Global Powertrain Congress Paper

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Increased Gear Fatigue Life With MEGAGEARS® and UNIMEGAGEARS®.

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Abstract

Power Engineering and Manufacturing, Ltd. (PEM) conducted many years of gear design research to improve the surface fatigue life, the bending fatigue life, as well as to increase the power density.

Introduction

Increased surface and bending fatigue life was necessary because our gear boxes encountered frequent shock loads when cutting trenches through rocky terrain. In addition to that we developed a novel design that adds a high speed flywheel on the input shaft to produce additional kinetic energy when encountering hard rock that tends to stall the cutter. The flywheel enables the machines to increase productivity by 25 to 50 percent on hydraulic drives, see Figure Number 1.

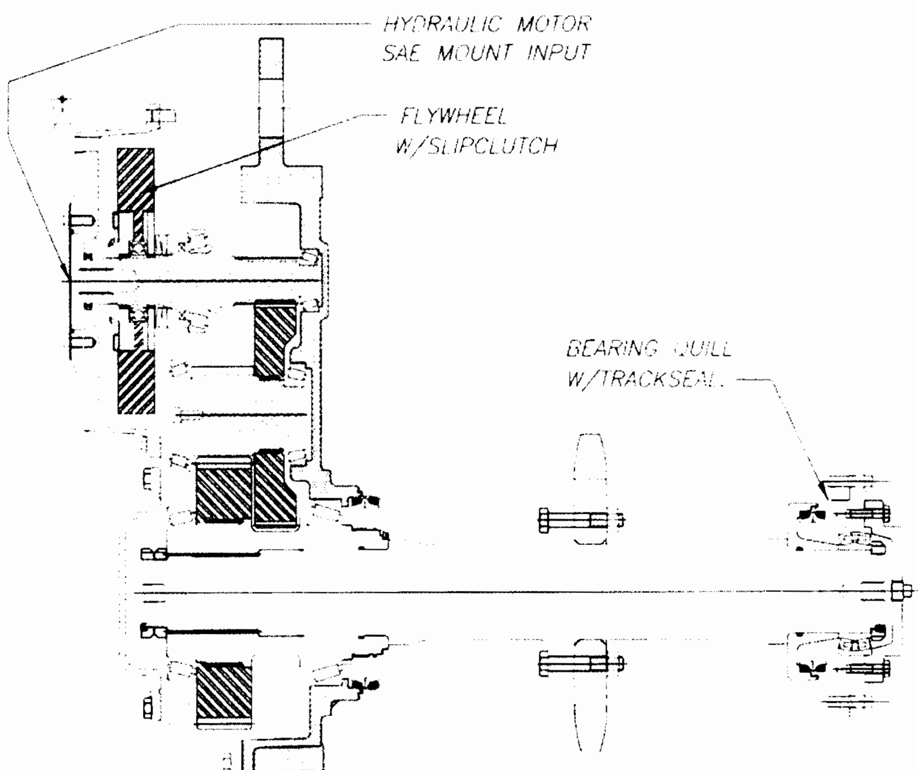
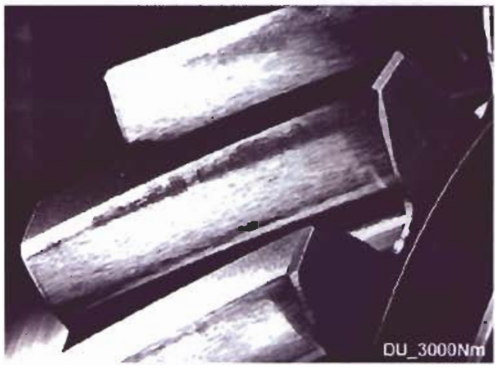


Figure Number 1

At the same time that we conducted research on how to make stronger gear teeth, we also researched how to increase the power density to become more competitive by using less material and less labor.

