SAE Technical Paper Series

Gear Box Design with Flywheel For Reduced Vibrations and Energy Savings

Saul Herscovici

Power Engrg. & Mfg., Ltd.

FREEDOM THROUGH MOBILITY 1905-1980 75th anniversary

SOCIETY OF AUTOMOTIVE ENGINEERS, INC.
400 COMMONWEALTH DRIVE
WARRENDALE, PENNSYLVANIA 15096

800993

Gear Box Design with Flywheel for Reduced Vibrations and Energy Savings

Saul Herscovici
Power Engrg. & Mfg., Ltd.

HYDROSTATIC POWER TRANSMISSIONS are gaining in popularity because of their capability to provide infinitely variable speed; high power density; and the ability to mount the motor remotely by simply extending hydraulic hoses to connect the pump and motor. The simplicity of applying hydrostatic power transmission to a remote area is quickly appreciated when a mechanical drive needs to be placed through different angles or longer distances.

The ability to provide constant torque at an ever varying speed from maximum speed forward to maximum speed in reverse without stopping to change gears is of great advantage in many applications.

The illustration in (Fig. 1) shows a diagram of a variable displacement axial piston pump & fixed displacement motor that is used to power the gear box with flywheel described in this paper. The motor has a fixed angle swash plate to provide maximum torque output at any speed, while the pump has a variable angle swash plate that can change positions from +18^{o} to -18^{o} to provide a variable speed forward and reverse.

When such a hydrostatic transmission is used in conjunction with a speed reducing gear box of the type shown in (Fig. 2), it can provide a maximum continuous horsepower system that operates at a variable speed. A mechanical multiple speed transmission cannot be operated at maximum continuous horsepower system that operates at a variable speed. A mechanical multiple speed transmission cannot be operated at maximum continuous horsepower because when the need arises to reduce the speed due to overloading conditions, the percent change in speed when shifting gears is below maximum horsepower at engine rated speed.

While hydrostatic transmissions

ABSTRACT

Severe vibrations have been encountered in power drives that use hydrostatic power transmissions to provide the power at the working implement. Such vibrations often lead to failure of the hydraulic motor, hydraulic pump, hoses, speed reducing gear box, drive chain and other working components that may be a part of the power transmission system.

This paper describes the design approach taken to provide inertia by use of a flywheel to a speed reducing gear box driven by a hydraulic motor in an application where high torque peaks are present. A further improvement is obtained by the use of a slip clutch for applications where the implement is being brought to a complete stop in milliseconds.

The use of the flywheel reduces or eliminates the vibrations and associated failures and improves efficiency.

