Torque Analysis Table

1	2	3	4	5	6	7	8	CLUTCH		BRAKE	
								9	10	11	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

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.

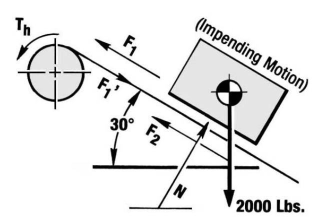
During deceleration the load torque is the same as for the acceleration phase. This is because friction always opposes the motion of the box. Once the box has stopped the friction force (F_2) acts in the other direction and therefore the Holding Torque will be different than the Load Torque. Compute 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 T_d	=	Inertial Torque T_i	+	Load Torque 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

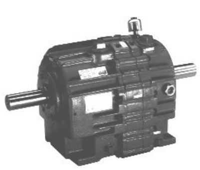
The objective of considering the **Average Thermal Horsepower** is to determine if the steady state power level is approaching the dissipation capacity of the unit. If the **Average Thermal Horsepower** exceeds the capacity for free convection, additional cooling such as a Fan or Water Cooling is required.

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

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

Cooling Options

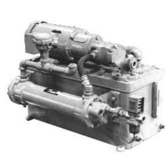
Force Control provides the designer with four different **Cooling Options** to select from for most sizes of the ***Posidyne*** product group. The ***Posistop*** group features the basic or conduction cooling method as standard. The ***Positorq*** group is usually equipped with **Forced Oil Lubrication**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

BASE UNIT (A) \ B	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)

BASE UNIT (A) \ B	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 $\mathbf{WK^2}$ of a given shaft or disc multiply the $\mathbf{WK^2}$ 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 $\mathbf{WK^2}$ of the inside diameter from the $\mathbf{WK^2}$ 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

$\mathbf{WK^2}$ 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 $\mathbf{WK^2}$ value 4 places in the same direction. Table is based on steel at 487 Lbs. per Cu.Ft. For materials other than steel, divide $\mathbf{WK^2}$ 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

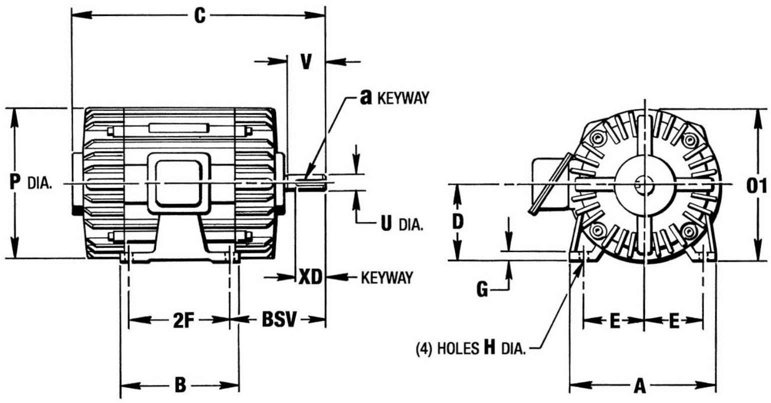
To determine Amps @ other voltages

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

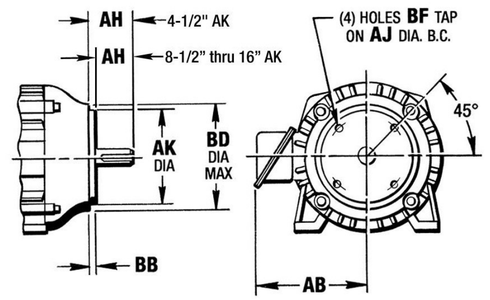
(X = Required Voltage)

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 Lbs.
	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	
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.

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.

