LOAD PROPORTIONAL SAFETY BRAKE

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INTRODUCTION

This brake is a self-energizing mechanical friction brake and is intended for use in a rotary drive system. It incorporates a torque sensor which cuts power to the power unit on any overload condition. The brake is capable of driving against an opposing load or driving, paying-out, an aiding load in either direction of rotation. The brake also acts as a no-back device when torque is applied to the output shaft.

The advantages of using this type of device are:

1. Low frictional drag when driving.

2. Smooth paying-out of an aiding load with no runaway danger.

3. Energy absorption proportional to load.

4. No-back activates within a few degrees of output shaft rotation and resets automatically.

5. Built-in overload protection.

DESCRIPTION

Figure 1 shows a section view of the brake which is made up of two major components, the braking component and the torque sensing component. The braking component is comprised of three separate elements; the energizing brake, the load sensing cams, and the holding brake. The torque sensor is comprised of load sensing detents, load reacting springs and a switch assembly. The brake absorbs energy in proportion to the load and dissipates the energy as heat through an oil filled, finned aluminum housing.

Figure 2 shows the brake assembly without the torque sensing element attached. This type of brake can be used in applications which do not require overload protection.

The input shaft (1) has one outer and one inner set of driving slots, see Figures 1 and 3. The inner set of slots contact lugs on the main brake shaft (2) and provide a straight through drive. They also provide a secondary load path in case of brake failure. The outer driving slots contact lugs on the spring cage (6) which is index splined to the input cam plate (10). This arrangement of driving slots and lugs maintains alignment of the ball detent position on the cam plates (10) (12) and keeps the holding brake de-energized. Because of the symmetry of these parts, the brake will

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function when driven in either direction. Return springs (33) provide an axial force to the output cam plate (12) which keeps the three balls (11) seated in their detent position and maintains a gap at the brake plates to eliminate holding brake drag. The energizing springs (7) which are retained in the spring cage (6) provide an axial force to the energizing brake which generates a reaction torque for energizing the holding brake. The reaction torque is transmitted by the stator plates (9) which are grounded to the external housing (3) through spline teeth on the stator (37). Anti-friction thrust bearings (5) (35) are used to carry the force of the energizing springs (7) to minimize frictional drag developed by the relative rotation of the input cam plate (10) and main brake shaft (2) during brake application. Nut (4) is used to react the energizing spring force and to adjust the gap at the energizing brake plates. The load sensing element of the brake is made up of circular ball cams on both the input and output cam plates (10) (12), three balls (11), and a ball cage (36), see Figure 4 and 5. The ball cams are designed to operate in either direction. The cam slope forms a vee which has a radius smaller than the ball radius. This provides a detent effect which prevents movement until the transmitted torque actuates the cam plates. Transverse to the direction of rotation the cam radius is larger than the ball radius. This allows the balls to seek a position where all the balls share the load, and compensates for tolerances on the individual parts. The ball cage (36) is used to capture the three balls and forces the balls to move in unison up the cam surfaces. The brake plates (8) (13) are a special design which have a controlled uniform coefficient of friction. The friction lining is non-metallic and is bonded to a steel core which is splined to the cam plates. The stator plates (9) (14) have an external spline with every other tooth removed, see Figure 3, and provide the mating surface for the brake plates (8) (13). The stator (37) also has every other tooth removed on the internal spline in the area of the holding brake only. Spacers (15) are provided in both the energizing and holding brake to change the braking capacity depending on the load. The load springs (34) are retained in spring cage (16). They supply an axial force to the holding brake in direct proportion to the load. The springs are selected so that the force generated by the load sensing cams is within their working range. This enhances the smoothness of the brake when driving an aiding load. Shim (32) is a laminated shim used to adjust the gap on the holding brake. Nut (18) is used to lock the pressure plate (17) to the main brake shaft (2) for reacting the load spring force. The torque sensor has an input and output detent shaft which retain three balls (20) in countersunk detents. The input detent shaft (19) is splined to the main brake shaft (2) and transfers load to the output detent shaft (31) through the three balls (20). In case of ball disintegration, a secondary load path is provided through a pair of driving lugs with enlarged clearances to allow normal operation. The output detent shaft (31) is splined to the output shaft (26) and has a flange which, when displaced, contacts an anti-friction bearing (22) on the switch assembly. Five belleville springs (30) provide the force to the output detent shaft for retaining three balls (20) in the detents. The spring force is reacted to the input detent shaft through washers (28), an anti-friction thrust bearing (29), and nut (27). A lever (24) and shaft (23) are used to transfer the detent shaft motion to actuate the switch (21). An adjustable stop (25) is provided for the lever (24) to maintain a clearance gap at anti-friction bearing (22) during normal operation.

PRINCIPLES OF OPERATION

Driving Against A Load (Figure 6)

When driving from a stopped position with the brake fully energized, (no-back activated) the initial rotation of the input shaft (1) releases the holding brake. The input shaft (1) supplies torque directly to the main brake shaft (2) through a pair of driving lugs, see Figure 3. The output cam plate (12) is driven by the main brake shaft (2) to align the ball detents. The holding brake allows the cam plate (12) to rotate because of the enlarged clearance between the stator plate (14) and the stator (37) spline teeth. The input shaft (1) simultaneously drives the spring cage (6) through another pair of driving lugs which provide torque to the input cam plate (10) to overcome the energizing brake torque. The cam plates are keyed so that the ball detents are aligned when the driving faces of both sets of lugs are in contact. With the ball detents aligned the return springs (33) provide an axial force to the output cam plate (12) which keeps the three balls(11) seated in the detents. Since the three balls (11) cannot rise up the cam surfaces the brake remains de-energized. Torque is then applied through a spline to the input detent shaft (19) of the torque sensor and transferred through three balls (20) to the detent shaft (31). The output detent shaft (31) is coupled to the output shaft (26) through a spline and transmits the torque to the drive system. When an overload condition is developed the load reacting springs (30) are not supplying enough force to keep the three balls (20) seated in the detents. The output detent shaft (31) is then displaced axially to operate a lever (24) which trips the switch (21) in the electrical control circuit and cuts power to the drive motor.

Stopped-No-Back (Figure 7)

When the drive motor is stopped, the load back drives the main brake shaft (2) and the output cam plate (12). The energizing springs (7) maintain a force on the energizing brake plates which provide a reaction torque to the load through the three balls (11). As torque is applied, relative rotation occurs between the input cam plate (10) and the output cam plate (12). This causes the output cam plate (12) to displace axially as the three balls (11) rise up the cam surfaces and remove the holding brake plate clearance. Further increase in back driving torque displaces the output cam plate (12) to compress the load springs (34) to some force level. When the spring force applied to the holding brake plates (13) (14) generates sufficient torque all movement stops and the load is fully reacted.

