Power Engineering & Manufacturing Ltd.
2635 WCF&N Dr. ♦ P.O. Box 4055 ♦ Waterloo, Iowa 50704-4055 U.S.A.
Phone (319) 232-2311 FAX (319) 232-6100

NATURAL FREQUENCY OF TORSIONAL VIBRATIONS IN SUGAR MILL DRIVES

By Saul Herscovici, President
Power Engineering & Manufacturing, Ltd.

Presented At The American Society Of Sugar Cane Technologists' Twenty-Ninth Annual Joint Meeting

Radisson Beach Resort
Ft. Walton Beach, FL
June 16-18, 1999

NATURAL FREQUENCY OF TORSIONAL VIBRATIONS

IN SUGAR MILL DRIVES

ABSTRACT

Natural frequencies of vibration are present in all mechanical drive line systems. This property is derived from the shafts that represent a torsional spring with mass plus the gears turbine rotor and rolls that posses a mass. Such a mass spring system can be defined mathematically for a roll drive installation from the tip of the turbine all the way to the end of the rolls. The natural frequency of the entire drive system can thus be calculated. It is desirable to have a natural frequency of vibration higher than the operating speed by more than fifteen percent. If the natural frequency of vibration is at or near the operating speed ,then a vibration damper must be incorporated in the drive system to reduce the vibrating amplitude to zero or as close as possible to zero. If the drive system is operated at or near the natural frequency of vibration without damping, then the system will vibrate and possibly leading to failure.

INTRODUCTION:

Two identical failures occurred on a planetary drive one year apart where an internal gear tooth in a planetary ring gear broke in a wedge shape about three inches long and a half inch wide at one end and curved to a point at the other end. When the design and manufacturing analysis showed no weakness or reason for breakage, we resorted to study the system's natural freqency.

WHAT IS A NATURAL FREQUENCY?:

The natural freqency of vibration is a law of nature that can best be illustrated by imagining that a ping pong ball is bouncing back and forth between two walls, but every time the ball hits the wall, the wall pushes the ball away and as a result, the ball moves away from the wall faster than it came. If the ball speed increases with every stroke between the walls, eventually it will reach a speed that will have sufficient energy to crush the ball. All systems do not increase the energy of the vibrating member until destruction occurs. Some vibrating members are excited and the vibrations fade away slowly.

The best example of a device that operates at the natural frequency of vibration is a musician's tuning fork. When excited, it vibrates at a fixed frequency for several seconds. It produces a reliable musical note because it always vibrates at the same frequency because its geometry is fixed. This is precisely why the musicians use it. They can produce the same musical note all around the world.

A good example of a damper is the shock absorber in the car. If all shock absorbers were removed from the car, it would undulate until the passengers would become sea sick. When good shock absorbers are present in the car, the car travels stable. The energy transferred from the axle housing through the springs when traveling over rough terrain, or a bump, is absorbed by the fluid in the shock absorber and converted into heat. The amount of heat is very small, and for that reason, the shock absorbers do not overheat.

Another example of a good damper, but never used except for demonstration, is a rubber band tightly set on a musician's tuning fork.

In most cases, the vibration energy that needs to be captured by a damper to eliminate vibrations or reduce the amplitude is a minuscule fraction of the total energy traveling through the system.