SAE/SP-80/469/$02.50

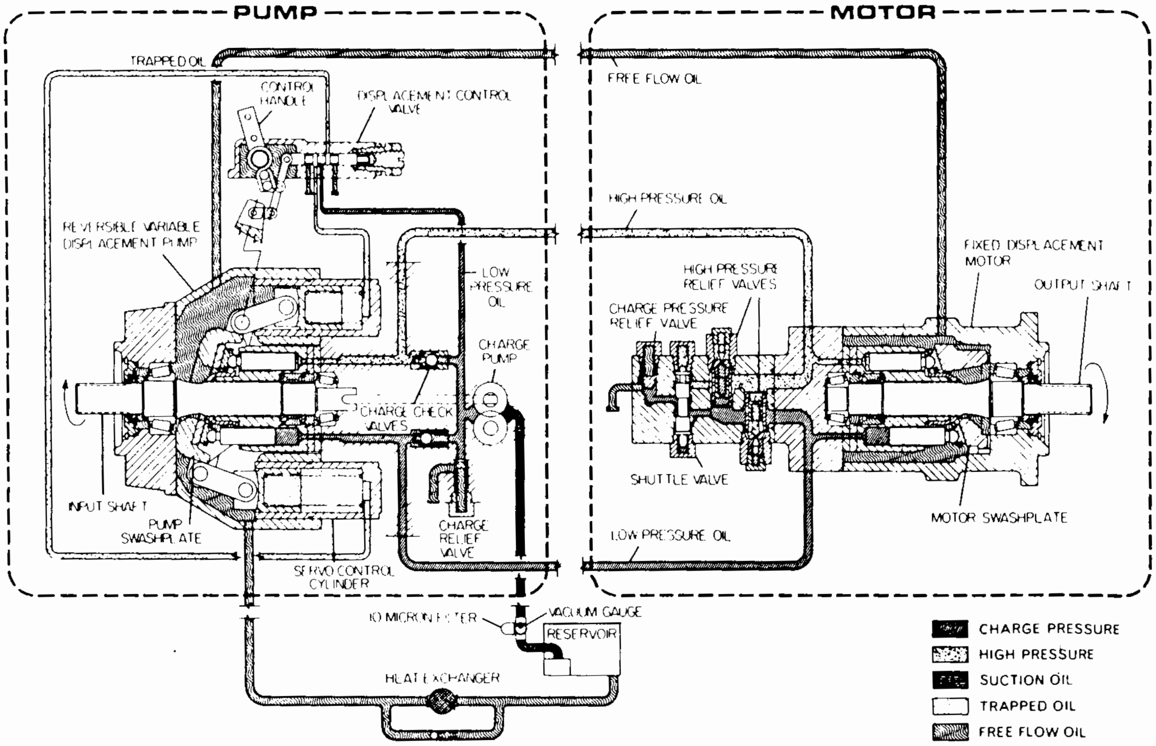


Fig. 1

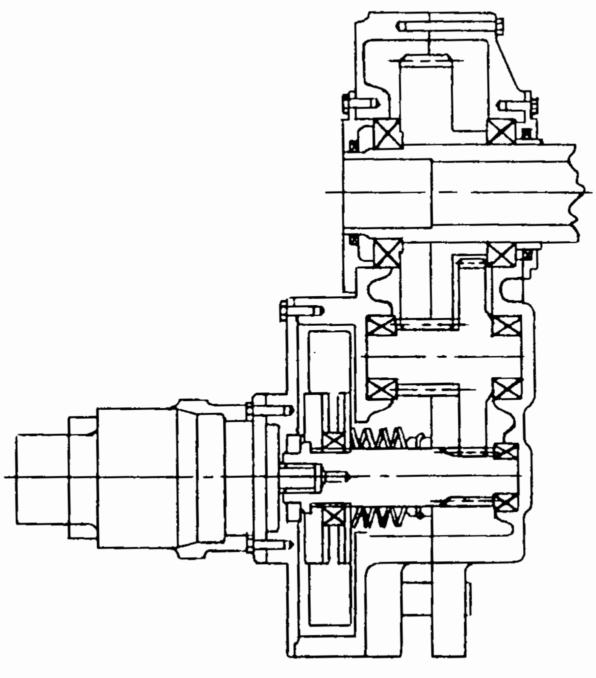


Fig. 2 - Speed reducing gear box with flywheel

are somewhat less efficient than mechanical transmissions, the equipment that uses hydrostatic transmissions in conjunction with a speed reducing gear box can be as productive and as fuel efficient as a straight mechanical transmission because of its ability to operate at maximum continuous horsepower at rated speed.

The speed reducing gear box shown on (Fig. 2) was designed and developed for a hydrostatically operated trenching machine to drive the excavating chain. The chain can dig a trench up to eight feet deep and up to twenty inches wide. The cutters may be positioned on the chain at intervals of up to 16 inches or more. The chain can operate at a speed of six to eight hundred feet per minute; thus, when the cutters cut and remove dirt to make the trench, the cutters generate a series of impacts at an approximate rate of four hundred impacts per minute and/or multiples of that. When the excavator was operated with the same gear box but with the flywheel removed, the hydraulic hoses were fatigued in a matter of hours and the engine stalled often because torque peaks were highly exceeding the engine torque. One day of operation without the flywheel was sufficient to prove that the inertia provided by the flywheel is necessary for satisfactory operation.

Rocks were often encountered deep in the ground and when the steel cutters impacted into the rock at a speed of six to eight hundred feet per minute, it created a high torque peak in the power drive that when sufficiently high would stall the engine.

When the cutters engage a large rock that cannot be dislodged, the chain comes to an instantaneous stop. The flywheel shown in (Fig. 2) has a slip clutch that allows the kinetic energy stored in the flywheel to be dissipated through friction whenever the chain comes to an abrupt stop and the gear box stalls in a matter of milliseconds.

The speed reducing gear box shown in (Fig. 3) was designed and developed for use on a mud pump that is used on water well or oil well drilling equipment. This pump has four pistons activated by a crankshaft that has the cranking arms unevenly arranged. This requires a fluctuating amount of power for each revolution. In addition to that, during the intake stroke the torque imparted to the piston is low because the fluid sucked into the piston chamber is lifted less than ten feet. During the compression stroke, the fluid is compressed to several hundred pounds of pressure per square inch.

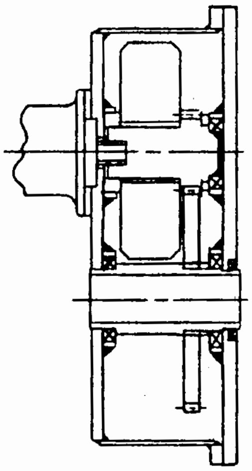


Fig. 3 - Speed reducing gear box with flywheel

Figure 4 shows hydraulic pressure recordings of a mud pump powered by a hydraulic motor with a 5:1 speed reducing gear box without a flywheel. It is interesting to note the pressure fluctuations in the pressure line required to provide the crankshaft torque for each piston. The pressure rise of the fluid in each mud pump piston is causing an equivalent pressure rise in the hydraulic pressure line. Equally important is the rapid fluctuation in the pressure return line. It can be seen that in some instances the pressure is reduced to a negative level. This means that the pressure return line operates under vacuum. When vacuum is present in the return line, the pistons in the hydraulic pump and motor can momentarily lift off the swash plate and, as they rotate, the piston shoes can impact on the ramp of the swash plate. This swash plate impacting leads to surface damage and eventually to complete failure.

Figure 5 shows the same application as in (Fig. 4), only the speed reducing gear box in this test has a flywheel that is 13.4 inches in diameter and 4 inches wide operating at 1750 rpm. This flywheel does not have a slip clutch because in this application an instantaneous stop cannot be encountered.

The size of the flywheel is determined by the amount of inertia torque required to overcome the vibration amplitude encountered by the hydraulic motor. The inertia torque is released while the flywheel is decreasing in speed; therefore, an acceptable speed reduction has to be taken into account.

It can be seen on (Fig. 4) that the hydraulic pressure line is subjected to a 5.3 cycles per second pressure fluctuation of about 2000 psi and the maximum level reaching 3000 psi. This speed reducing gear box is powered by an axial piston fixed displacement type hydraulic motor that provides a torque of 72 ft.-lb. per 1000 psi of hydraulic pressure. Therefore, the torque fluctuation in this instance is 144 ft.-lb. for 2000 psi pressure fluctuation.

If the flywheel inertia can develop a torque of 144 ft.-lb. over a period of 90 milliseconds, then the hydraulic motor will operate at constant pressure. Ninety milliseconds is half the cycle when energy is transferred from the flywheel into work while the second half of cycle of ninety milliseconds the engine stores more energy into the flywheel by increasing its speed for the next cycle.

The size of the flywheel is determined by trial and error in correlation to the required reduction in speed for a given set of operating conditions until an acceptable relationship is obtained.

