SOM PAC® 1 SERIES
CLUTCH/BRAKE DRIVE SYSTEMS

World Class Oil Shear Clutch / Brake Technology
Som Pac® Series Clutch / Brake Drives

Som Pac® drives are completely assembled clutch/brake drive systems operating in a bath of oil within a rugged, sealed housing which is impervious to outside contaminants. These oil shear clutch / brake drives contain multiple plate disc packs that are immersed in oil. Torque is transmitted by the shearing of the oil across the disc providing cooling and lubrication to the disc surfaces. The result is no wear on the disc surfaces, superior heat transfer and long trouble free performance.

Som Pac® Features Include:

- Fully Enclosed Clutch/Brake Drive System
- Continuous Oil Flow
- Clutch/Brake Mechanically Interlocked
- Air Engaged Clutch
- Spring Set Brake (In Standard Units)
- Brake Torque Can Be Adjusted Externally
- Easy Installation
- Reduced Maintenance Costs
Som Pac® 1 Series Clutch/Brake Drive Systems

Som Pac® Applications Include:
- Conveyors
- Transfers
- Shuttles
- Machine Tools
- Index Tables
- Assembly Machines
- Turnovers
- Palletizing Machines
- Welding Machinery
- Electric Motor Manufacturing
- Coil Feeding Equipment
- Spinning Machinery
- Packaging Machinery
- Presses
- Grinding Machines
- Winding Equipment
- Cement Block Machines
- Testing Equipment
- Container & Drum Manufacturing

Som Pac® Model 2402 3/4 & 1202

Som Pac® Model 1203 & 2003 on Test Stand

Som Pac® Model 2410

Som Pac® Model 1202 & 2410

Som Pac® Model 1202

Som Pac® Model 1202 FF Units Ready for Shipment
Operation

Som Pac® 1 drives are completely assembled clutch/brake systems, operating in a bath of oil within a rugged, sealed housing.

Torque is transmitted by the viscous shear of an oil film maintained between the friction surfaces of adjacent discs. In start-stop applications, this oil film is maintained until the last 10% from synchronous speed. The result is that wear of friction surfaces is virtually eliminated.

The kinetic energy absorbed by the drive when starting and stopping machinery is transferred to the oil contained in the housing. Two methods are used to cool this oil. A shrouded fan is used to force air across the housing. If more thermal capacity is required, the unit is water cooled either by a manifold plate mounted under the cover plate of the housing or copper tubing mounted inside the housing.

A positive flow of oil is pumped thru the discs by vanes on the hub whenever the output shaft is rotating. Even though pumping of the oil by the output shaft limits the slippage that can be tolerated, these drives have been proven extremely durable in most application since 1968.

The drive is shown with the clutch engaged. Actuation pressure in the cavity to the left of the piston forces the piston to the right, clamping the clutch disc stack between the clutch actuator and a center ring. The piston transmits the clamping force to the clutch actuator through the ball bearing, the brake actuator assembly, and the screws and rods of the hub of the output shaft. The piston is mounted within drive housing and does not rotate.

When air is exhausted from the left side of the piston, the brake springs force the piston to the left and clamp the brake disc stack through the ball bearing and the brake actuator. Since the clutch actuator is solidly connected to the brake actuator assembly by the screws and spacer rods, motion to the left automatically releases the clutch and engages the brake at the same time. The torque capacity of the brake can be changed externally by using either 3, 4 or 6 springs. The springs are accessible for removal or installation by removing plugs at the output end of the housing.

Alternatively, air can be used to actuate the brake instead of springs.

When a model with an “L” suffix is specified, 3 light springs are used for mechanical reasons. When it is necessary to position the spindle for loading of the work piece on machine tool applications, “N” (neutral position) is specified as a suffix.
Example

Motor – 20HP – 1750 RPM
WR2 of Load = 600 lbs.-ft.2
WR2 of Couplings = 300 lbs.-in.2
Shaft #1 = 1-3/4 dia x 15” lg.
Shaft #2 = 3-1/2 dia x 26” lg.
Cycle Rate = 6 per minute
Air Pressure Available - 60 PSI
Acceleration/Deceleration Time = .75 seconds

Calculations

Step #1

\[ T = \frac{HP \times 63,000 \times 2.75}{1750} \]  
(Formula 1)

Step #1

The inertia of coupling #1 can be ignored since it is on input side of drive.