Fig. 1

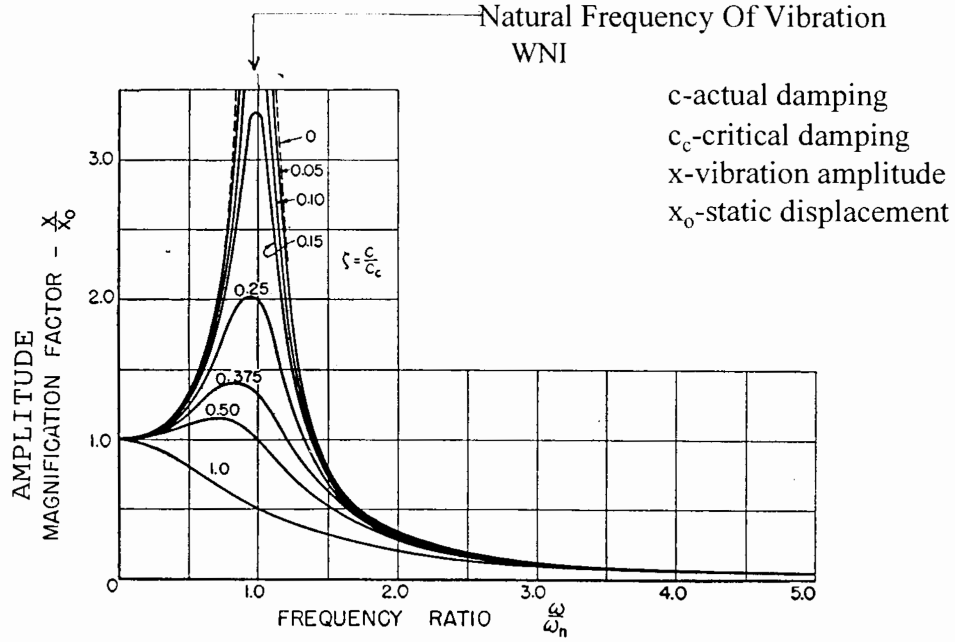


Figure 1 shows a series of curves of vibration amplitude vs frequency ratio of actual damping to critical damping. Two characteristics can be observed: One is that the amplitude of vibration diminishes as the amount of damping increases. Note that when the damping is at its maximum level, as shown on the lowest curve marked 1.0, there are no vibrations present. As the amount of damping is reduced, the vibration amplitude increases as shown on Figure 1. C denotes the amount of available daruping and C_c is the critical damping that is required to eliminate vibrations. All systems have some amount of damping, but most of the time it is too small to eliminate damaging vibrations. Even the tuning fork has friction damping from two sources: One is the friction with the air when the forks move rapidly, and the second is the steel internal friction represented by its hysteresis property. Damping also exists inside the gear box and that is the viscous damping provided by the oil. Some friction damping is also present in the gear box because the gears and bearings experience sliding as they rotate. More damping is provided by the sugar cane as it passes through the rolls. But, here again, this damping is insufficient to eliminate vibration when any of the rotating shafts operate at the natural frequency. For this reason, additional damping is required.

The tuning fork example may be the simplest vibrating device, and the vibrations are initiated by pinching the forks one single time. In a sugar mill drive, not only that we have a complex system with many masses and many torsional springs, but torque is applied continuously to every drive member. The force imposed upon every member can and does excite vibrations.

Figure 2 shows that a phase angle change of the vibrating masses occurs as a function of damping. The smaller the amount of damping, the larger the phase angle change becomes.

Fig. 2

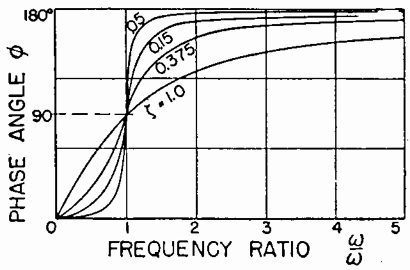


Figure 3 shows a schematic representation of a sugar mill drive that has eight rotating masses and seven torsional springs. The arrows on the rotating masses show an arbitrary but possible instantaneous direction of vibration motion of each member. The members can have vibration movement in opposite direction to each other inside a gear box and cancel each other to show no vibration at the input or output where it can be observed. Such a vibrating condition can still lead to damage, and it is very difficult to detect.

The second thing is that the natural frequency of vibration occurs at a single point. This feature allows often to make a simple geometry modification to a shaft to move the natural frequency of vibration above or below the operating speed. Unfortunately, this is not the case with sugar mill drives because the shafts are long and thus make good torsional springs, and also because the mass of the rolls is enormous.

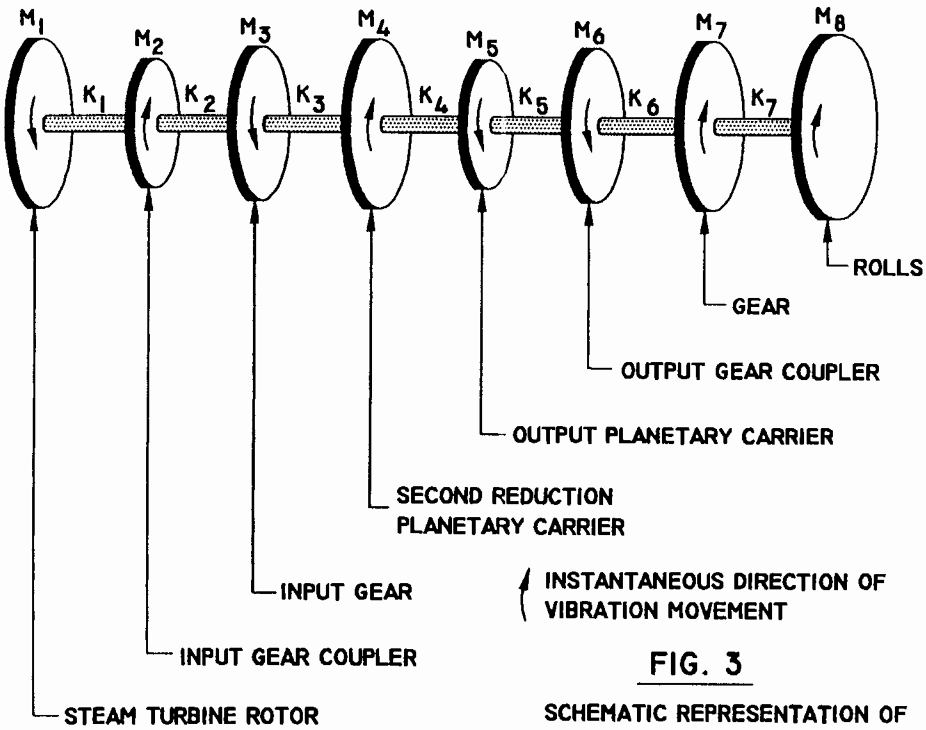


FIG. 3

SCHEMATIC REPRESENTATION OF OF A SUGAR MILL DRIVE

Equation 1 shows how to calculate the natural frequency of vibration for a simple system that consists of one mass and one spring as shown on Figure 2. K is the spring rate, M is the mass, and *f* is the system natural frequency. It can be seen from Equation 1 that as the spring rate, K, increases, the natural frequency *f* increases, and as the mass, M, increases, the natural frequency, *f*, decreases. It is best to make the spring rate as high as possible and at the same time to make the mass as small as possible in order to maximize the natural frequency of vibration to make it be substantially above the operating speed range.