The kinetic energy at the top speed and the speed reduction required to eliminate the pressure fluctuation at the hydraulic pump and motor has been calculated as follows:

$$KE = 1/2\ IW^2 \qquad (1)$$

$$W_1 = \frac{2\pi RPM}{60} = \frac{2\pi 1750}{60} = 183.2 \text{ rad/sec} \qquad (2)$$

$$M_f = \pi r^2 L \times .265 = \pi(6.7 \text{ in})^2$$

$$(4 \text{ in})\ (.265 \frac{\text{lb}}{\text{in}^3}) = 149.5 \text{ lb}_f \qquad (3)$$

$$I = \frac{M_f r^2}{2g} =$$

$$\frac{(149.5 \text{ lb}_f)\ (6.7 \text{ in})^2\ (\frac{1 \text{ ft}^2}{144 \text{ in}^2})}{2 \times 32.2 \text{ ft/sec}^2} =$$

$$.724 \text{ lb-ft-sec}^2 \qquad (4)$$

$$KE_1 = 1/2\ IW_1^2 = 1/2\ (.724 \text{ lb-ft-sec}^2)$$

$$(183.2 \frac{\text{rad}}{\text{sec}})^2 = 12{,}150 \text{ lb-ft} \qquad (5)$$

$$KE_2 = KE_1 - 144 \text{ ft-lb} \qquad (6)$$

$$KE_2 = (12{,}150 - 144) \text{ ft-lb} =$$

$$(12{,}006) \text{ ft-lb} \qquad (7)$$

$$W_2 = \sqrt{\frac{2\ (KE_2)}{I}} =$$

$$(\frac{2\ (12{,}006) \text{ ft-lb}}{.724 \text{ lb-ft-sec}^2})^{1/2} = 182 \text{ rad/sec} \qquad (8)$$

$$RPM_2 = (182 \frac{\text{rad}}{\text{sec}})\ (\frac{60 \text{ sec}}{\text{min}})$$

$$(\frac{\text{revolution}}{2\pi \text{ rad}}) = 1739 \text{ RPM} \qquad (9)$$

$$\Delta RPM = RPM_1 - RPM_2 =$$

$$(1750 - 1739) \text{ RPM} = 11 \text{ RPM} \qquad (10)$$

This shows that the gear box with the hydraulic motor and flywheel shown on (Fig. 3) operates between 1739 and 1750 rpm operates at 1744.5 ± 5.5 rpm.

The 11 rpm variation of the hydraulic motor every ninety milliseconds will tend to pulsate the hydraulic lines and the engine because of the equivalent variation in the volume of the hydraulic oil used by the motor. This pulsation can be removed by the addition of a hydraulic accumulator. The accumulator will accommodate the oil volume pulsation, thus allowing the engine to operate at constant speed.

The fixed displacement piston-type hydraulic motor used on this application has a volume displacement of 5.43 cubic inches per revolution; therefore, the required size of the accumulator is:

$$\Delta \text{Accumulator Vol.} - \text{Avg. } (\Delta RPM)$$

$$\times (\text{Time}) \times (\frac{\text{Displ.}}{\text{Revolution}}) = 1/2\ (11 \frac{\text{rev}}{\text{min}})$$

$$(.090 \text{ sec} \frac{\text{min}}{60 \text{ sec}})\ (\frac{5.43 \text{ in}^3 \pi}{\text{revolution}})$$

$$= 2.69 \text{ in}^3 \qquad (11)$$

Figure 6 shows that the use of an accumulator enables the engine and the hydrostatic pump to operate at constant speed even though the speed of the hydraulic motor is fluctuating.

When a gear box that has a flywheel with a slip clutch (Fig. 2) comes to a complete stop in a fraction of a second, it can generate a torque at a level determined by the slip clutch torque which should be set somewhat higher than the shaft torque on which it is mounted.

If a gear box is designed with a flywheel that does not have a slip clutch (Fig. 3) and the application is such that it can be brought to a complete stop in ninety milliseconds as in the previous illustration, the torque generated by the flywheel mass moment of inertia may be sufficiently high to break gear teeth, shear shafts or damage any other components.

The torque generated by the flywheel mass moment of inertia when brought to zero rpm from 1750 rpm within ninety milliseconds is calculated by determining the rate of deceleration as follows:

$$\alpha = \frac{\Delta W}{\Delta t} = \frac{1750 \frac{\text{rev}}{\text{min}}\ (\frac{2\pi \text{ rad}}{\text{rev}})\ (\frac{\text{min}}{60 \text{ sec}})}{(.090 \text{ sec})}$$

$$= 2036 \frac{\text{rad}}{\text{sec}^2} \qquad (12)$$

$$T = I\alpha = (.724 \text{ lb-ft-sec}^2)\ (2036 \frac{\text{rad}}{\text{sec}^2})$$

$$= 1474 \text{ ft-lb} \qquad (13)$$

Average torque to decelerate or accelerate the flywheel 11 rpm in .090 seconds: (14)

$$T = (.724 \text{ lb-ft-sec}^2)\left(\frac{\frac{11 \text{ rev}}{\text{min}} \; \frac{2\pi \text{ rad}}{\text{rev}} \; \frac{\text{min}}{60 \text{ sec}}}{.090 \text{ sec}}\right) = 9.26 \text{ ft-lb}$$