WR2 of coupling #2 = 130/144 = .903 lbs.-ft.2

Shaft #1

WR2/in of dia from chart = .0018
WR2 = .0018 x 15 = .027 lbs.-ft.2

Pinion

(Use pitch dia of gears for cylinder outside diameter.)

WR2 = \( \frac{.281 \times 4.375 \times (4.5^2 - 1.75^2) \times (4.5^2 + 1.75^2)}{1466.8} \)
= .336 lbs.-ft.2

Total Inertia at Drive Speed

Coupling = .903
Pinion = .336
Drive (preliminary est. is 1207) = .735

Shaft #2

WR2 = \( \frac{.281 \times 26 \times 3.5^2 \times 3.5^2}{1466.8} \)
= .747 lbs.-ft.2

Gear

RIM: WR2 = \( \frac{.281 \times 4.25 \times (30^2 - 27.5^2) \times (30^2 + 27.5^2)}{1466.8} \)
= 193.85 lbs.-ft.2

WEB: WR2 = \( \frac{.281 \times 1.25 \times (27.5^2 - 6^2) \times (27.5^2 + 6^2)}{1466.8} \)
= 136.64 lbs.-ft.

HUB: WR2 = \( \frac{.281 \times 6 \times (6^2 - 3.5^2) \times (6^2 + 3.5^2)}{1466.8} \)
= 1.317 lbs.-ft.2

Total Inertia at Shaft #2

Shaft = .747
Gear = 193.850
136.640
1.317
Load = 600.000
932.550 lbs.-ft.2

Equivalent Inertia Drive Speed

Speed of RPM = 1750 \times \frac{18}{120} = 262.5
WR2 = \( 932.55 \times \left( \frac{262.5}{1750} \right)^2 \)  
(Formula 3)
= 20.98 lbs.-ft.2

Total Inertia of Machine, Drive, Coupling & Load Referred to Drive

WR2 = 1.974 + 20.98 = 22.96 lbs.-ft.2

Step #3

\[ T = \frac{22.96 \times 1750}{17.9 \times .75} = 2993 \text{ in.-lbers.} \]  
(Formula 5)

Step #4

Heat = \( \frac{3.4 \times 22.96 \times \left( \frac{1750}{100} \right)^2}{33,000 \times 4.33 \text{ THP}} \times 6 \)  
(Formula 6)

Calculate Static Clutch Torque at 60 PSI

This torque is more than calculated in Step #1 so the 1207-6 would be the proper selection for driving load but is less than step #5. Also, the torque in step #5 is larger than step #1. The designer must either use a larger drive and motor or increase the acceleration time. (Deceleration time is O.K.) The designer now checks what the acceleration time would be using the torque calculated in step 1.

\[ T = \frac{22.96 \times 1750}{17.9 \times 1980} = 1.134 \text{ seconds} \]

After reviewing total machine performance lets assume this acceleration time will be O.K. The final selection is thus, 2407-6
**Som Pac® 1 Series Clutch/Brake Drive Systems**

**Drive Selection**

The selection of the correct Som Pac® is determined by the analysis of the following factors.

1. Static Torque required to drive machine.
2. Acceleration/Deceleration Torque required to start and stop machine. This requires the following machine
   a. Acceleration Time
   b. Deceleration Time
   c. Inertia of Machine and Drive
   d. Speed of Drive
3. Heat Load imposed on the drive by starting and stopping machine. This is determined by the following.
   a. Factors 2c and 2d above.
   b. Number of Cycles Per Minute.

**Som Pac® Selection Guide**

<table>
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<th>MOTOR HP</th>
<th>DRIVE RPM</th>
<th>MODEL NUMBER</th>
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**Static Torque**

Som Pac® drives are generally driven by a squirrel cage motor. It is assumed that the machine designer has correctly determined the correct horsepower motor to drive the machine. Static torque calculation of clutch is then based on the motor horsepower.

\[
T = \frac{HP \times 63,000 \times 2.75}{N}
\]  
(Formula 1)

Where:

- \(T\) = Static Clutch
- \(HP\) = Motor Horsepower
- \(N\) = RPM of Drive (not the same as motor RPM if reduction is used between motor and drive)

Note: If a flywheel is used on the input side of the drive, the above formula is NOT valid. Torque must be determined by actual torque required by machine.