EQ. 1 $f = \sqrt{K / M}$

K-spring rate (lb-in)
M-Mass (w/g in lb-sec 2/in)
W-weight (lb)
g-force of gravity (32.2 lb/sec^2)

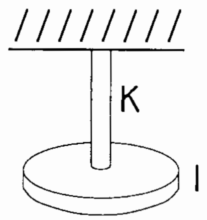
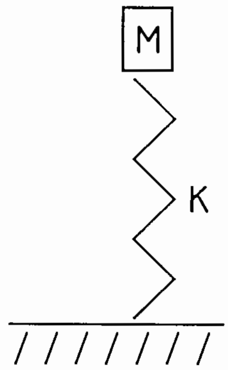


FIG. 4



To further complicate the phenomenon of natural frequency, the mass moment of inertia increases or decreases with the square of the speed of the rotating members. This means that if one shaft rotates twice as fast as another shaft, then Equation 3 shows that the mass moment of inertia of the faster shaft is four times greater than that of the slower shaft when both shafts are equal.

$$\text{EQ. 2} \qquad (I_2 / I_1)^2 = (2 / 1)^2 = 4$$

The amplitude of the torsional natural frequency of a mechanical system that has many springs and masses can be solved by trial and error by the Holzer method.

Once the natural frequency is calculated, the amplitude of vibration can be calculated.

The amplitude of vibration can be reduced by the addition of damping, as shown on Fig. 1. It is most effective to incorporate damping in the high speed gear coupler that is located between the steam turbine and the gear box. Some gear couplers such as incorporate a rubber member that has vibration damping capacity. The manufacturer specifies the damping, torque and speed operating characteristics.

One hundred percent reduction in the amplitude of vibration may be too difficult to achieve, but it should be strived to accomplish over ninety percent reduction at the input gear coupler. Some additional damping is provided by the lubricating oil in the gear boxes and the sugar cane as it passes between the rolls. This amount of damping provided by the gear oil and sugar cane is insufficient to provide trouble-free operation at the natural frequency of vibration. In addition to that, at times the rolls are rotated without sugar cane. This is even a more critical operation because no damping is provided by the sugar cane.

RESULTS:

During the 1996 and 1997 sugar cane seasons, two gear boxes, our WH-325005, were operated at an input speed near or even at the natural frequency speed without any damping. Vibrations could be felt at the base of both gear boxes and even on the concrete floor. During the 1998 season, these two gear boxes were equipped with input gear couplers that have a rubber damper. Vibrations at gear box base or at the concrete could not be observed.

The WH-8M gear box installed at the Patout Sugar Mill as shown on Figure 5, was also operated during the 1998 sugar cane season. Calculations show that the turbine speed is at or near the natural frequency speed. All efforts made to change shaft geometry to raise the natural frequency above the operating speed were in vain. To reduce or eliminate vibrations, the input gear coupler was equipped with a rubber member that has damping characteristics. No vibrations were observed at any speed, including the natural frequency speed.

It must be proceeded with caution, because it is not know if there are vibrations internally inside the gear box in such a phase angle that they self cancel before they arrive at the output shaft.