Driving An Aiding Load - Paying Out (Figure 8)

When driving an aiding load, the input shaft (1) rotates in the direction of the load and the outer set of driving lugs contact the spring cage (6) which applies torque to the input cam plate (10), see Figure 3. When this torque exceeds the difference between the energizing brake torque and the reaction torque of the three balls (11), the main brake shaft (2) rotates under the influence of the load. The load springs (34) modulate the force of the load sensing cams on the holding brake plates (13) (14) which generate a holding torque permitting the load to pay out smoothly at drive motor speed.

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T_1	=	Input Torque	T_o	=	Output Torque
T_D	=	Drag Torque	T_M	=	Main Brake Torque
T_B	=	Torque Across Ball	F_E	=	Energizing Spring Force
F_M	=	Modulating Spring Force	F_R	=	Return Spring Force
F_A	=	Axial Force At Ball	F_t	=	Tangential Force At Ball
μ	=	Coefficient Of Friction	r_D	=	Drag Brake Plate Radius
r_M	=	Main Brake Plate Radius	r_B	=	Radius To Ball
n_D	=	Number Of Drag Brake Surfaces	n_M	=	Number Of Main Brake Surfaces
$\emptyset$	=	Cam Slope Angle			

MAXIMUM CAM SLOPE ANGLE

For a given load the axial force on the ball is: (See Figure 7)

$$T_o = F_A(\mu r_M n_M + \text{Tan}\emptyset r_B) - F_R \mu r_M n_M$$

$$F_A = \frac{T_o + F_R \mu r_M n_M}{\mu r_M n_M + \text{Tan}\emptyset r_B}$$

Minimum Holding Brake is:

$$T_D = T_B$$

$$F_A \mu r_D n_D = F_A \text{tan}\emptyset r_B$$

Substituting for F_A:

$$\frac{T_o + F_R \mu r_M n_M}{\mu r_M n_M + \text{tan}\emptyset r_B}(\mu r_D n_D) = \frac{T_o + F_R \mu r_M n_M}{\mu r_M n_M + \text{tan}\emptyset r_B}(\text{tan}\emptyset r_B)$$

Solution of the above equation resolves into a quadratic of the form:

$$A \text{tan}\emptyset^2 + B \text{tan}\emptyset - C = 0$$

Substituting values for all the known terms and solving for $\emptyset$ will give the maximum cam slope angle which will hold the load without slipping.

BRAKE STABILITY

The brake as described in Figures 1 and 2 is easily proportioned to be self-energizing and capable of holding a load in the static or stopped

condition. The critical condition is driving or paying out an aiding load smoothly without brake chatter. Chatter is an unstable condition in which the brake is rapidly cycling between locked-up and released.

The following items are required to insure stable operation of the brake:

1. Friction material with a stable coefficient of friction throughout the operating range of temperature and brake plate pressure.

2. Combination of friction material and oil which minimizes the fluctuation between static and dynamic coefficient of friction.

3. Correct load balance between the energizing brake and the main holding brake.

4. Modulating spring rate to balance maximum axial load developed by the ball cams.

Another method of producing a stable brake is to reduce the torque capacity of the energizing brake to a value below the ball torque and then add a constant drag brake to the input shaft. The affect of reducing the energizing brake is to make the brake reversible, but the addition of the constant drag brake in conjunction with the other braking elements makes the brake irreversible and capable of reacting the full load. When driving an aiding load, the input torque need only overcome the margin of the constant drag brake which allows the load to pay-out. Figure 9 shows a constant drag brake applied to the input shaft with a ratcheting device which bypasses the brake when driving against a load.

Figure 10 shows the drive system of the Boeing Model 747 On-Board Loader which used two brakes in a dual load path arrangement. The final design of the brake as delivered was as shown in Figure 1.

<center>ANALYSIS OF ON-BOARD LOADER BRAKE</center>

Weight of platform and payload - 34,000 lb.
Torque at cable drum - 102,000 in-lb.
Torque at brake - 1,090 in-lb.

Brake Design Parameters:

r_D = 2.47 in.	F_E = 60 lb.
r_M = 2.47 in.	F_R = 30 lb.
r_B = 2.50 in.	F_M = 552 lb.
n_D = 4	μ = .12-.10 oper.
n_M = 6	μ = .06 Min.

$$\frac{T_o + F_R \mu r_M n_M}{\mu r_M n_M + \tan\phi r_B} \, (\mu r_D n_D) \;=\; \frac{T_o + F_R \mu r_M n_M}{\mu r_M n_M + \tan\phi r_B} \, (\tan\phi r_B)$$

Substituting values for all known terms and the minimum coefficient of friction, the above equation resolves into the following quadratic:

$$172.975 \tan \phi^2 + 20.5079\, \phi - 14.5885 = 0$$

Solution of the quadratic gives the maximum cam slope angle which will allow the brake to support the load without slipping.

Solution: $\phi = 13.34^\circ$

To allow for tolerances we will use $\phi = 12.50^\circ$

DRIVING AN OPPOSING LOAD (See Figure 6)

Input Torque:

$$T_1 = F_E \mu r_D n_D + T_o$$

$$= 60\,(.11)\,2.47\,(4) + 1090$$

$$T_1 = 1155 \text{ in-lb}$$

STOPPED WITH LOAD APPLIED (See Figure 7)

Axial Force:

$$F_A = \frac{T_o + F_r \mu r_M n_M}{\mu r_M n_M + \tan\phi r_B}$$

$$= \frac{1090 + 30\,(.11)\,2.47\,(6)}{.11\,(2.47)\,6 + \tan 12.50^\circ\,(2.5)}$$

$$F_A = 521.4 \text{ lb}$$

For brake to be irreversible:

$$T_D > T_B$$

$$T_D = F_A \mu r_D n_D$$

$$= 521.4\,(.11)\,2.47\,(4)$$

$$T_D = 566.7 \text{ in-lb}$$

$$T_B = F_t r_B$$

$$= F_A \tan\phi r_B$$

$$= 521.4 \tan 12.50° (2.5)$$

$$T_B = 288.9 \text{ in-lb}$$

$$T_D > T_B$$

$$566.7 > 288.9 \therefore \text{Brake is irreversible}$$

BRAKE HOLDING MARGIN

$$T_M = F_M \mu r_M n_M$$

$$= (521.4 - 30) .11 (2.47) 6$$

$$T_M = 801.1 \text{ in-lb}$$

Margin:

$$\frac{T_D + T_M}{T_o} = \frac{566.7 + 801.1}{1090}$$

$$\text{Margin} = 1.25$$

DRIVING AN AIDING LOAD (See Figure 8)