(Example: Stamping Presses)

The brake is usually sized based on dynamic torque requirements. In those cases that require the brake to hold a platen or other component in the vertical plane, the drive selected should have a brake torque rating of 120% of the actual calculated torque.
Machine Inertia

It is necessary to calculate the inertia of the machine before acceleration and deceleration torques can be calculated. Heat load calculations also require this information.

The rotating elements of the machine, no matter how complex, can usually be defined by a series of solid or hollow cylinders. Inertia of a machine component is usually expressed as lbs.-ft.² (WR²).

The formula for calculating the inertia of solid or hollow cylinders is:

\[ WR² = \frac{WL(D^2 - d^2)(D^2 + d^2)}{1466.8} \]  
(Formula 2)

Where:

\[ WR² = \text{Inertia} - \text{lbs.-ft.²} \]
\[ W = \text{Weight per cubic inch of material in pounds.} \]

Steel = .281 lbs.  Aluminum = .093 lbs.

D = Outside diameter of cylinder in Inches

d = Inside diameter of cylinder in Inches

\[ d = 0 \] in the case of solid cylinder.

L = Length or thickness of cylinder in inches.

**WR² Per Inch (Lbs.-Ft.²):**

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Chart shows WR² for steel. For other materials multiply above values by the following:

- Cast Iron: .92
- Aluminum: .33
- Brass: 1.09

For hollow cylinders subtract the inside diameter WR² from the outside diameter WR².

The typical machine has several different shafts, each with its gears, sprockets, etc., and rotating at different speeds. The WR² of all the components on each shaft are added to keep separate from the components on the other shafts. The WR² of the components on the shafts that DO NOT rotate at the same speed as the drive must be translated to an equivalent WR² at drive speed.

\[ \text{EQ. WR²} = \frac{WR²}{N_1^2} \]  
(Formula 3)

Where:

\[ \text{EQ. WR²} = \text{WR² mounted on shaft of drive that would have the same effect as the WR² on the different speed shaft.} \]
\[ WR² = \text{Total WR² on shaft that rotates at speed different from drive.} \]
\[ N_1 = \text{RPM of subject WR²} \]
\[ N_2 = \text{RPM of drive.} \]

Some machines have assemblies that travel linearly such as transfer tables or shuttles and platens. The equivalent WR² is calculated as follows.

\[ \text{EQ WR²} = \frac{WV^2}{39.5 N_2^2} \]  
(Formula 4)

Where:

\[ W = \text{Weight of assembly-insert any load (lbs.)} \]
\[ V = \text{Maximum linear velocity of assembly (ft./min.)} \]
\[ N = \text{RPM of drive.} \]

Add the equivalent WR² of the components at the different speeds. This total is the WR² used in subsequent calculations. DO NOT FORGET the flexible coupling that connects the drive to the machine nor the “Start/Stop WR²” of the drive that is shown on the specification charts. The WR² of commercial couplings is usually given in lbs.-In.². To convert this to lbs.-ft.² divide by 144.
**Acceleration/Deceleration Torque**

\[ T = \frac{WR^2 N}{17.9t} \]  

(Formula 5)

Where:

- \( T \) = Required equivalent Static Torque (in.-lbs.) of clutch or brake in drive. (Torque values listed in specification charts are “Static Torque”).
- \( N \) = RPM of drive
- \( t \) = Acceleration/Deceleration Time (seconds)

Note: Calculate clutch and brake torques separately if starting and stopping times are different.