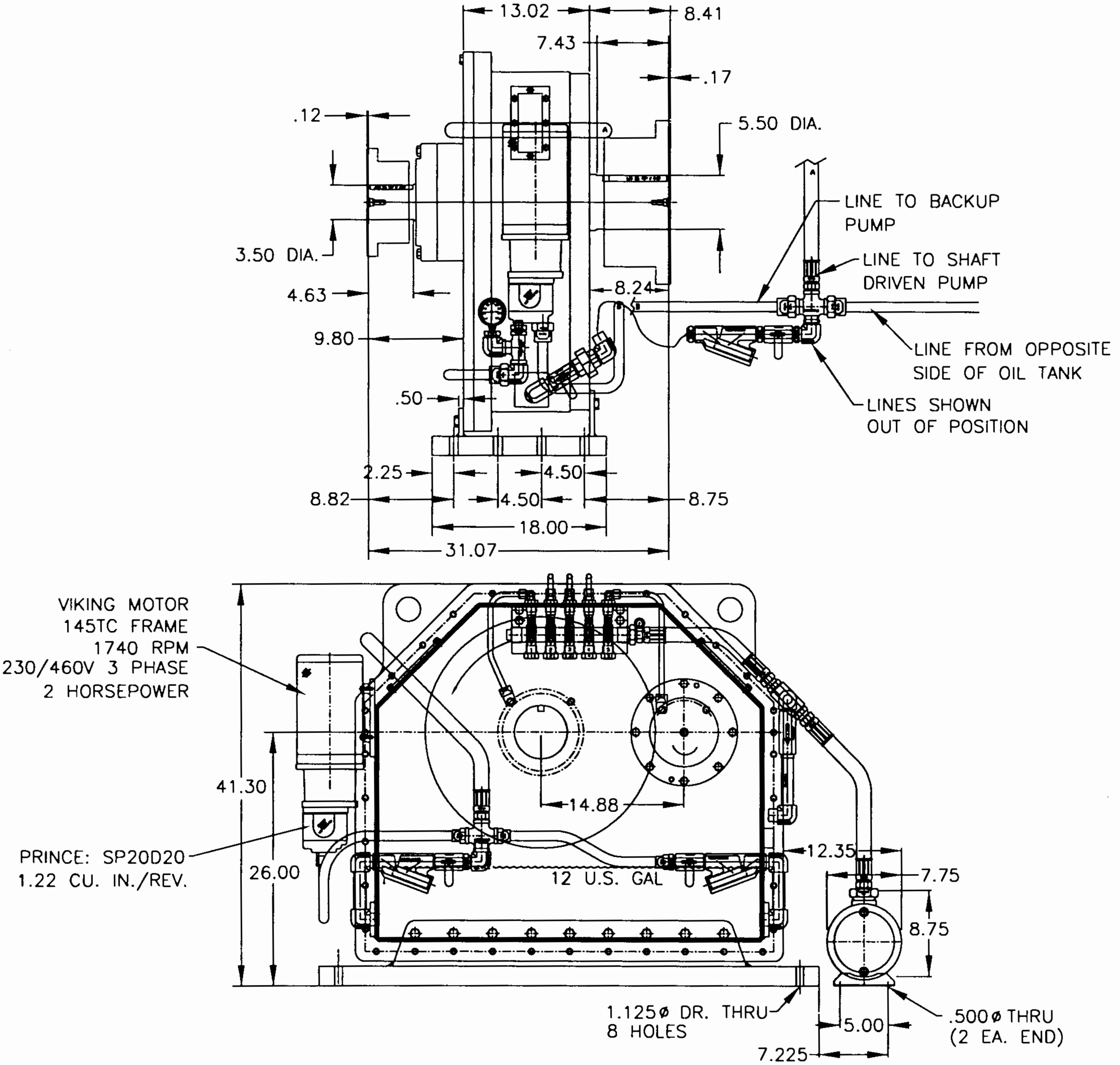
The WH-500,002 that operates at the New Iberia Sugar Mill has a turbine speed that is substantially above the natural frequency speed.

We also manufactured one shredder drive gear box and two knive drive gear boxes. The torsional natural frequency of these gear boxes are substantially higher than the operating speeds because there is only one gear reduction in the drive and the shafts are short.

SUGAR MILL SHREDDER DRIVE GEAR BOX

FMSR-16

FOOT MOUNTED SPEED REDUCER



Other ratios available. For more imformation call factory engineering department.

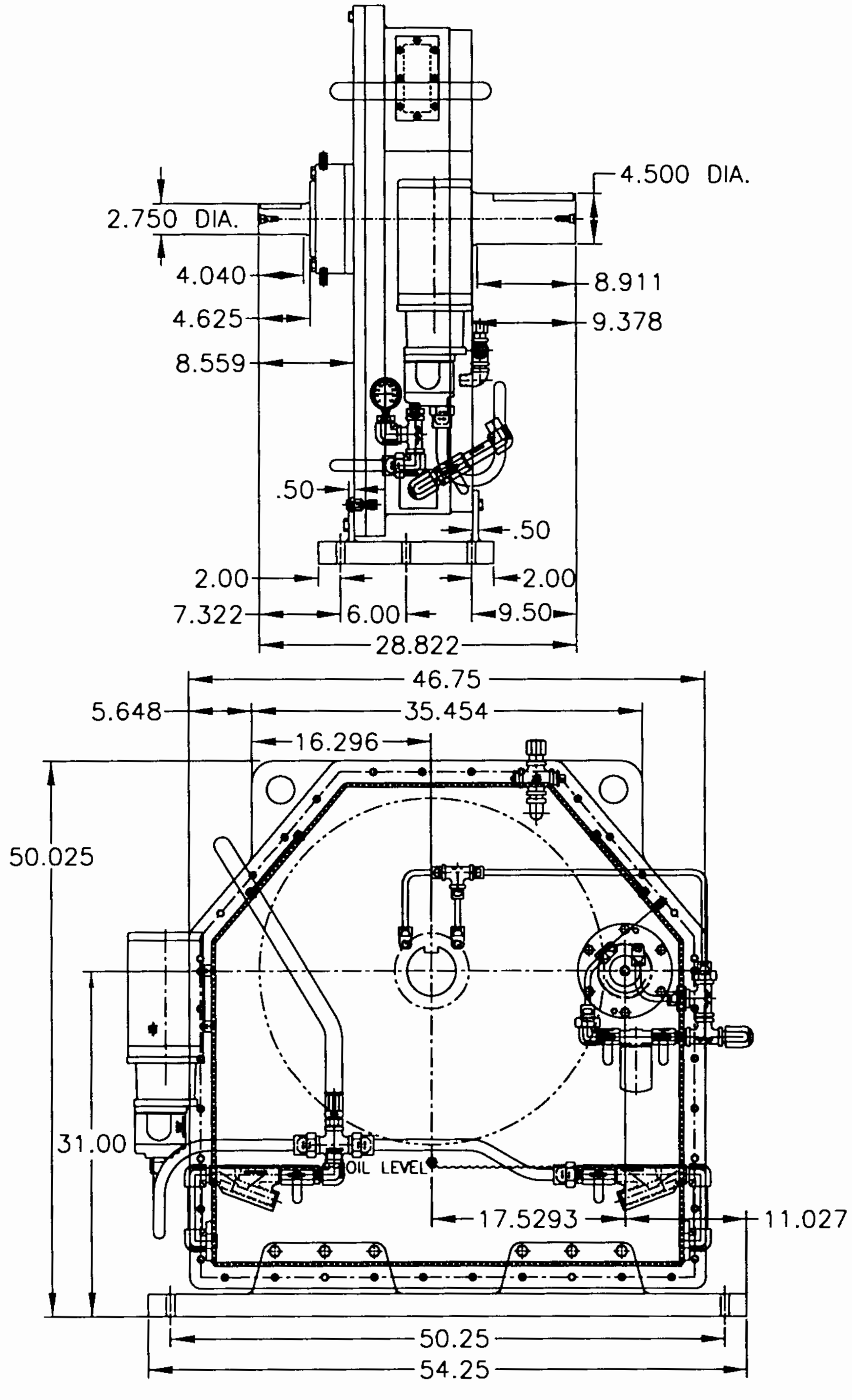
MODEL	RATIO	OUTPUT TORQUE RATING, FT-LB			WEIGHT (LBS)
		AGMA 1	AGMA 2		
		INTERMITTENT	CONTINUOUS	STALL	
FMSR-16	3.94:1	18000	9000	27000	3000#