Input Torque:

$$T_1 = F_A (\mu r_D n_D - \tan\phi r_B)$$

$$= 521.4 \left[.11 (2.47) 4 - \tan 12.50° (2.5) \right]$$

$$= 566.7 - 288.9$$

$$T_1 = 277.8 \text{ in-lb}$$
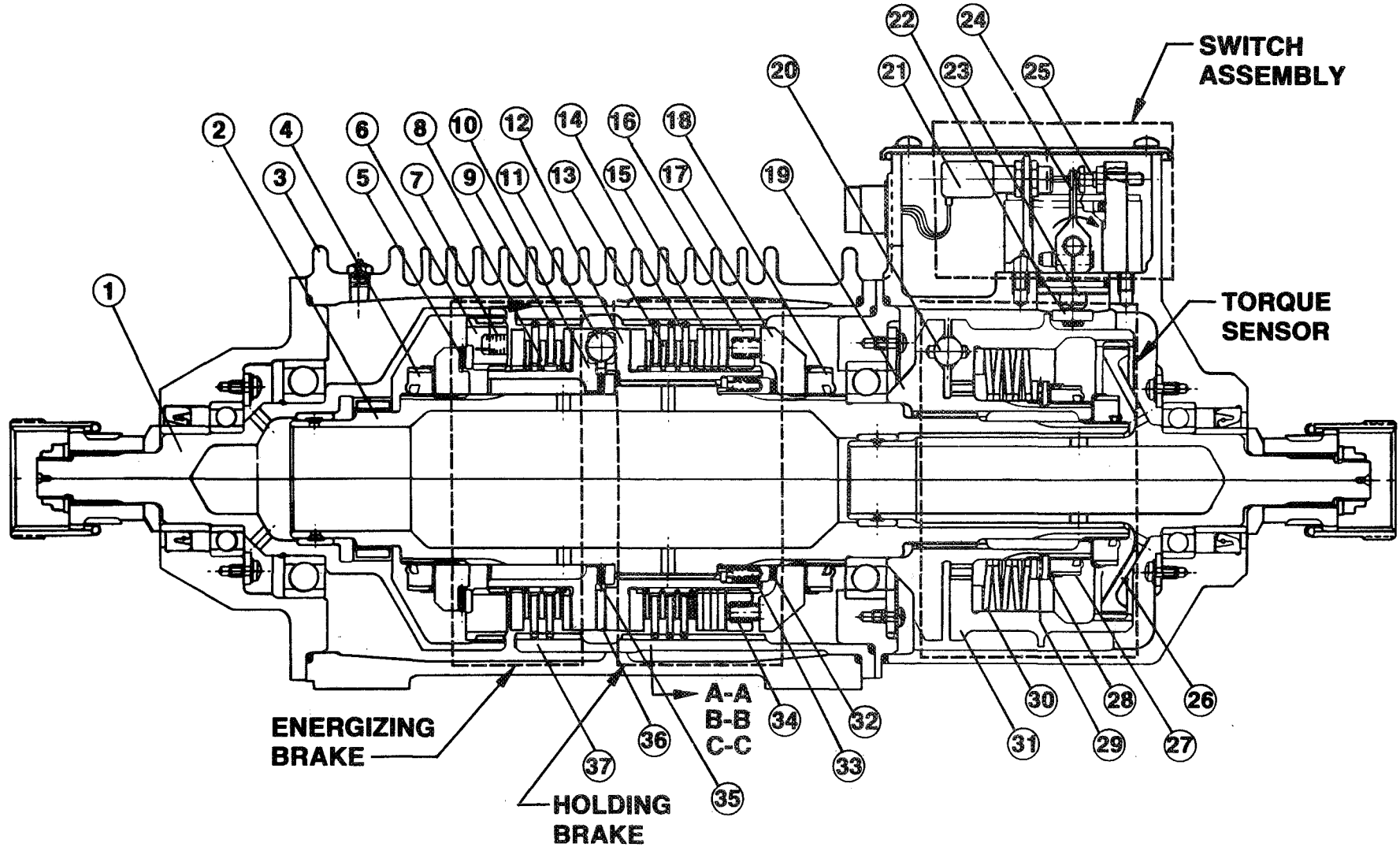
Figure 1

Energy Absorbing Brake With Torque Sensor

INPUT SHAFT

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OUTPUT SHAFT
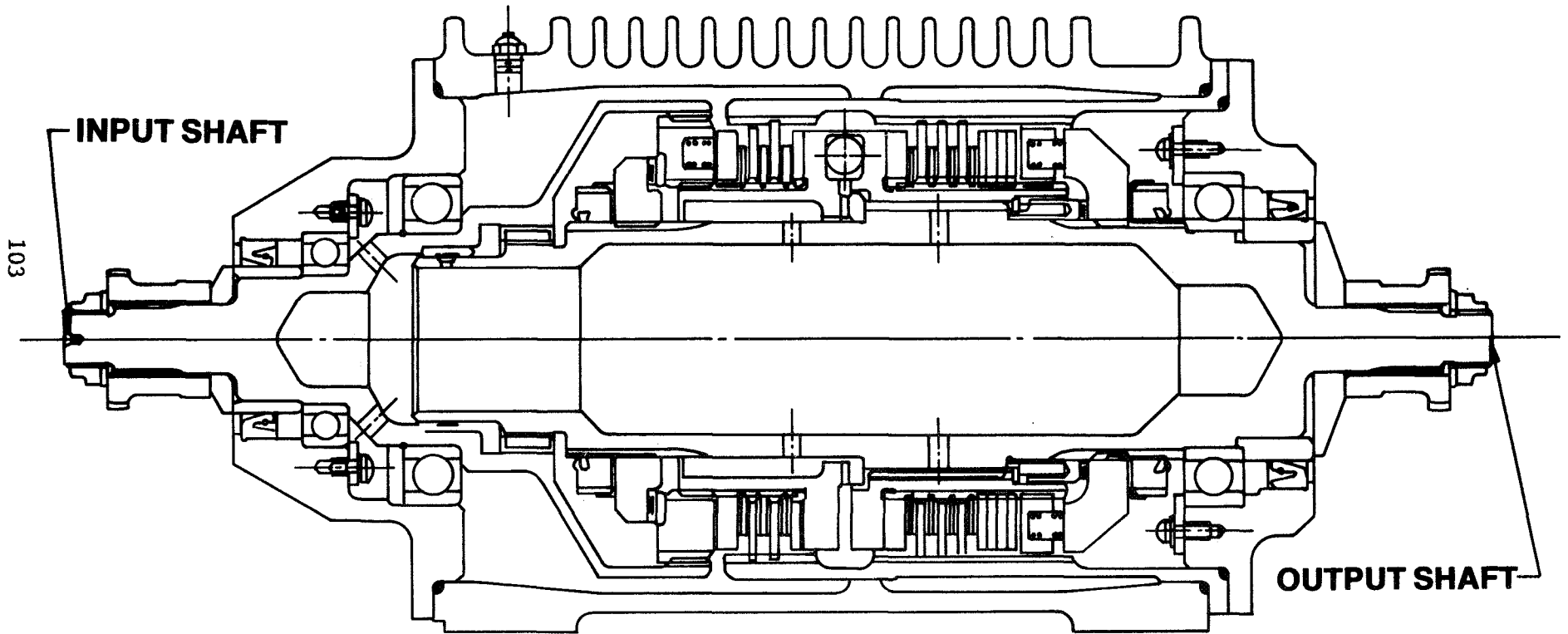
Figure 2