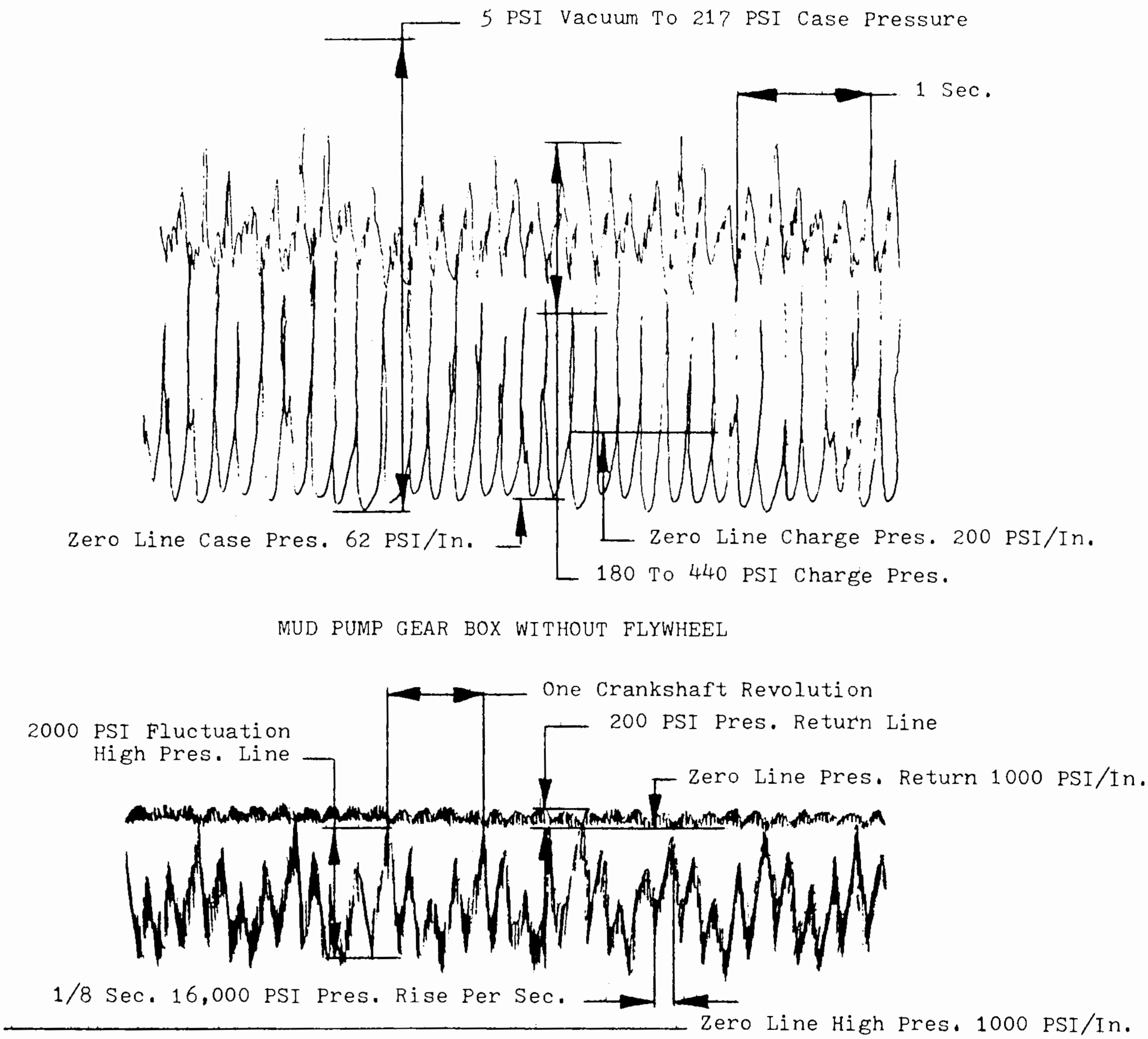


Fig. 4

From the above calculations, it is determined that a torque of 1474 ft-lb is generated when the flywheel is being stopped in ninety milliseconds while the maximum operating torque at 3000 psi is 216 ft-lb. This shows that the inertia torque peak is approximately six and one-half times larger than the operating torque. If the flywheel stopping time is further reduced, the inertia torque peak further increases.

This extremely high torque developed when the gear box is stalled in milliseconds, illustrates the need of a slip clutch that allows the flywheel to continue to rotate and dissipate its energy through friction in the form of heat.

This slip clutch is partially submerged in oil; thus the heat is removed by the oil.