3500 HP AT 4000 RPM INPUT AT AGMA 3.9

SUGAR MILL KNIFE DRIVE GEAR BOX

FMSR-17

FOOT MOUNTED SPEED REDUCER



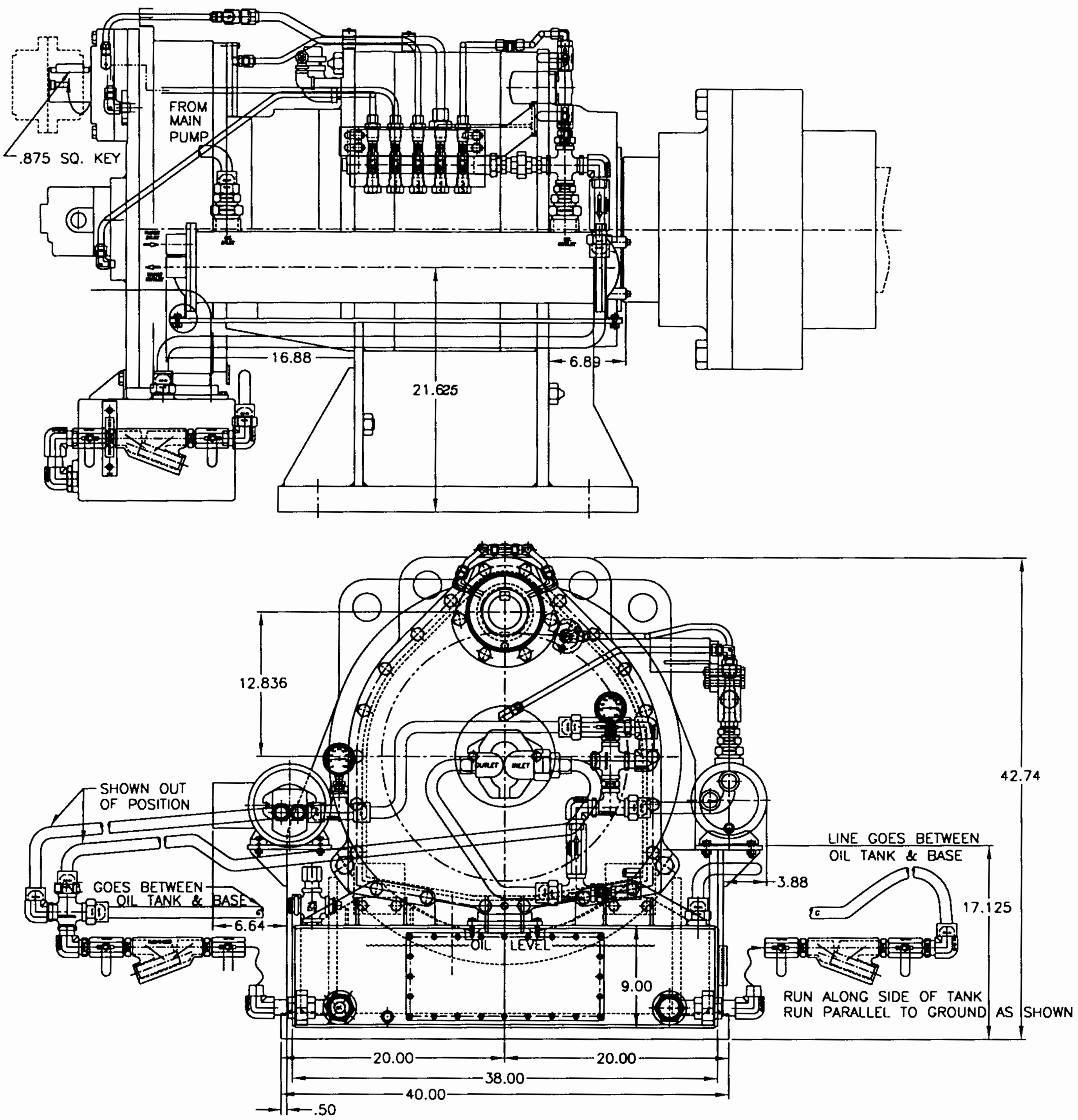
Other ratios available. For more imformation call factory engineering department.