Energy Absorbing Brake

DRIVING LUGS

DRIVING SLOTS

A-A
DRIVING AGAINST LOAD

B-B
DRIVING AIDING LOAD

C-C
STOPPED
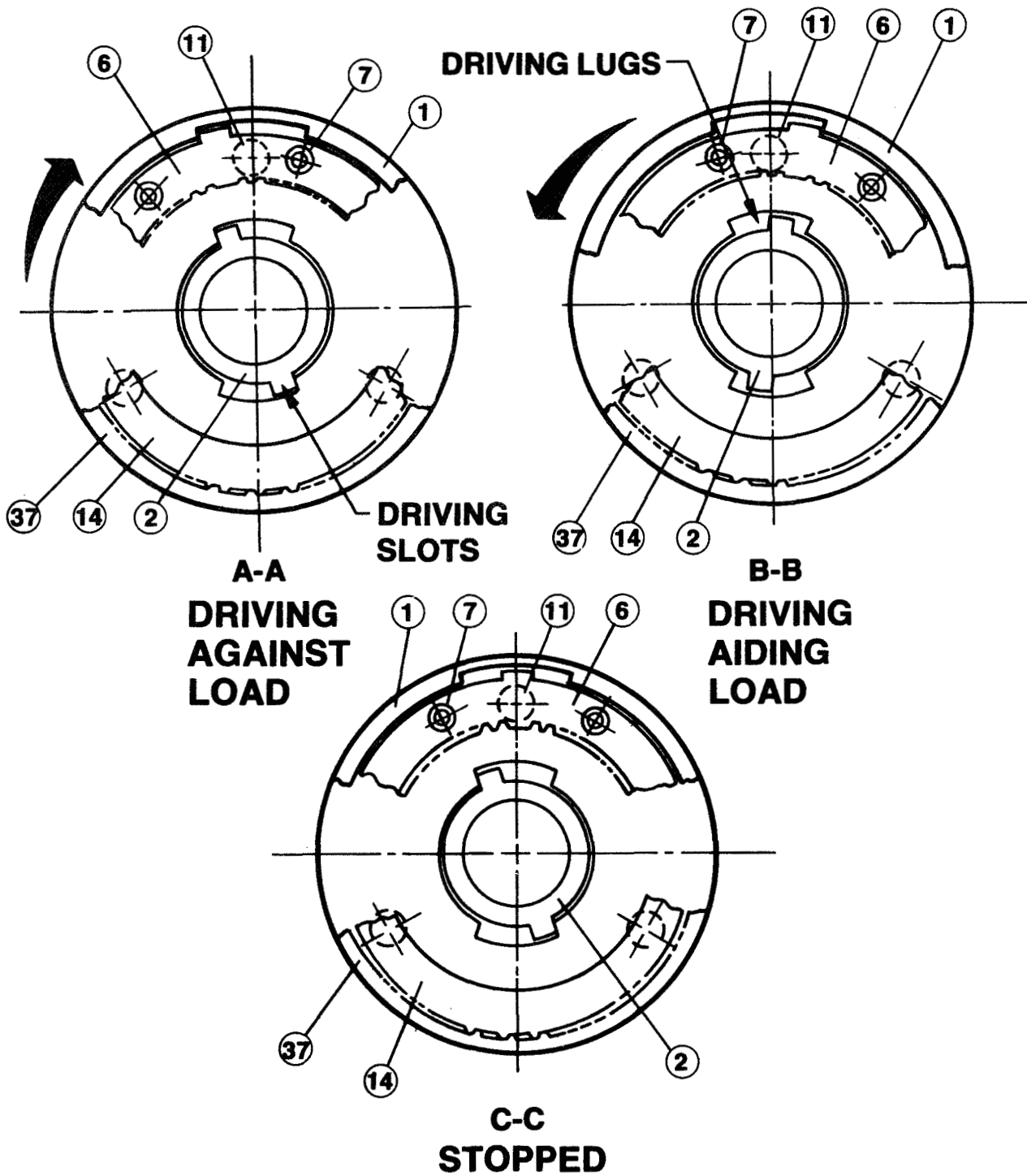
Figure 3
Sections Through Driving Members