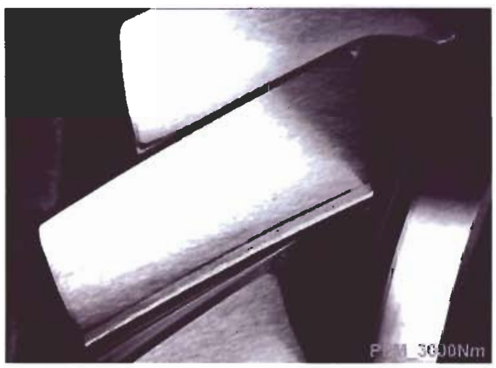
Body

Picture Numbers 1, 2, and 3, show tests results that were conducted for PEM by the Design Unit of The University of New Castle Upon Tyne in England, under the direction of Professor Dieter Hofmann.
Picture Number 1 is of a conventional gear with a 20° pressure angle. It was tested at 1264 horsepower at 3000 rpm for 50 million load cycles. Note there are scoring marks and some micro pitting.



Picture 1

Standard gear 23 teeth, 4.233 NDP, 20° PA, 30° HA **1264 HP** at 3000 RPM, 50 million load cycles SCS@PCR=231,000 PSI, 6.753 O.D.



Picture 2

MEGAGEAR® 19 teeth, 4 NDP, 35° PA, 30° HA **1264 HP** at 3000 RPM, 50 million load cycles SCS@SAP=186,000 PSI, 6.813 O.D.

Picture 3

MEGAGEAR® 19 teeth, 4 NDP, 35° PA, 30° HA **1686 HP** at 3000 RPM, 50 million load cycles SCS@SAP=215,000 PSI, 6.813 O.D.

Picture Number 2 is of a MEGAGEAR® tested under identical conditions as the conventional gear in Picture Number 1. Note that there is some polish near the root diameter at the start of active profile, but there are no score marks or pits, and the grinding marks are intact. The same loads that caused damage on the conventional gear did not cause any damage to the MEGAGEAR®.

Picture Number 3 shows a MEGAGEAR® that was tested at 1686 horsepower also at 3000 rpm for 50 million load cycles. At this 33 percent greater horsepower it can be seen that score marks and micro pits are present, similar to Picture Number 1. This proves that the Power Engineering and Manufacturing, Ltd. MEGAGEARS® are able to carry 33 percent more horsepower for the same surface fatigue as the conventional gear. The University Gear Testing Laboratory is a highly qualified laboratory. The manufacturing and testing conditions between the different production lots were treated identically. The gear material was from the same melt, provided by The Timken Company. The University Laboratory heat treated, ground, and tested the gears as identically as possible. For that reason the test results are very reliable.

The ability of increasing the amount of horsepower in the same amount of metal is possible because the areas of contact between the gear teeth at the point of load transfer is larger at the MEGAGEARS® teeth and the UNIMEGAGEAR® teeth than at the conventional gear teeth, as seen on Figure Number 5.
The following 4 tables show improved features of MEGAGEARS® and UNIMEGAGEARS® over conventional gears.

Table Number 1
Power Density

Type of Gear	Horse-power	Surface Compressive Stress (psi)	Percent Horsepower	Max Bending Stress psi	Max Tooth Deflection in	Lube Film Thickness Micro inches	Max Oil Film Temp Rise (°F)
Conventional	1264	231,000	100	42,300	.0047	109	400
MEGAGEAR®	1946	231,000	154	30,600	.0037	175	286
UNIMEGAGEAR®	2130	231,000	168	37,800	.0029	190	290

Table Number 1 shows the power density of the MEGAGEARS® and UNIMEGAGEARS® in relation to conventional gears. By keeping the same surface compressive stress at the profile contact ratio of 1.0 to have equal surface fatigue life, the calculations show that the MEGAGEAR® can carry an additional 54 percent more horsepower with the same amount of metal and the UNIMEGAGEAR® can carry an additional 68 percent more horsepower with the same amount of metal when compared to a conventional gear.

The MEGAGEARS® can operate *equally well* in both directions of rotations. The UNIMEGAGEARS® are suitable for applications where the load is predominantly in one direction of rotation such as in hoisting. The desired fatigue life is calculated for the high pressure angle flank. Usually the low pressure angle flank will have between 25 and 50 percent as much fatigue life as the high pressure angle flank.

It is not possible to establish for all gears, a single fixed percentage of fatigue life, power density, or reduction in weight because each pair of gears may have different diametral pitches, different length addendums, and various modifications such as lower profile contact ratio, tip relief, etc.