Figure 7 shows the need of the flywheel slip clutch in limiting the

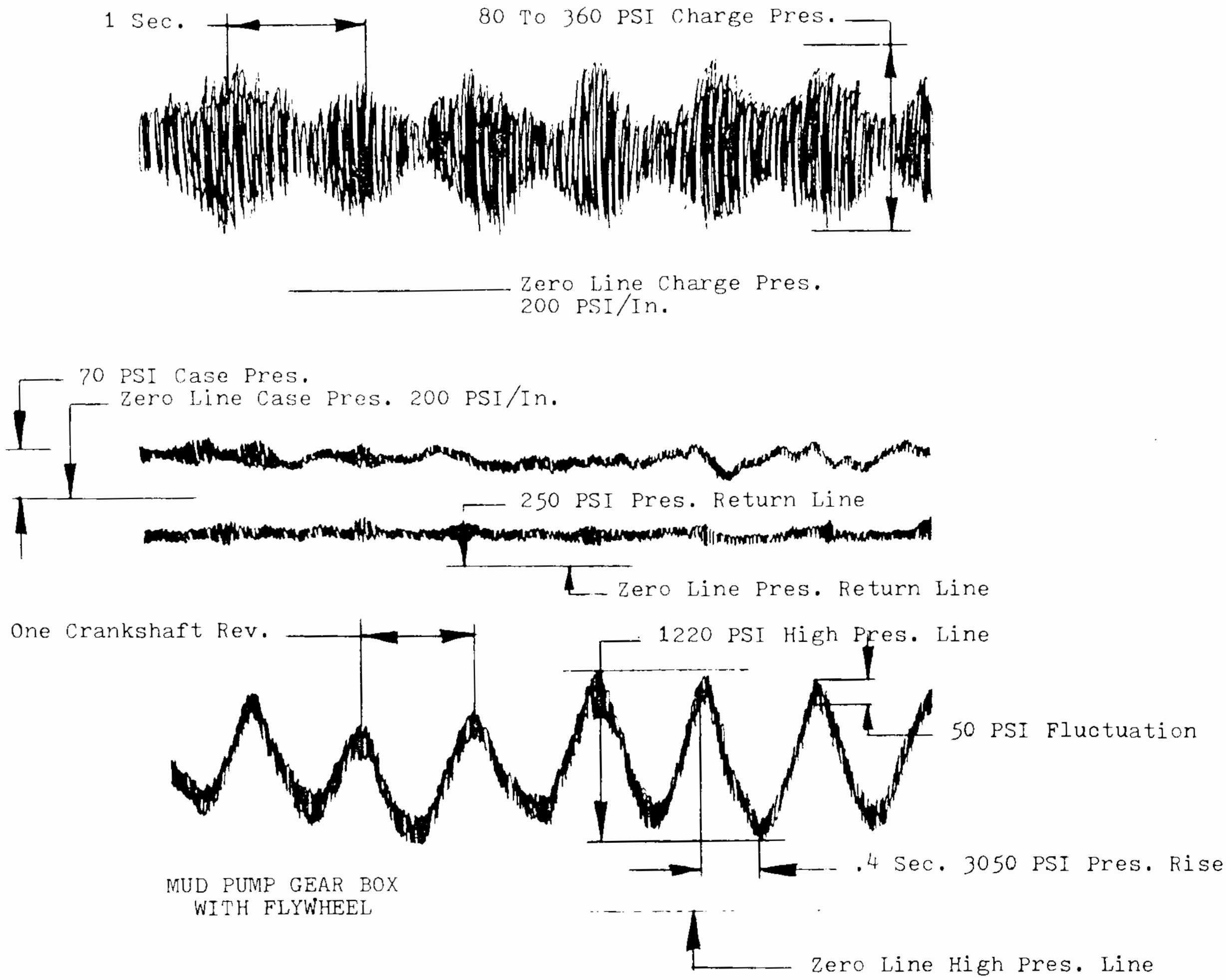


Fig. 5

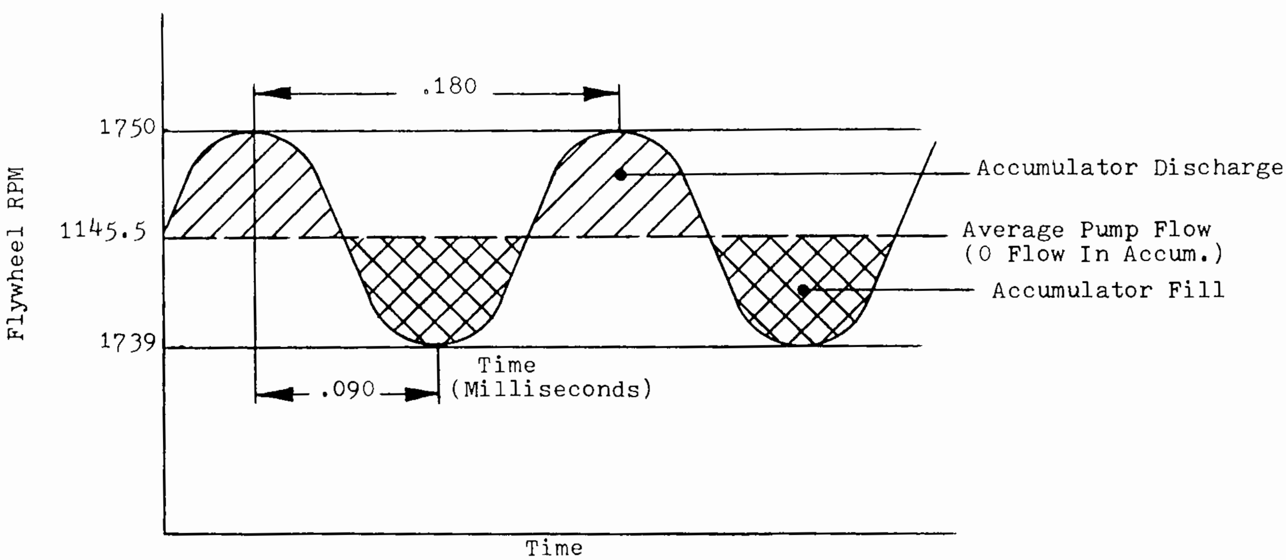


Fig. 6 - Accumulator fill and discharge cycle

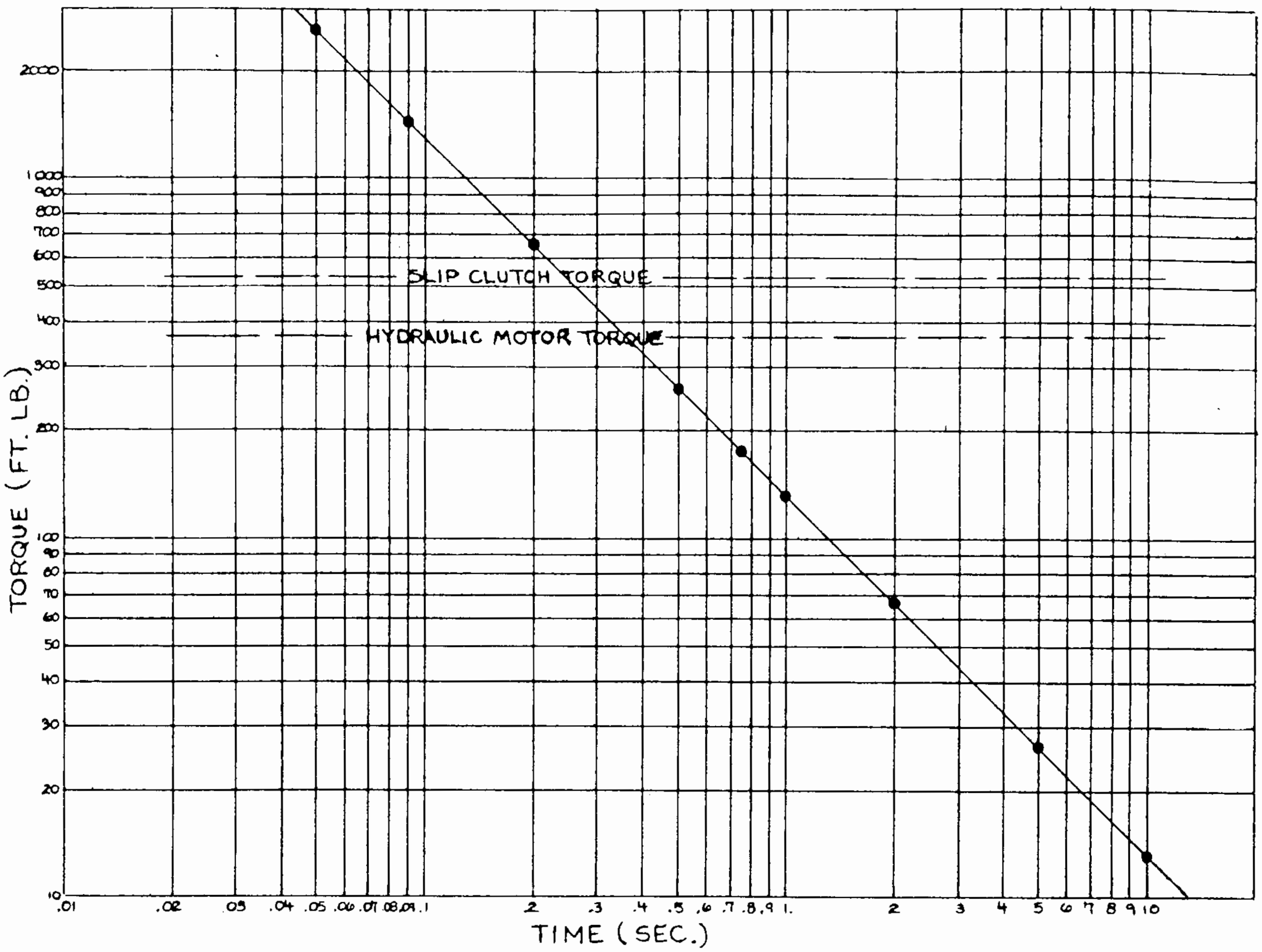


Fig. 7

torque seen by the gear box and power drive components.

Table 1 shows different rates of deceleration used in (Fig. 7) that further illustrates the high levels of energy that the flywheel can develop when brought to a complete stop over different lengths of time.

The flywheel with slip clutch that is shown in (Fig. 2) is designed in such a way that when the flywheel inputs torque into the gear box at its maximum level that is also the slipping level, it transmits torque in addition to the torque transmitted by the hydraulic motor. Therefore, if the clutch is set to slip at one and one-half times the working torque, the torque level in the power transmitting components will be at two and one-half times the working torque level because the torque generated by the hydraulic motor combines with the flywheel torque at the slip level.

Table 1 - Flywheel Kinetic Energy Vs. Time

Stopping Time (Sec)	Deceleration Rate (Rad/Sec)	Torque (Lb-Ft)
.050	3665	2653
.090	2036	1474
.200	916	663
.500	367	266
.750	244	177
1.000	183	132
2.000	92	67
5.000	37	27
10.000	18	13

Caution must be exercised in the selection and design of components for the gear box with flywheel because of the presence of the increased torque level that can exist (as pointed out in the above paragraph). Even though the presence of the increased torque level is of a short duration, it can cause damage and failure to the power transmitting components.