RELIEF GROOVE

BALL

CAM GROOVE

$13\% - 11\% R_b$

R_b

ϕ

$98\% - 95\% R_b$

PLAN VIEW

Figure 4

**SECTION A-A
(UNWRAPPED)**

$107\% - 105\% R_b$

SECTION B-B

R_b

CAM GROOVE

RELIEF GROOVE
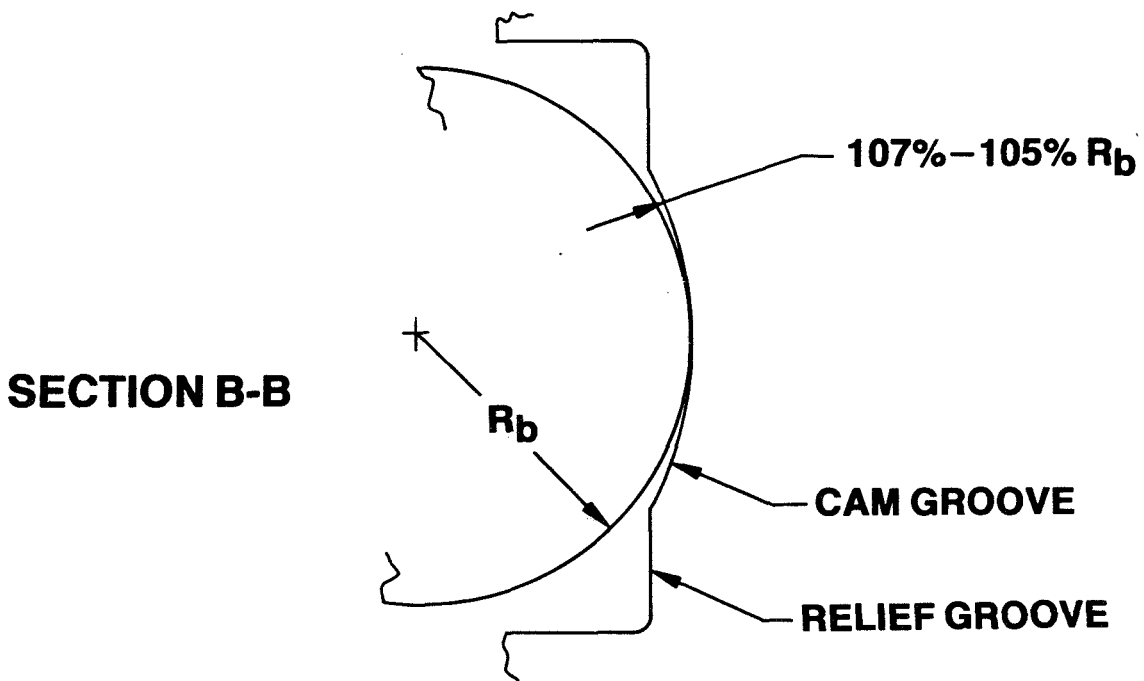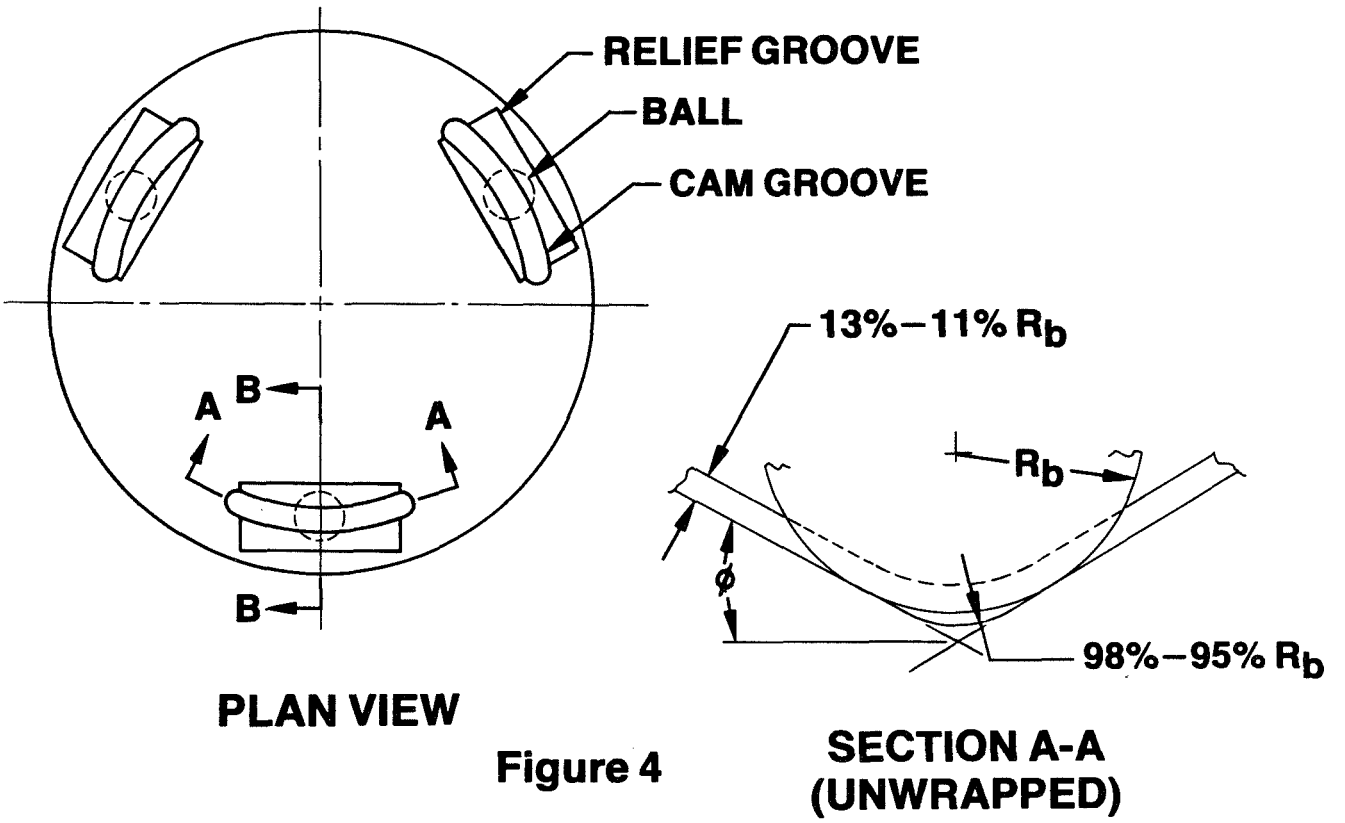
Figure 5