More importantly at Power Engineering and Manufacturing, Ltd., we calculate the fatigue life of the gear with the lowest number of teeth at the start of active profile or at the profile contact ratio of 1.0. This is where the maximum surface compressive stress and consequently the lowest surface fatigue life occurs. This controls life of the gear box. We developed the MEGAGEAR® and UNIMEGAGEAR® technology to increase the fatigue life of the gears and gear boxes. This makes the MEGAGEARS® and UNIMEGAGEARS® very suitable for applications where there are high impact loads.

Table Number 2
Required Weight Increase of Conventional Gears to Match the Horsepower of the MEGAGEARS® and UNIMEGAGEARS®

Equivalent Gear	Horse-power	Surface Compressive Stress at PCR=1.0 psi	Pressure Angle	Center Distance in	Outside Diameter in	Rough Weight lb	Percent Weight lb	Max Bending Stress psi
Conventional	1264	231,000	20°	6.299	6.753	23	100	42,300
MEGAGEAR®	1946	231,000	20°	7.677	8.188	33	143	52,500
UNIMEGAGEAR®	2130	231,000	20°	7.979	8.486	36	156	53,900

Table Number 2 shows the required increases in weight that is necessary for the conventional 20° pressure angle gears to carry the same amount of horsepower as the MEGAGEARS® and UNIMEGAGEARS®, as shown on Table Number 1. For the conventional gear to carry the same 1946 horsepower as the MEGAGEARS® do in place of 1264 horsepower, the gear has to have 43 percent more metal. Likewise, for the conventional gear to carry 2130 horsepower as for UNIMEGAGEAR® does in place of 1264 horsepower, the gear has to have an additional 56 percent more metal.

The reduction in weight may also be achieved when we overhaul a gear box that has conventional gears and we replace them with MEGAGEARS® or UNIMEGAGEARS®. When we replace a conventional gear with MEGAGEAR® we can save 10 to 20 percent weight and provide an equal or longer surface fatigue life. When we replace the conventional gear with UNIMEGAGEAR® we can save 15 to 25 percent weight. Here also the surface fatigue life is equal or greater. Note that the bending stress of the MEGAGEAR® and UNIMEGAGEAR® is about half as much at that of the conventional gear at equal horsepower. The bending fatigue life is to the sixth power of the bending stress ratio, see Equation Number 1.

Equation Number 1 $$BendingFatigueLife = \left(\frac{BendingStress(A)}{BendingStress(B)} \right)^6$$

MEGAGEARS® and UNIMEGAGEARS® bending fatigue life $=2^6=64$

For practical considerations it can be assumed that MEGAGEARS® and UNIMEGAGEARS® will not fatigue in bending, see Table Number 3.

Table Number 3

Weight Reduction with MEGAGEARS® and UNIMEGAGEARS® When Replacing Conventional Gears

Type of Gear	Weight Reduction Percent	Rough Weight lb	Center Distance in.	Outside Diameter in.	Max Bending Stress psi	Horse-power	Oil Film	Temp Rise	Surface Compressive Stress at PCR=1
Conventional	0	23.0	6.229	6.753	42,300	1264	109	400	231,000
MEGAGEAR®	17	19.1	5.167	5.696	20,500	1264	147	286	231,000
UNIMEGAGEAR®	22	17.9	4.970	5.493	20,700	1264	157	268	231,000

Table Number 4

Surface and Bending Fatigue Life at Constant Horsepower

Type of Gear	Surface Compressive Stress at PCR=1.0	Life Percent	Max Tooth Deflection in	Oil Film Thickness Micro Inches	Max Oil Temp Rise (°F)	Bending Stress (psi)	Bending Fatigue Life %
Conventional	231,000	100	.0036	109	400	42,300	100
MEGAGEAR®	186,000	424	.0026	188	207	19,900	9,200
UNIMEGAGEAR®	178,000	586	.0024	206	196	22,600	4,300

Table Number 4 shows the gear surface fatigue life increase when conventional gears are being replaced with the MEGAGEARS® or UNIMEGAGEARS® and the horsepower remains constant. It can be seen that when conventional gears are being replaced with MEGAGEARS the surface fatigue life increases to 424 percent. Likewise when conventional gears are being replaced with UNIMEGAGEARS®, the surface fatigue life increases to 586 percent.