Figure 8 shows the horsepower output of an engine used in a hydraulic transmission application where the speed reducing gear box does not have a flywheel. The high torque fluctuations shown on (Fig. 4) cause an equivalent fluctuation in engine horsepower output. Full advantage of the engine flywheel cannot be taken because the engine governor cannot respond in milliseconds as the vibration frequency demands. For that reason, the usable engine horsepower is lower than the rated engine horsepower. Here the friction engine losses become a larger user of fuel than at rated engine horsepower. In addition to that, some unburned fuel is emitted because of the high torque fluctuations from one cycle to another.

the system operation.

The elimination of torque peaks completely removed or reduced to an acceptable level the rate of failure of the hydrostatic transmission; the speed reducing gear box; the implement and machine frame.

Because of low torque peaks and a steady state operation, the engine can be operated at higher torques at the rated engine speed; thus, enhancing the equipment productivity to be equal or exceeding that of a mechanically driven implement.

DEFINITIONS

T - Torque

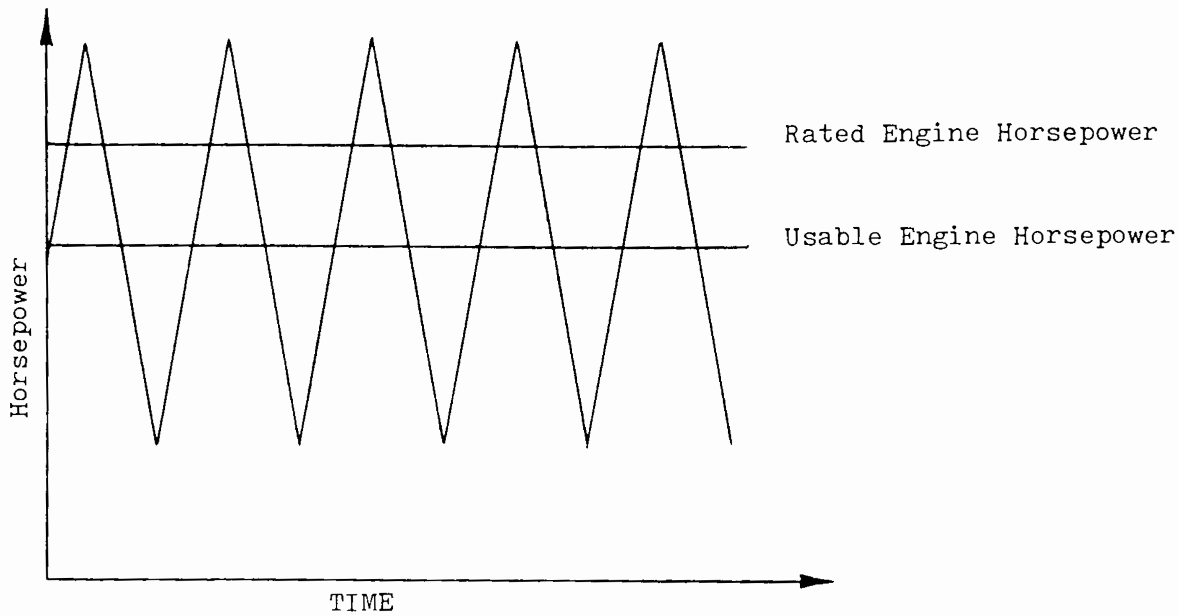


Fig. 8 - Engine operates a hydrostatic power transmission with a speed reducing gear box without a flywheel