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

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

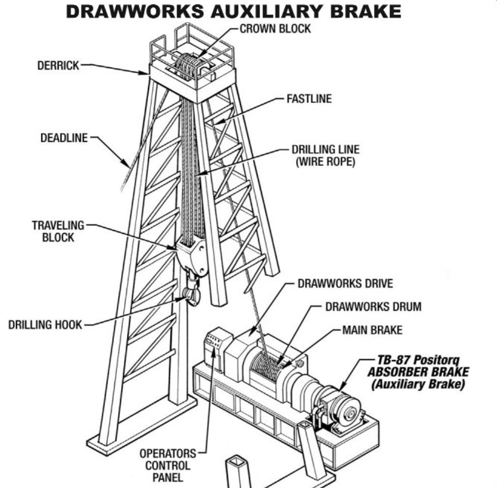
APPLICATION BULLETIN

APPLICATION: **Drawworks Auxiliary Brake**

INDUSTRY: **Oil and Gas Exploration**

PRODUCT: ***Positorq*** **Absorber Brake**

DRAWWORKS AUXILIARY 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

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Mining Industry

APPLICATION BULLETIN

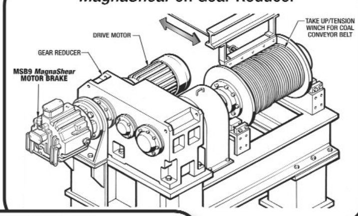
APPLICATION: Conveyor Constant Tension Winch

INDUSTRY: Mining, Tunneling

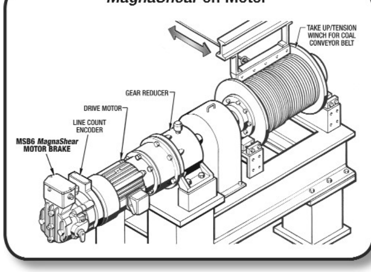
PRODUCT: *MagnaShear* Motor Brake

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.

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Packaging Industry

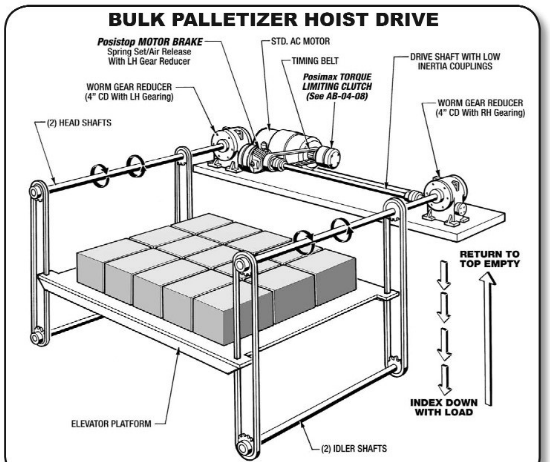
APPLICATION BULLETIN

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.

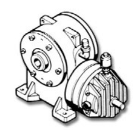
HOW THEY WORK: Material, usually packed in cases, is positioned in layers on retractable slide plates just above an empty pallet on the raised hoist platform. The slide plates retract and the product drops approximately one inch onto the pallet. The hoist drive indexes down a distance equal to the height 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.



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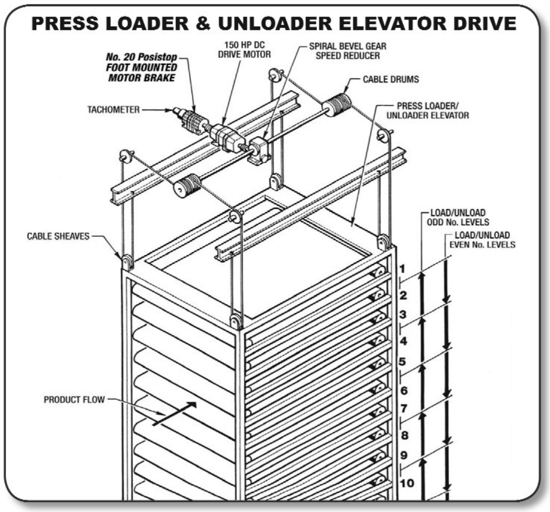
OSB Industry

APPLICATION BULLETIN

APPLICATION: **Press Loader & Unloader Elevator Drive**

INDUSTRY: **(OSB) Oriented Strand Board Plants**

PRODUCT: **Oil Shear *Posistop* Motor Brake with Tachometer**



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.