PEM continuously overhauls other brands of gear boxes. When MEGAGEARS® and UNIMEGAGEARS® are replacing conventional gears, the surface and bending fatigue lives are increased by threefold or more.

Equation Number 2 $$SurfaceFatigueLife = \left(\frac{SCS_1}{SCS_2}\right)^{6.666} = 1.21^{6.666} = 2$$

(Terms are defined at the end of the paper)

Equation Number 2 shows that if it is possible to reduce the surface compressive stress by 21 percent when replacing conventional gears with MEGAGEARS® and UNIMEGAGEARS®, the surface fatigue life doubles. On the majority of the gear boxes that we overhaul the reduction in the surface compressive stress is reduced by more than 21 percent the surface fatigue life is doubled. At the same time the bending fatigue failures are completely eliminated.

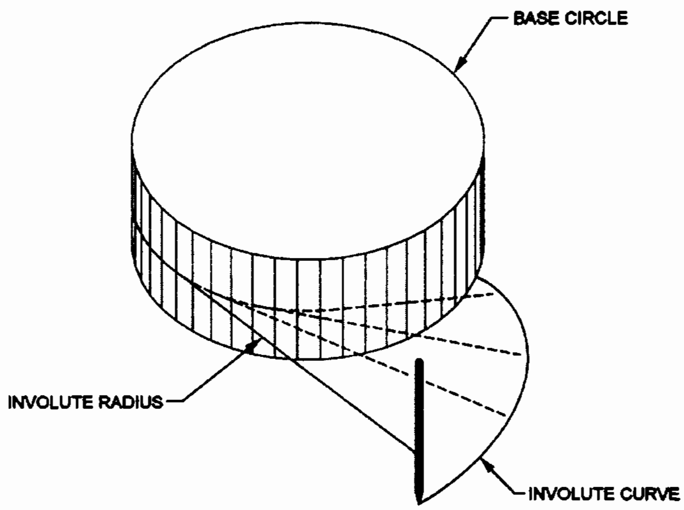
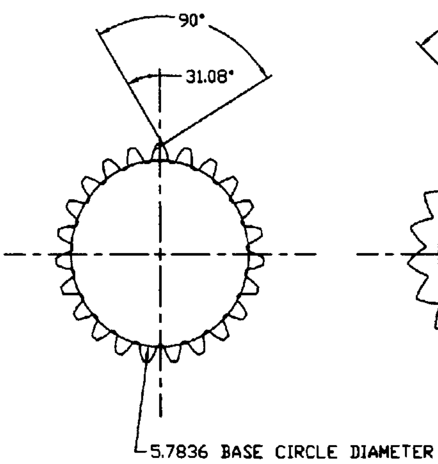
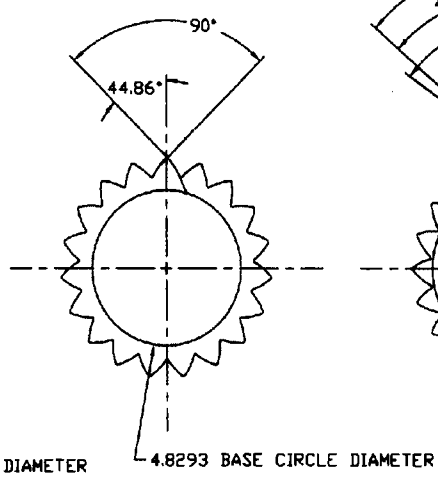


Figure Number 2 Development of the Involute Profile

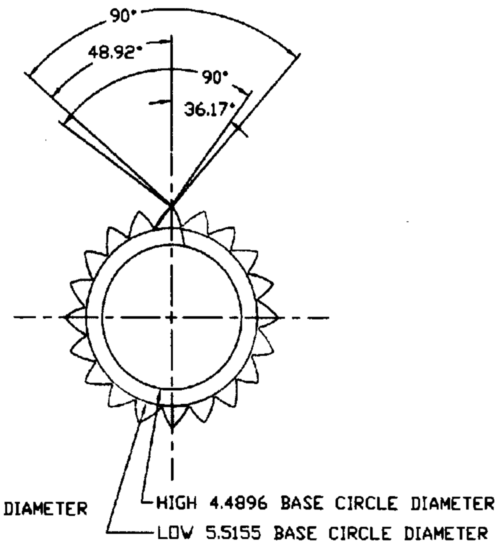
At this point it is appropriate to define the gear tooth profile. Figure Number 2 shows how the involute curve is developed. If a pencil is tied to a string that is fixed to a drum whose diameter is the same as the base circle of a gear, when unwinding the string the pencil scribes a line that is the involute curve. Austrian scientist Leonhard Euler (1707-1783) derived the involute mathematics that results in constant angular velocity. Any section of the involute curve can be used to form the gear tooth profile. The further away the gear tooth is from the base circle diameter the bigger the radius of curvature is. This yields a larger area of contact between the teeth, thus reducing the surface compressive stress and increasing the surface fatigue life.