Figure 9 shows that the rated and usable engine horsepower are the same for a hydrostatic power transmission that is used with a speed reducing gear box that has a flywheel. This system is more fuel efficient because the losses of the system described in Figure 8 are not present.

CONCLUSION

It was proved on several occasions that high failure rates cannot be reduced in hydrostatically driven implements which have high torque fluctuations without the use of the flywheel.

The addition of the flywheel provided sufficient inertia to reduce significantly the amplitude of the damaging torque peaks and stabilize

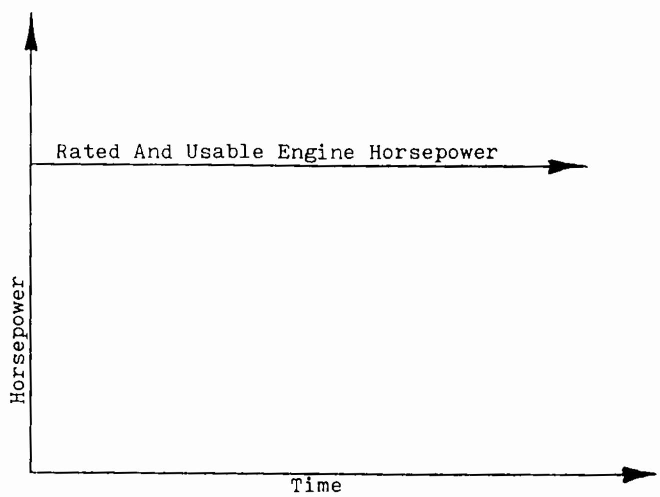


Fig. 9 - Engine operates a hydrostatic power transmission with a speed reducing gear box with a flywheel

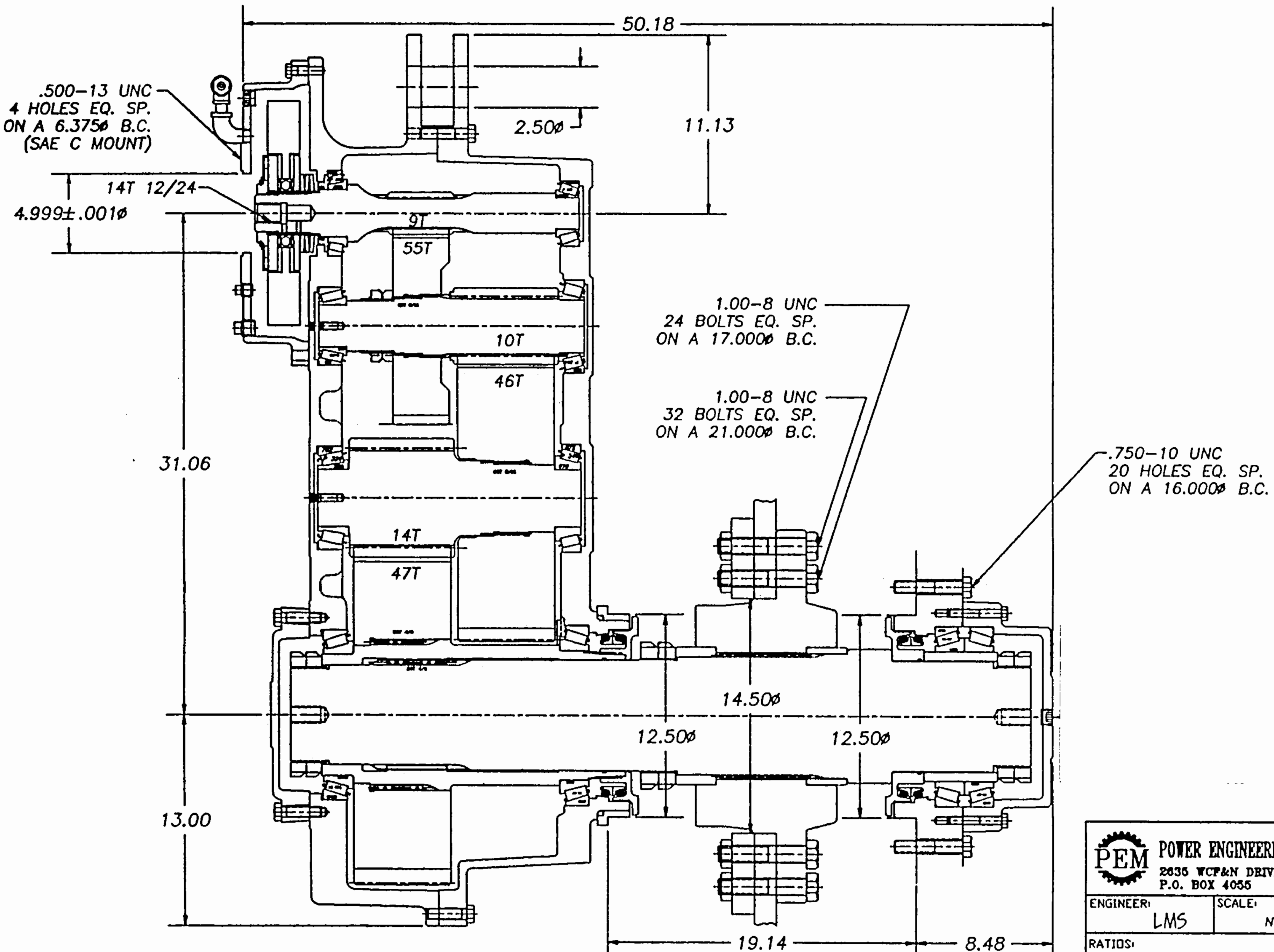
I - Mass moment of inertia
α - Angular acceleration
M_f - Mass (expressed in pounds of force)
KE - Kinetic energy
W - Angular velocity
t - Time
Δ - Change
r - Radius
g - Constant of gravity