Ball Cam Groove
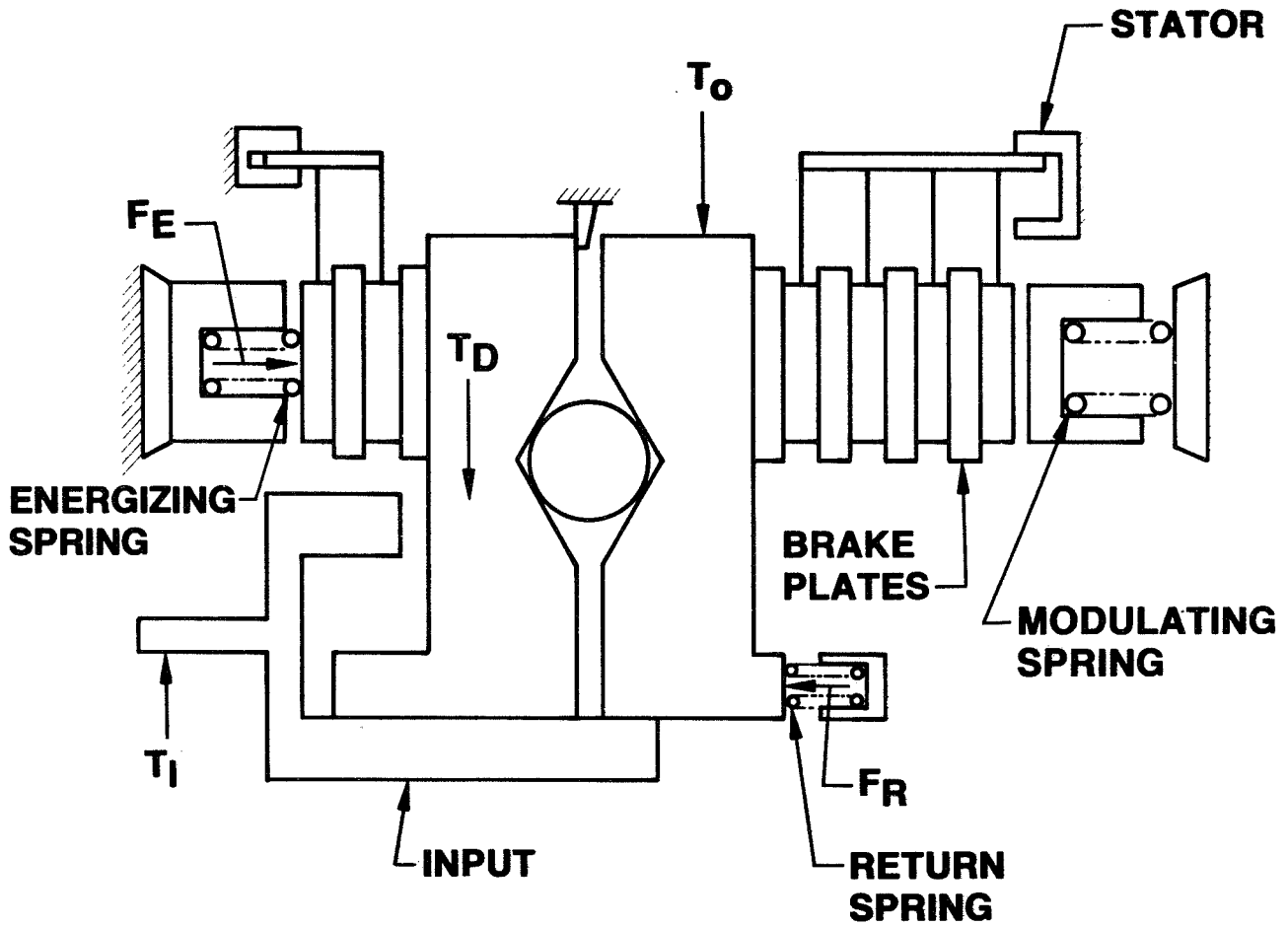
$$T_I = T_D + T_o \qquad\qquad T_D = F_E \mu r_D n_D$$
$$T_I = F_E \mu r_D n_D + T_o$$

Figure 6

Driving an Opposing Load
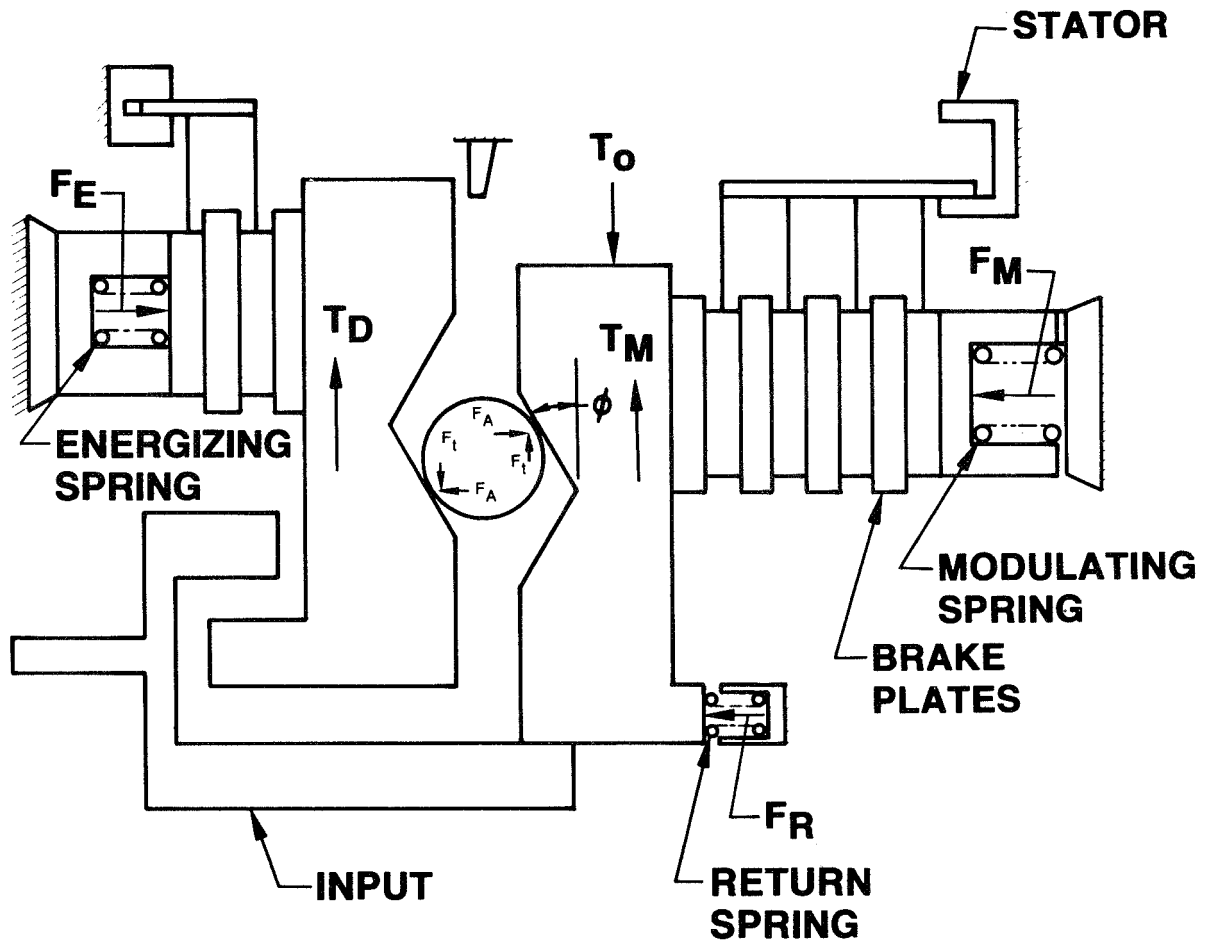
$$T_O = T_M + T_B$$
$$T_O = F_M \mu r_M n_M + F_t r_B$$
$$T_O = (F_A - F_R) \mu r_M n_M + F_A \tan \phi r_B$$
$$T_O = F_A (\mu r_M n_M + \tan \phi r_B) - F_R \mu r_M n_M$$