MODEL	RATIO	OUTPUT TORQUE RATING, FT-LB			WEIGHT (LBS)
		AGMA 1	AGMA 2		
		INTERMITTENT	CONTINUOUS	STALL	
FMSR-17	7.909:1	6218	3109	9327	3500#

1200 HP AT 4750 RPM INPUT AT AGMA 4.7

WH-325005

PLANETARY SPEED REDUCER

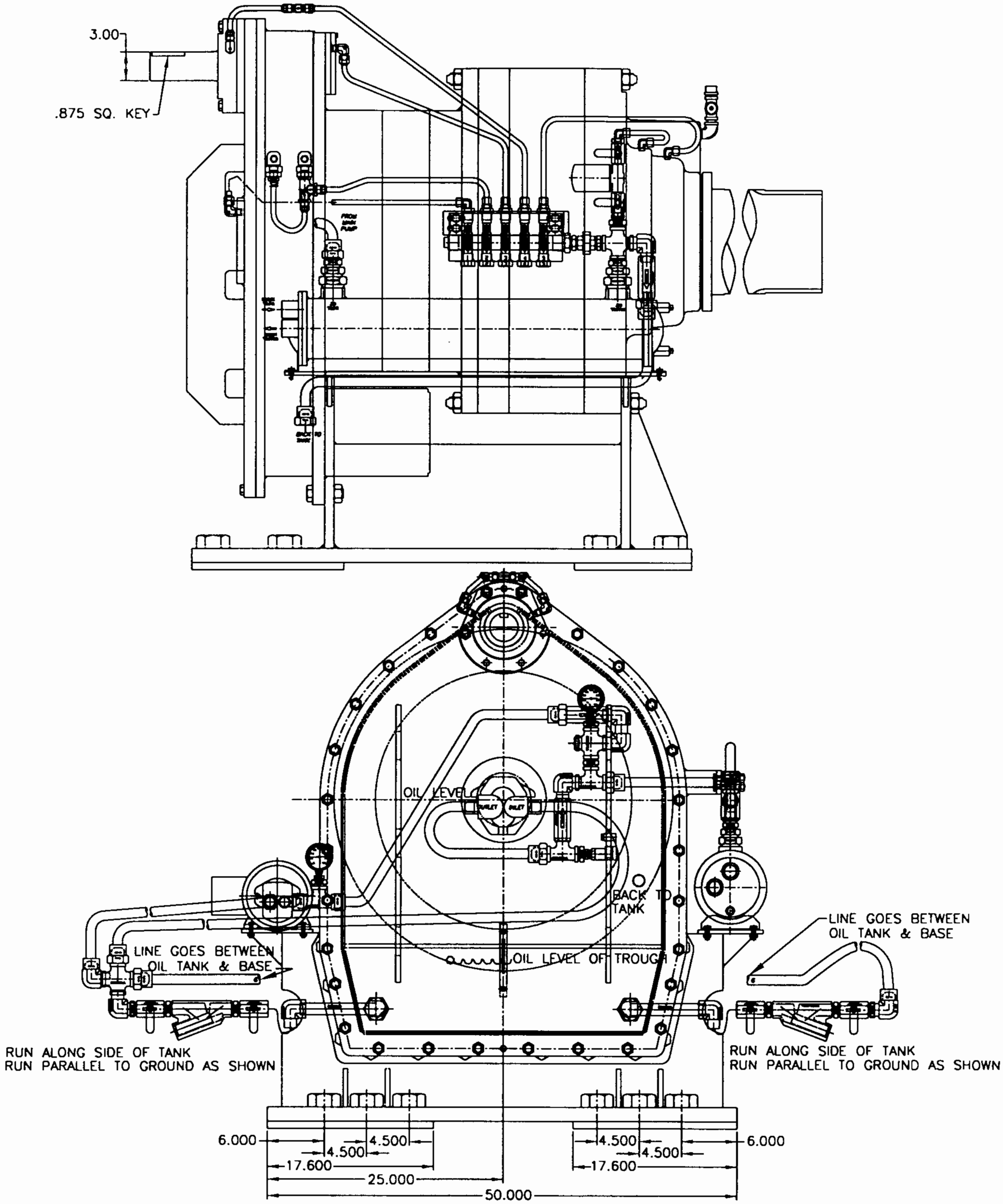


Other ratios available. For more imformation call factory engineering department.