CONVENTIONAL GEAR
FIGURE 3



MEGA GEAR
FIGURE 4



UNIMEGA GEAR
FIGURE 5

Figure Numbers 3, 4, and 5, show the gears as calculated for Table Number 1. Note that Figure Number 3 shows a 20° pressure angle gear, which we refer to as a conventional gear. This type of gear is being used almost exclusively all over the world. Note that the base circle diameter is at the root diameter. This results in a short involute radius at the point of profile contact ratio of 1.0 or at the start of active profile.

Figure Number 4 is of a MEGAGEAR® used in Table Number 1. Note that the base circle diameter of the MEGAGEAR® is smaller than that of the conventional gear. This produces a larger involute radius at the profile contact ratio of 1.0, or at the start of active profile.

Equation Number 3: Surface Compressive Stress $$SCS=\sqrt{\frac{\mathrm{E}}{\pi\cdot\left(1+\nu^2\right)}\cdot\frac{T}{BR}\cdot\frac{\frac{1}{r_1}+\frac{1}{r_2}}{FW\cdot PCR\cdot 2\cdot VF}}$$

Equation 3 shows that when calculating the surface compressive stress, the reciprocals of the involute radii of both gears in mesh at the point of load transfer are being used.

Equation Number 3 shows the surface compressive stress as is calculated by the Hertz equation where the involute radii at load transfer point is being taken to construct 2 cylinders in axial contact that are being squeezed with a force that is proportional to the horsepower.

Equation Number 3 shows that the axial length of the cylinders is equal to the face axial length of gear mesh as designated by the letters FW that stand for force width. It can be seen from Equation Number 3 that if a gear has undercut, and the start of active profile is at the beginning of the undercut, the involute radius is zero because the base circle diameter is at the beginning of the undercut. If we substitute zero for the involute radius in the denominator of the reciprocal of 1/R in Equation Number 3, it can be seen that this quantity will become an infinite number. The surface compressive stress becomes infinite when the contact point is at the base circle diameter and the surface fatigue life consequently becomes zero.

Figure Number 5 shows a UNIMEGAGEAR®. Here the high load carrying flank is the one that has a higher pressure angle and it is constructed from the smaller base circle. Because the base circle is smaller the involute radius at the profile contact ratio of 1.0 or at the start of active profile is longer than the involute radii of the conventional gear or the MEGAGEAR®.

The larger the involute radii of the cylinders are, the larger the contact area between the gear teeth at load contact is. The lower the surface compressive stress results in a larger surface fatigue life.

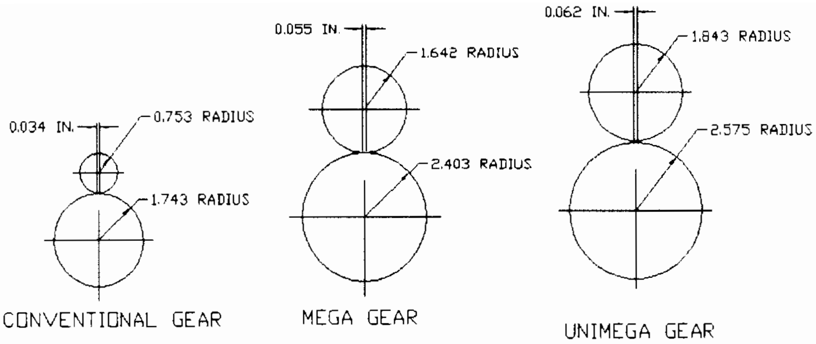


FIG. 6

INVOLUTE RADII AND CONTACT WIDTH AT THE START OF ACTIVE PROFILE.