$$T_M = F_M \mu r_M n_M$$
$$T_B = F_t r_B$$
$$F_A = F_M + F_R$$
$$F_t = F_A \tan \phi$$

FOR BRAKE IRREVERSIBLE:

$$T_D > T_B$$

$$T_D = F_A \mu r_D n_D$$

Figure 7
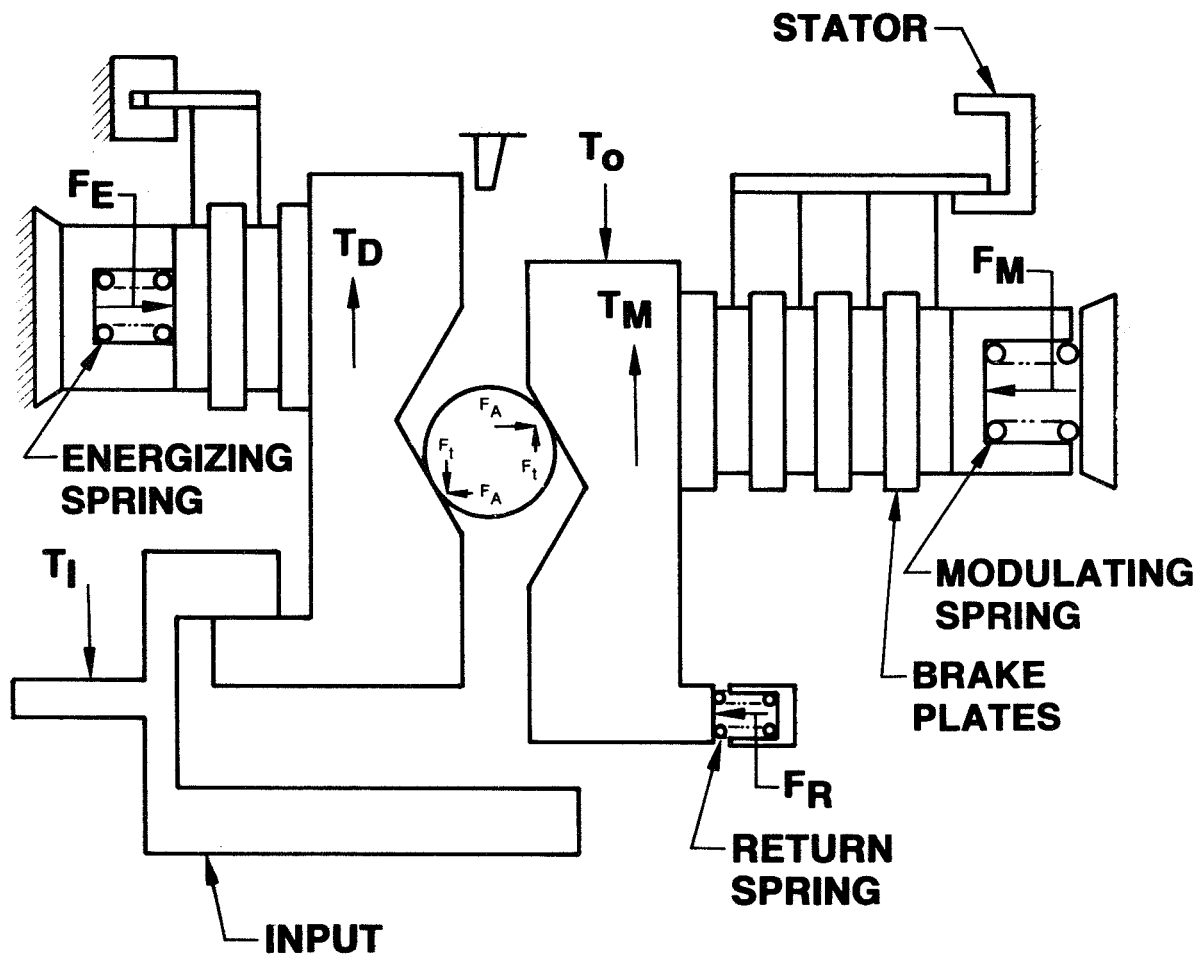
Stopped With Load Applied

$$T_I = T_D - T_B$$
$$T_I = F_A \mu r_D n_D - F_t r_B$$
$$T_I = F_A \mu r_D n_D - F_A \tan\phi \, r_B$$
$$T_I = F_A (\mu r_D n_D - \tan\phi \, r_B)$$