MODEL	RATIO	OUTPUT TORQUE RATING, FT-LB AGMA 1 INTERMITTENT	AGMA 2 CONTINUOUS	STALL	WEIGHT (LBS)
WH-325005	113.102:1	375000	175000	500000	9000#

750 HP AT 3600 RPM INPUT AT AGMA 3.0

WH-500002
PLANETARY SPEED REDUCER

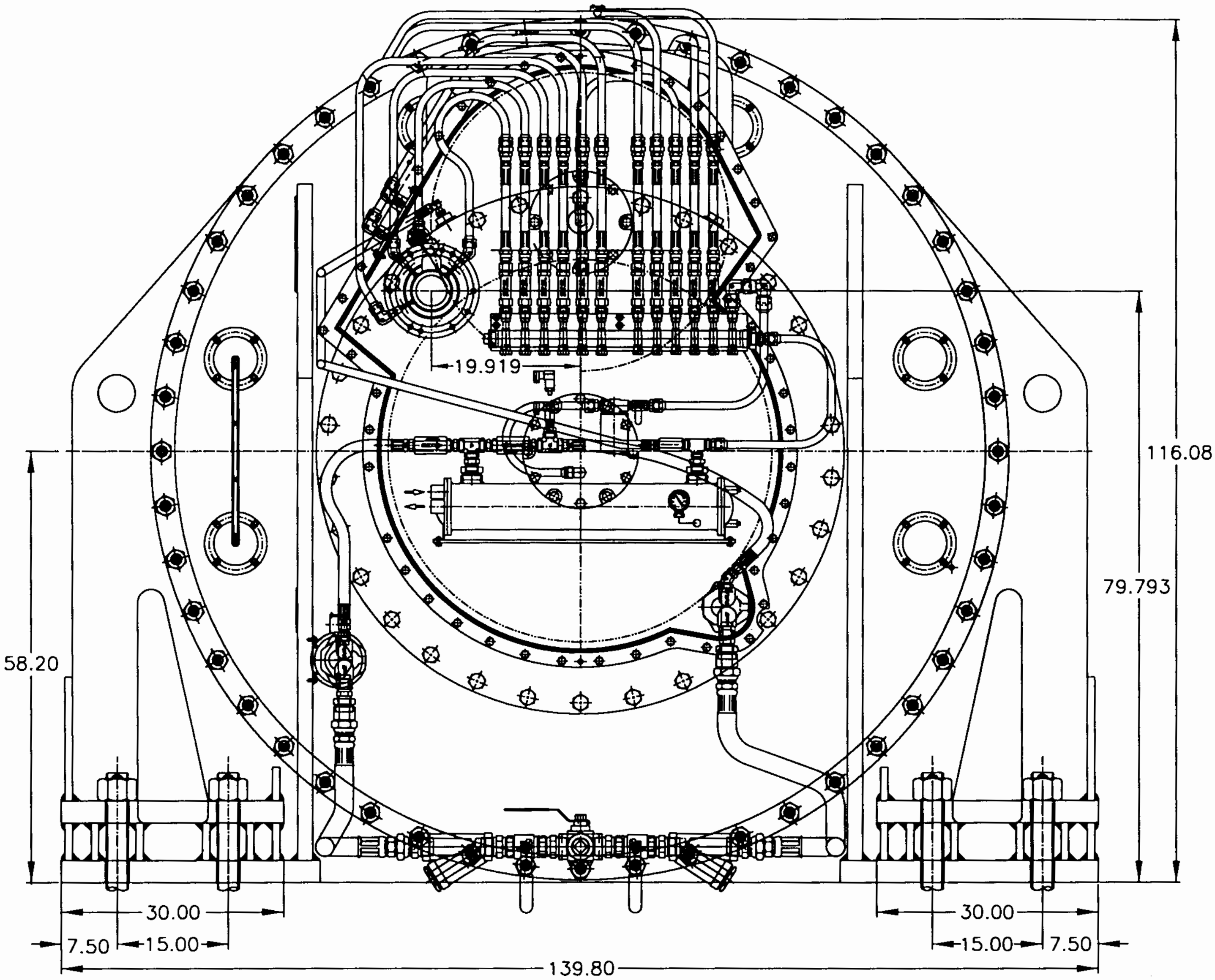


Other ratios available. For more imformation call factory engineering department.