Figure Number 6 shows the circle that are used as cylinders to calculate the surface compressive stress based on the involute radii at the load transfer point of profile contact ratio of 1.0 or start of active profile. Note that because the involute radii on the MEGAGEARS® and UNIMEGAGEARS® are larger than on the conventional gear, the contact width at load transfer is also larger. The contact width when multiplied by the axial mesh length, provide a larger area for MEGAGEARS® and UNIMEGAGEARS® than at the conventional gears. The same force distributed over a larger area reduces the surface compressive stress which results in a larger surface fatigue life.

The bending fatigue lives of MEGAGEARS® and UNIMEGAGEARS® are also greater than those of the conventional gears as can be seen in Table Number 4. The increase in the bending fatigue life is to the sixth power of the bending stress reduction ratio. It is important also to note that in conventional gears, the maximum bending stress occurs near the root diameter. When a conventional gear tooth fatigues, it breaks off the gear near the root diameter. This can cause high impact loads due to skipping a tooth and can break even more teeth. However, the bending stresses in MEGAGEARS® and UNIMEGAGEARS® are so low that bending fatigues are totally eliminated. This is one mode of failure that does not exist with MEGAGEARS® or UNIMEGAGEARS®.

The most damaging force can be an impact force at the tip of the tooth as shown by the perpendicular vectors at the tip of the tooth.

Figure Number 3 shows that on the conventional gear tooth that the vector force has a certain amount of metal to absorb the energy. A very high force can actually shear the tip of the tooth. The MEGAGEAR® and UNIMEGAGEAR® shown on Figure Numbers 4 and 5 have a larger amount of metal to absorb the energy of a very high force because the vector force of the MEGAGEARS® and UNIMEGAGEARS® flows throughout a larger cross sectional area of the teeth than that of the conventional gear tooth.

The vector force at the tip of the tooth for the MEGAGEAR® and UNIMEGAGEAR® is pointed inside the tooth material because of the higher pressure angle at the tip of the tooth. The radial force member of the vector force imposes a compression stress on the tooth material. This helps to reduce the tooth tensile stress, thus further increasing the bending fatigue life.

Other advantages of the MEGAGEARS® and UNIMEGAGEARS®, is that the oil film thicknesses between the gear teeth when under pressure at load transfer is greater than the oil film of the conventional and is helping improve the gear tooth durability.

Table Numbers 1 and 4 show that the lube oil film thickness between the teeth are also greater at the MEGAGEARS® and UNIMEGAGEARS® than at the conventional gears. Having a larger area of contact between the teeth helps to increase the oil film thickness. This is a good feature because it reduces the frequency of mechanical load transfer that may have concentrated loads that can bring a premature fatigue failure. A hydrostatic load transfer is desirable because it allows to achieve the longest surface fatigue life possible due to a more uniform load distribution.

Lube oil film temperature at the sliding point under load is very important because the oil viscosity is such that the oil film thickness becomes thinner, the higher the oil temperature is. About one percent of the horsepower is gear mesh efficiency loss. This entire amount of lost horsepower is transformed into heat that is absorbed by the lubricating oil. Note that Picture Number 2 show that the MEGAGEAR® teeth retained the grinding marks without wear. This is evidence of a good oil film that provides hydrostatic load transfer and more importantly that the design was done for surface fatigue life and not for wear.

Conclusion

The features of the MEGAGEARS® and UNIMEGAGEARS® developed by Power Engineering and Manufacturing, Ltd. as illustrated above have a higher power density, a higher surface fatigue life, and the bending fatigue failures due to bending stress are totally eliminated.

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DEFINITIONS:

PEM: Power Engineering and Manufacturing, Ltd.

SYMBOL	DESCRIPTION	UNITS
π	Pi = 3.14159	
ν	Poisson's Ratio = 0.30 for steel	
E	Modulus of elasticity = 30*10^6 for steel	psi
BR	Base Radius	in
FW	Face Width	in
LLOA	Length of the line of action	in

PCR	Profile contact ratio	.
r1	Radius of curvature	in
r2	Radius of curvature	in
SCS	Surface compressive stress	psi
T	Torque	in-lb
VF	Velocity Factor	