ISSN 0148-7191

Society of Automotive Engineers, Inc.
400 COMMONWEALTH DRIVE, WARRENDALE, PA 15096

____ page booklet.

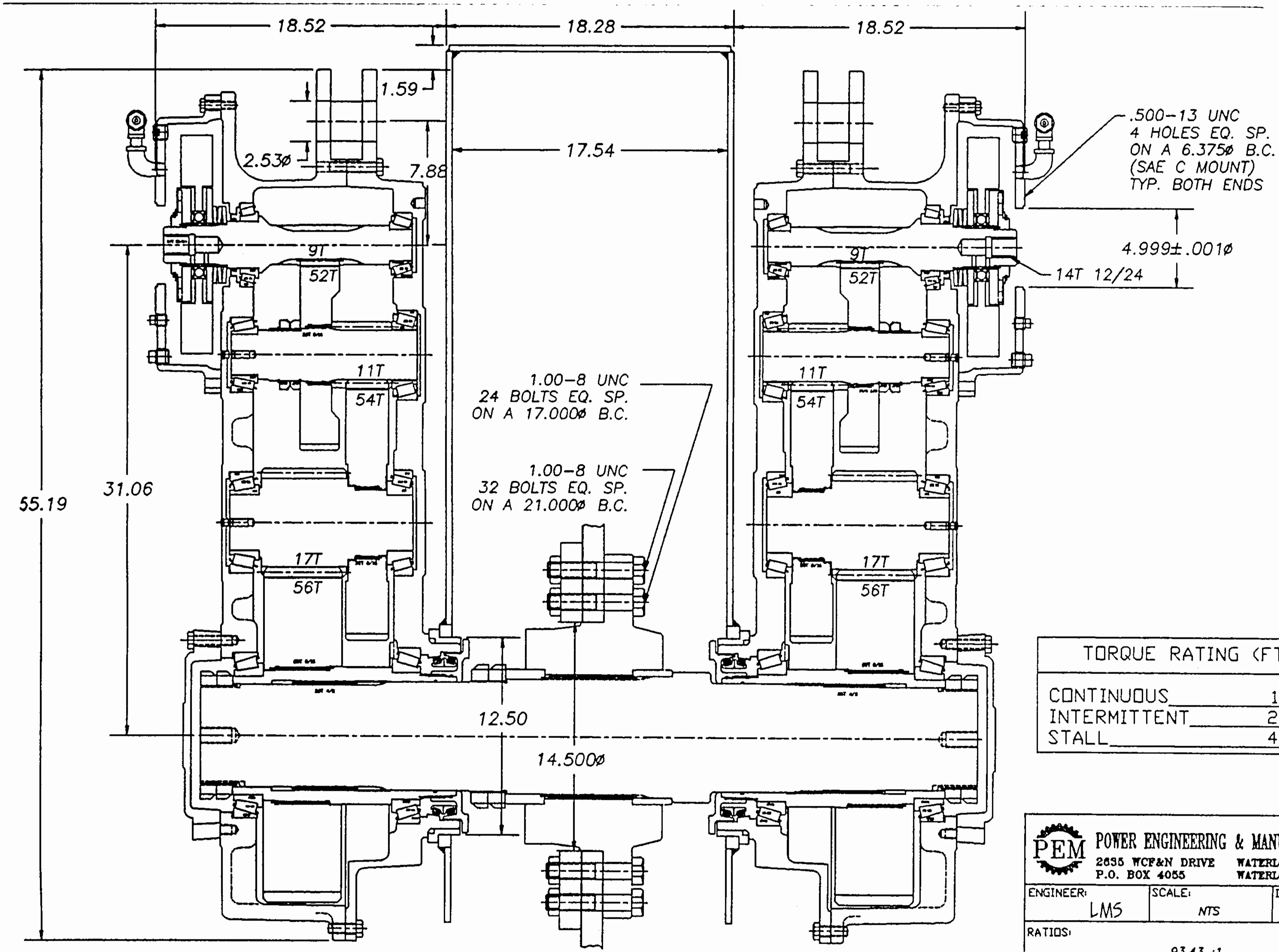
Printed in U.S.A.



PLOT DATE: 01-24-93

PEM POWER ENGINEERING & MANUFACTURIN
2635 WCF&N DRIVE WATERLOO, IA 507
P.O. BOX 4055 WATERLOO, IA 507

ENGINEER: LMS	SCALE: NTS	DATE: 1/20/9
RATIOS: 94.37 :1		
OUTPUT RATED TORQUE (FT-LB)		WEIGHT:
NAME: SRF-19		



TORQUE RATING (FT-LB)	
CONTINUOUS	16,700
INTERMITTENT	29,600
STALL	44,400

PEM POWER ENGINEERING & MANUFACTURING LT
2635 WCF&N DRIVE WATERLOO, IA 50703 U
P.O. BOX 4055 WATERLOO, IA 50703 U

ENGINEER:	SCALE:	DATE:
LMS	NTS	1/19/94

RATIOS: 93.43 :1

OUTPUT RATED TORQUE (FT-LB) WEIGHT:

NAME:

PLOT DATE: 01-20-93