MODEL	RATIO	OUTPUT TORQUE RATING, FT-LB			WEIGHT (LBS)
		AGMA 1	AGMA 2		
		INTERMITTENT	CONTINUOUS	STALL	
WH-500002	178.06:1	500000	250000	750000	11000#

700 HP AT 3600 RPM AT AGMA 2.8

WH-8M
WHEEL HUB



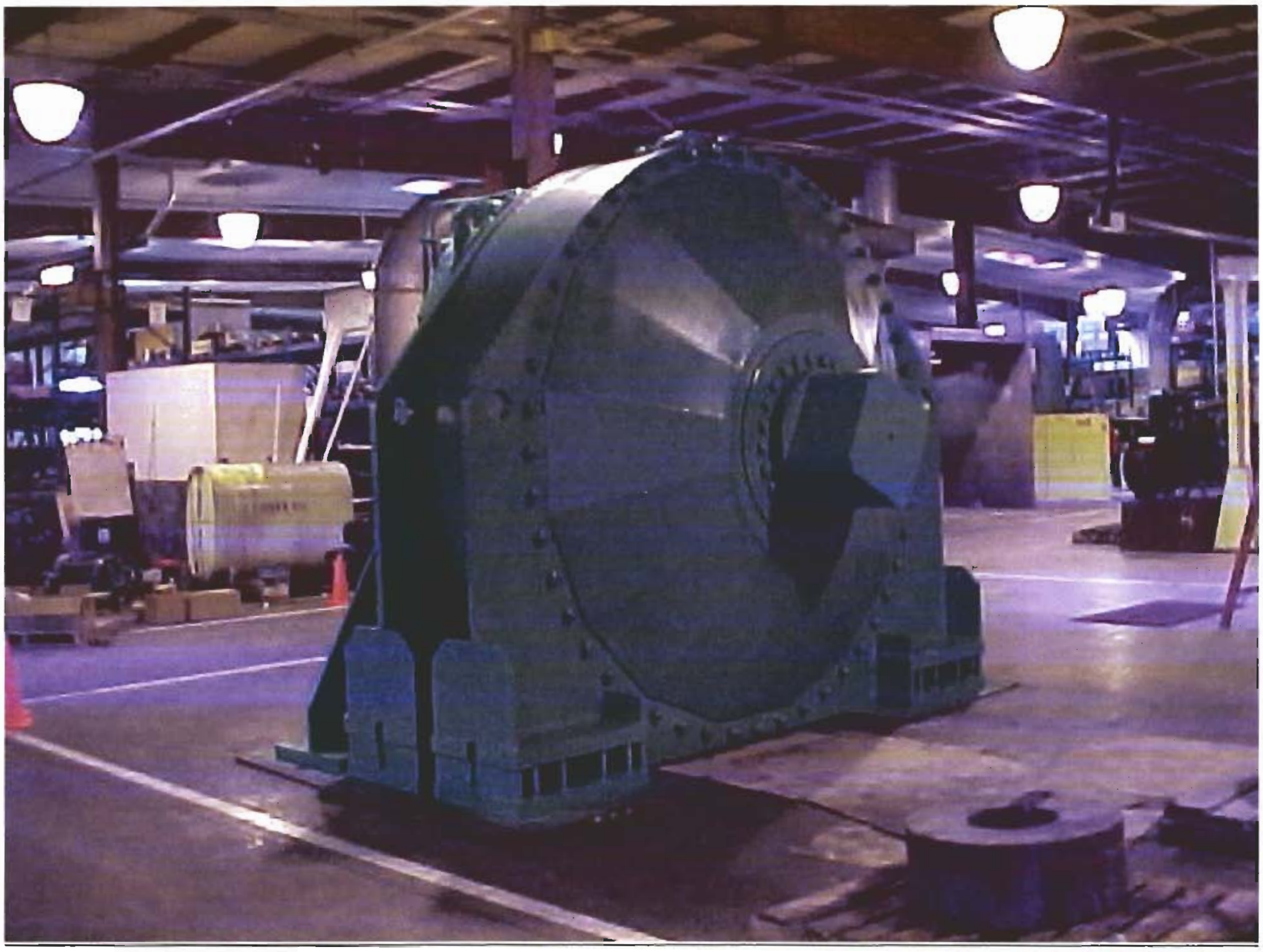
Other ratios available. For more imformation call factory engineering department.

MODEL	RATIO	OUTPUT TORQUE RATING, FT-LB			WEIGHT (LBS)
		AGMA 1	AGMA 2		
		INTERMITTENT	CONTINUOUS	STALL	
WH-8M	1147.3:1	6921600	3460000	10000000	112500#

1425 HP AT 4200 RPM INPUT AT AGMA 3.4

Power Engineering & Manufacturing Ltd.
2635 WCF&N Dr. ♦ P.O. Box 4055 ♦ Waterloo, Iowa 50704
Phone (319) 232-2311 FAX (319) 232-6100
E-Mail info@pemltd.com ♦ http://www.pemltd.com

WH-8M GEAR BOX



INSTALLATION DIMENSIONS	POWER RATINGS
HEIGHT = 116" WIDTH = 139.8" DEPTH = 113.2" BASE = 139.8" X 72"	HORSEPOWER = 1500 TORQUE (OUTPUT) = 2,044,421 FT-LBS RPM = 3.2 RATIO = 1147.3:1 WEIGHT = 112,500 LBS

3.4 AGMA SURFACE DURABILITY FACTOR