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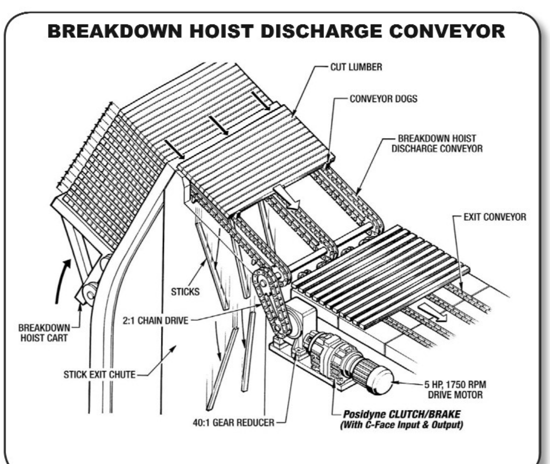
Dimensional Lumber Industry

APPLICATION BULLETIN

APPLICATION: **Breakdown Hoist Discharge Conveyor**

INDUSTRY: **Dimension Lumber Mills**

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



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.

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Metal Processing Industry

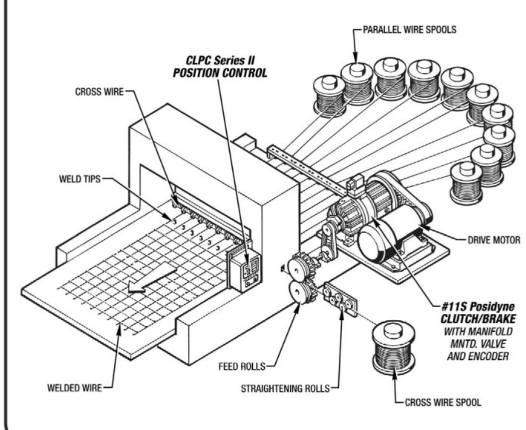
APPLICATION BULLETIN

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.

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Metal Forming Industry

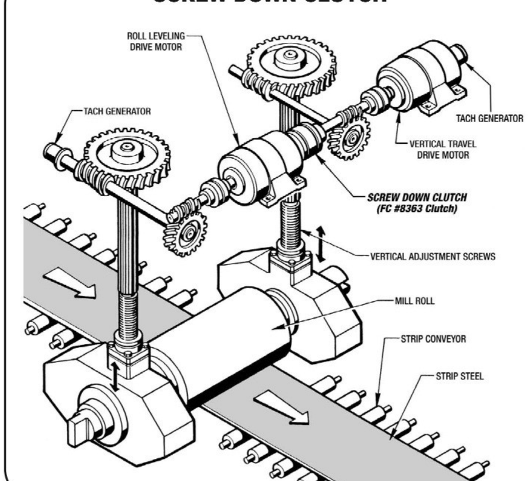
APPLICATION BULLETIN

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.

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Special 24 Month Warranty

Force Control Limited Warranty

Upon written approval of the application by Force Control Industries, Inc. **the standard Warranty period will be extended to 24 months from date of shipment.**

Force Control Industries, Inc. ("Force Control") warrants its products to be free from defects in material and workmanship under normal and proper use for a period of one year from the date of shipment. Any products purchased from Force Control that upon inspection at Force Control's factory prove to be defective as a result of normal use during the one year period will be repaired or replaced (at Force Controls' option) without any charge for parts or labor. This limited warranty shall be void in regard to (1) any product or part thereof which has been altered or repaired by a buyer without Force Control's previous written consent or (2) any product or part thereof that has been subjected to unusual electrical, physical or mechanical stress, or upon which the original identification marks have been removed or altered. Transportation charges for shipping any product or part thereof that the buyer claims is covered by this limited warranty shall be paid by the buyer. If Force Control determines that any product or part thereof should be repaired or replaced under the terms of this limited warranty it will pay for shipping the repaired or replaced product or part thereof back to the buyer. EXCEPT FOR THE EXPRESS WARRANTY SET OUT ABOVE, FORCE CONTROL DOES NOT GRANT ANY WARRANTIES EITHER EXPRESSED OR IMPLIED, INCLUDING IMPLIED WARRANTIES OF MERCHANTABILITY OR FITNESS FOR USE. The warranty obligation set forth above is in lieu of all obligations or liabilities of Force Control for any damages. Force Control specifically shall not be liable for any costs incurred by the buyer in disconnecting or re-installing any product or part thereof repaired or replace under the limited warranty set out above. FORCE CONTROL EXPRESSLY EXCLUDES ALL LIABILITY FOR ANY INDIRECT OR CONSEQENTIAL DAMAGES THE BUYER MAY SUSTAIN IN CONNECTION WITH THE DELIVERY, USE, OR PERFORMACE OF FORCE CONTROL PRODUCTS. Under no circumstances shall any liability for which Force Control is held responsible exceed the selling price to the buyer of the Force Control products that are proven to be defective. This limited warranty may be modified only in writing signed by a duly authorized officer of the company. This limited warranty applies exclusively to Force Control products; warranties for motors and gear reducers and other component parts may be provided by their respective manufactures. Any legal action for breach of any Force Control warranty must be commenced within one year of the date on which the breach is or should have been discovered.