$$T_D = F_A \mu r_D n_D$$
$$T_B = F_t r_B$$
$$F_t = F_A \tan\phi$$

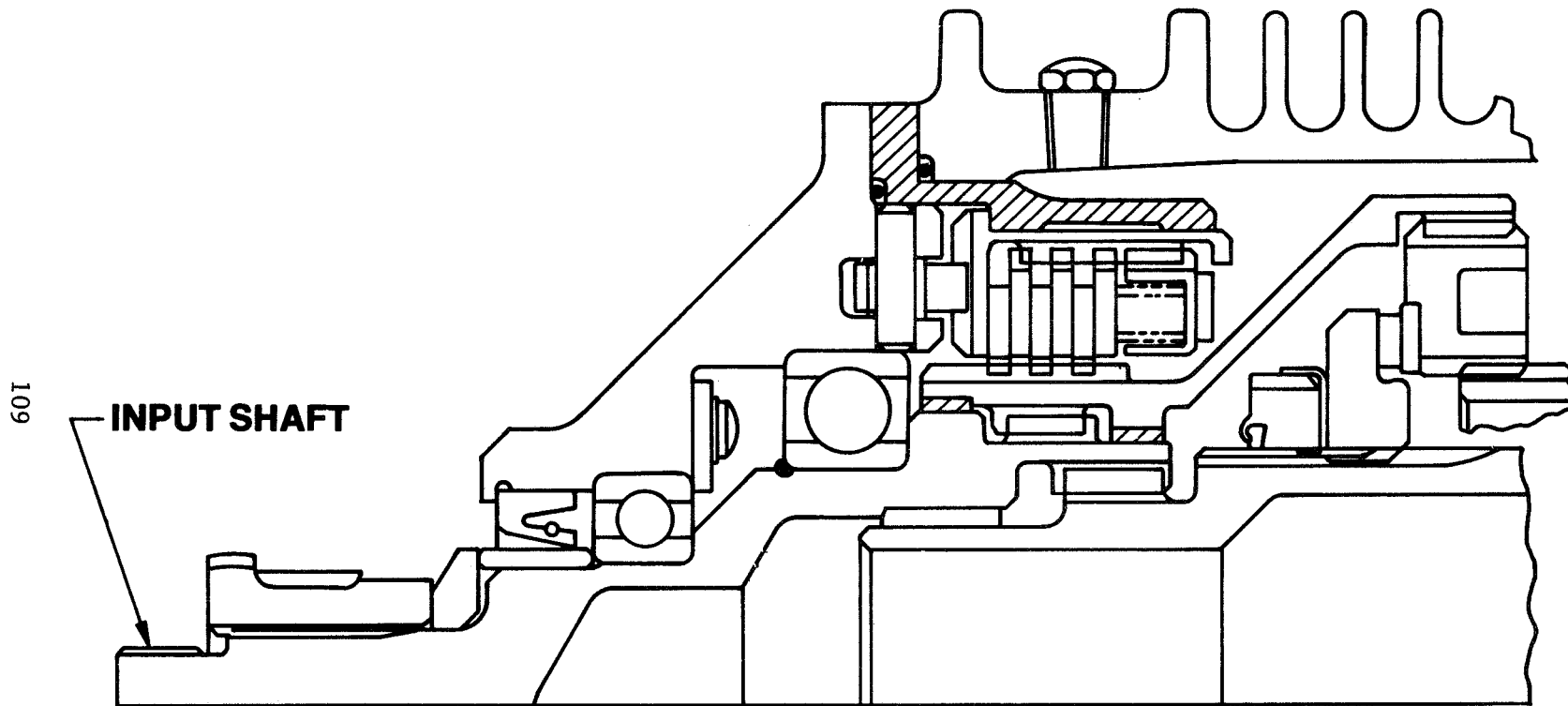## Figure 8
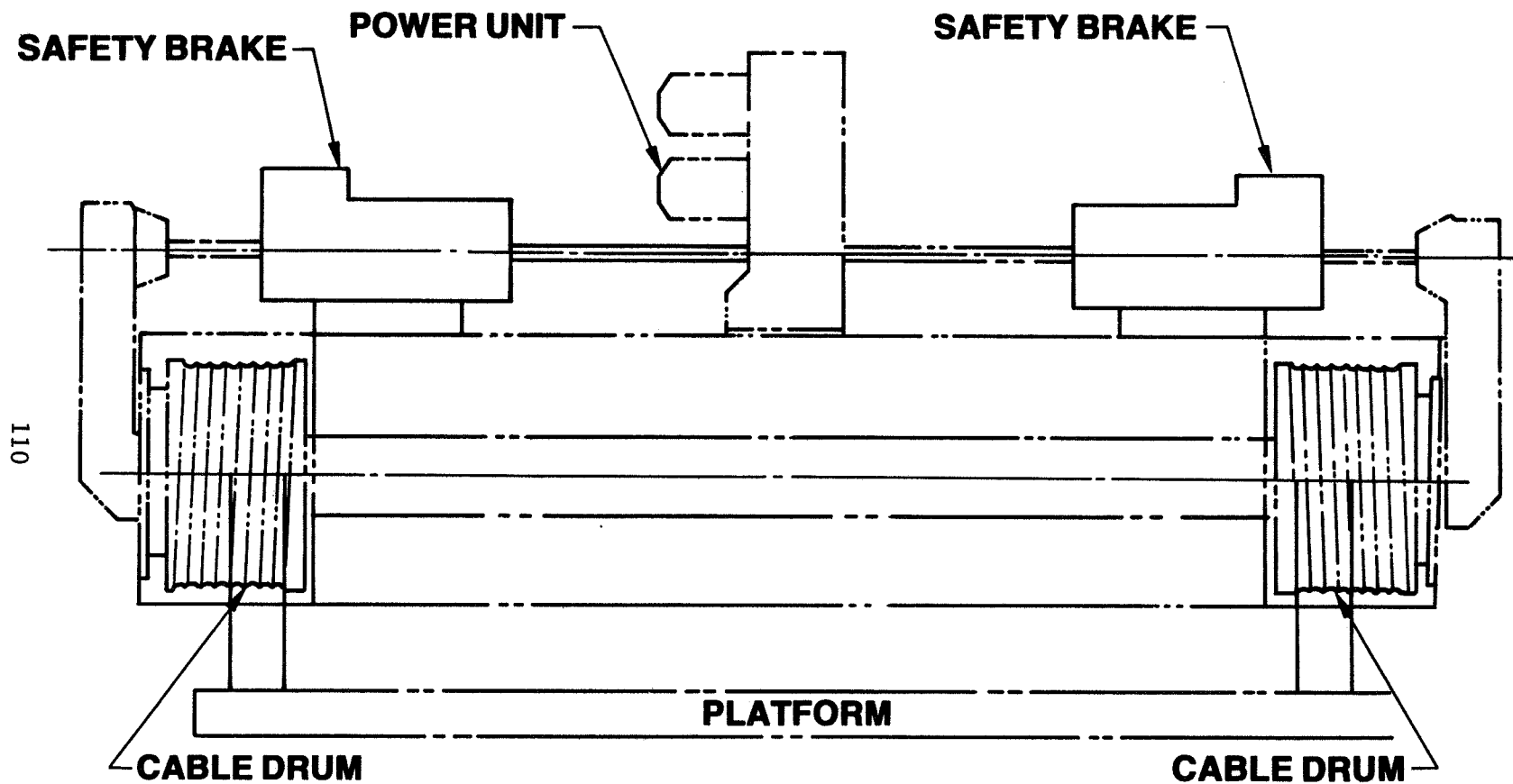## Driving An Aiding Load

INPUT SHAFT

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Figure 9

Constant Drag Brake

Figure 10
On-Board Loader Drive System