**Heat Load**

The heat load or thermal capability of the drives is given in the specification charts as Thermal Horsepower (THP). The required THP for the drive is calculated as follows.

\[ \text{THP} = \frac{3.4 \cdot (WR^2) \cdot \left( \frac{N}{100} \right)^2 \cdot C}{33,000} \]  

(Formula 6)

Where:

- \( WR^2 \) = Total inertia of machine, coupling and drive (lbs.-ft.²)
- \( N \) = RPM of drive
- \( C \) = Cycles/Min.
- \( 3.4 \) = Constant composed of content 1.7 which is derived from basic formula for rotary motion

K.E. (ft.-lbs.) at 100 RPM = 1.7 \((WR^2)\) and constant 2 which takes into consideration that each cycle includes one clutch and one brake operation

\( 33,000 \) = Energy (ft.-lbs./Min.) generated by 1 horsepower

**Static Torque Rating at Any Air Pressure**

The static clutch torque of Som Pac® are rated at 80 psi. Using the following formula to determine clutch torque rating at a different air pressure.

\[ T_1 = \left( \frac{P_1 - P_2}{80 - P_2} \right) T_2 \]  

(Formula 7)

Where:

- \( T_1 \) = Clutch static torque at operating air pressure of \( P_1 \)
- \( T_2 \) = Rated clutch torque at 80 PSI
- \( P_1 \) = Operating air pressure (PSI)
- \( P_2 \) = Clutch engaged PSI: (see specification charts)

The static brake torque of all drives with spring set brakes is modified by changing the number of brake springs (with the exception of the 2402 3/4). Som Pac® I and III are designed so the number of brake springs can be changed externally. Som Pac® II can be ordered with either an A or B spring.
Final Selection of Drive

The Som Pac® drive give the engineer an almost unlimited flexibility in specifying a drive that best suits his operating conditions.

The specific steps required in order to make the final drive selection are as follows.

1. Calculate static torque required. (Formula 1)

2. Calculate machine inertia (Formula 2, 3 & 4). Don’t forget coupling between machine and drive nor inertia of drive.

3. Calculate equivalent static torque required for starting and stopping machine (Formula 5). This step can be skipped if actual acceleration/deceleration time is not of specific importance.

4. Calculate heat load. (Formula 6)

5. Make preliminary selection of drive size in either Som Pac® I, II or III on steps 1, 2, 3 & 4. Rating of drive selected must be equal or greater than calculated values. Specify Som Pac® III if accel/decel time of more than 1 second was used in Formula 5 (step 3). If acceleration/deceleration rate is to be programmed, select a Som Pac® III and specify use of Valve 2401.

6. Determine what maximum air pressure is available at plant location where drive will be installed. If air pressure is less than 80 PSI, calculate Formula 7 for preliminary drive selection made in step 6.


8. If step 3 was skipped, check for actual acceleration 6


10. If step 3 was skipped, check for actual acceleration and deceleration time for drive selected in step 5 or step 7 by calculating following formula which is Formula 5 rewritten.

\[ t = \frac{W R^2 N}{17.9T} \]  \hspace{1cm} \text{(Formula 8)}

Note: The torque value of T here is either rated torque or torque calculated in step 6.

If T is more than 1 second for either clutch or brake, Som Pac® III or a larger Som Pac® I must be used. Consult factory for possible modifications if time is more than 1 second.

Machine Started Under Load

The steps given above are only valid if the machine is stated under a “no load” condition. The typical machine to which these drives are applied are started in this condition, e.g., machine tools, automation equipment and presses.

However, if the machine must perform work during the acceleration of the machine to full speed, this must be considered when calculating starting torque (step 3) and heat load (step 4). Examples of this type of machine are certain pump applications, bulk conveyors, fans and grinding mills. Unless the reader knows how to make these calculations, he is urged to refer his application to the factory since each type of machine requires its own type of analysis.
## Som Pac® 1 Series Clutch/Brake Drive Systems

### Specifications

<table>
<thead>
<tr>
<th>MODEL</th>
<th>NO. BRAKE SPRINGS</th>
<th>STATIC CLUTCH TORQUE (IN.-LBS.)</th>
<th>STATIC BRAKE TORQUE (IN.-LBS.)</th>
<th>CLUTCH ENGAGE PSI</th>
<th>START/STOP W/P (LBS.-FT.)</th>
<th>HP/Thermal Capacity</th>
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### Notes:
1. Dynamic pickup torque is 70% of Static Torque.
2. Models with "L" number of springs are used for air set brakes. Specify "N" (neutral position) when output shaft is to be "free-to-turn" after brake air is exhausted.
3. Use Som Pac® 3 when acceleration-deceleration control is required or if these times exceed one second.

### Dimensions

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