A Return Goods Authorization (RGA) number must be obtained from the factory and clearly marked on the outside of the package before any equipment will be accepted for warranty work. Force Control will pay the shipping costs of returning the owner parts that are covered by warranty.

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All Products Catalog

APC-2011

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X-Class *Posidyne* Clutch/Brake

Posidyne Clutch/Brake

MagnaShear Electric Brake

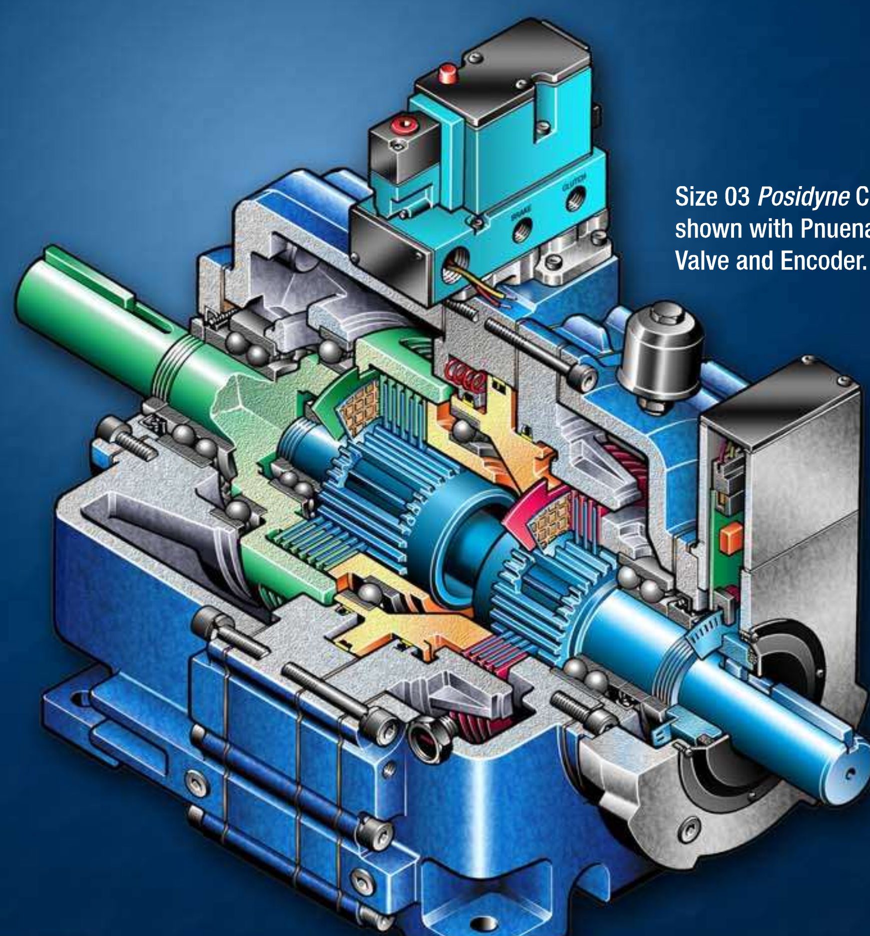
Positorq Absorber Brake

CLPC-LC Position Control

PosiDrive Servo Systems

Custom Drive Systems

Posistop Motor Brake



Size 03 *Posidyne* Clutch/ Brake shown with Pnuenatic Control Valve and Encoder.

Worldwide Leader In Oil Shear Technology