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Low Cycle and Static Bending Strength of Carburized and High Hardness Through Hardened Gear Teeth

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American Gear Manufacturers Association

TECHNICAL PAPER

Low Cycle Fatigue and Static Bending Strength of Carburized and High Hardness Through Hardened Gear Teeth

W. Pizzichil, Philadelphia Gear Corp

ABSTRACT:

A summary of the testing methods employed and the results generated for unidirectional and reverse bending tests of very coarse and medium pitch gear teeth is presented. Actual measured stresses were compared with FEM theoretical stresses and AGMA stress numbers.

The purpose of this testing was to evaluate which type hardening method would yield a gear tooth that could carry the highest load without catastrophic breakage failure in a single, or very low cyclic load application. This testing simulated the output pinion and a planet gear for a jack–up gear drive used on oil drilling platforms. There were three separate tests conducted over a period of time.

The first test was performed statically on an actual 12 inch circular pitch pinion loaded against a rack. A second static test utilized a half scale model of a 12 CP gear tooth fillet loaded to failure in a single cycle and also cycled to failure at extremely high loads. The third test was a reverse bending test simulating a planet of a planetary gear used in the final reduction stage.

The last two tests were accomplished simultaneously for both through hardened and carburized test pieces. The results demonstrated the benefits of the relatively ductile through hardened gears and their ability to locally yield and distribute stresses without catastrophically failing in this type of application.

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LOW CYCLE FATIGUE AND STATIC BENDING STRENGTH OF CARBURIZED AND HIGH HARDNESS THROUGH HARDENED GEAR TEETH

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INTRODUCTION:

The particular application for the gear drives that were tested is to elevate off shore drilling rig platforms (jack-ups) above sea level once they have been floated to the desired drilling location, see Figure 1. The drives are fixed to the deck of the vessel. They have output pinions which mesh with racks that are attached to three moveable triangular legs at each corner of the vessel. The legs are lowered to the ocean floor and the platform is raised to the desired height.

COMMENT

This paper presents the results of actual testing performed on large scale, costly test pieces. It is recognized that there are statistical variations associated with this type of testing that make absolute conclusions unrealistic. There is no attempt made to characterize the metallurgical phenomena that governs the behavior.

LITERATURE SURVEY

The literature reveals that work has been carried out evaluating the performance of through hardened racks for jack-ups by Honda (1,2,3). There also is extensive work available relating to the failure of case hardened gears. References (4,5,6,7,8) represent a sampling of this literature. The testing presented in this paper was performed to gather data on very large carburized gears, where very little testing has been performed.

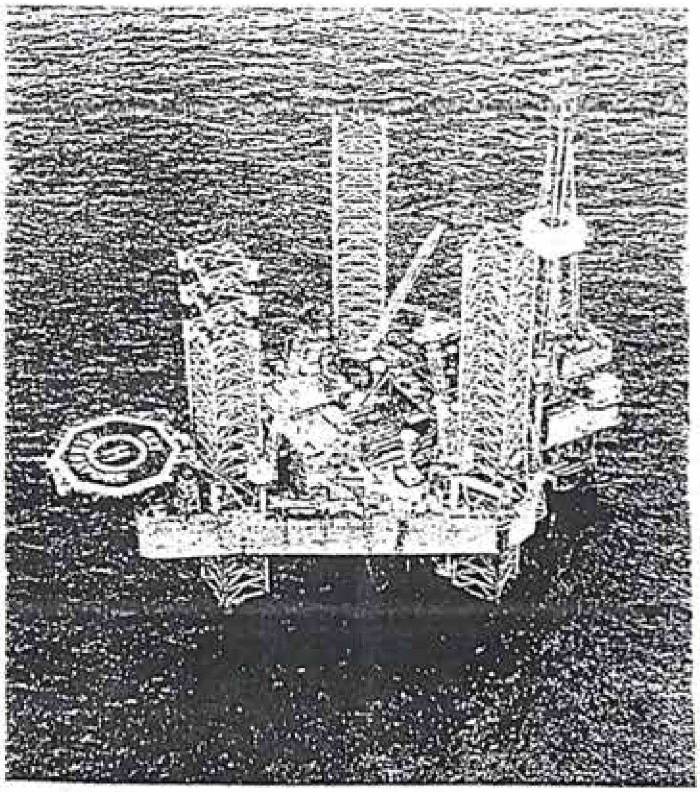

FIGURE 1. OFFSHORE DRILLING RIG PLATFORM (JACK-UPS)

TEST 1

SINGLE CYCLE BENDING FAILURE TEST (12 CP Actual Pinion).

TEST OBJECTIVES AND METHOD

The main objective of this testing was to determine the single cycle, ultimate fracture strength of a 7 tooth, 12 CP, 8.5 inch face width, carburized, spur pinion, loaded at the highest point of single tooth contact.

This test was necessary to determine the maximum holding capability of the pinion supporting the jack-up rig. A second objective was to evaluate the effects of load sharing in the narrow band of two tooth contacts with the mating rack. Tooth pair load sharing potential is evaluated by the transverse contact ratio parameter. The nominal, transverse contact ratio is approximately 1.1. The final objective was to determine if the mating, through hardened, rack had equal or better ultimate fracture capacity compared to the carburized pinion, when loaded at it's highest point of single tooth contact.

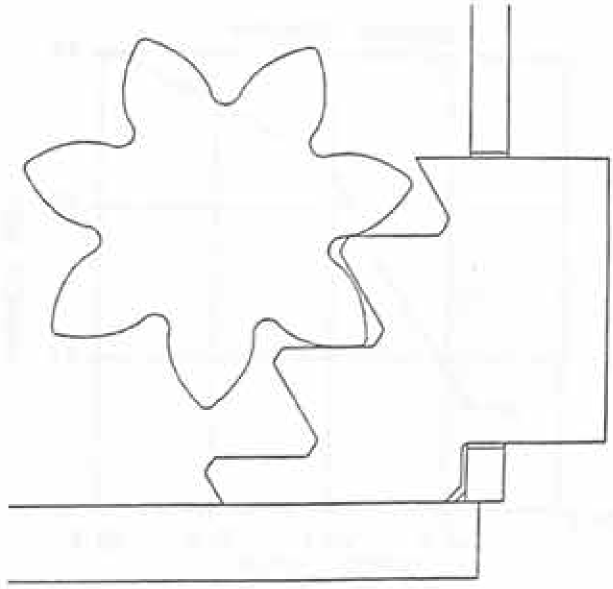

FIGURE 3

PINION TEST

Figure 2 shows the test stand that was designed to support the jack-up gear drive with it's outboard, 7 tooth pinion in a similar way that the jack tower supports the drive in the jack up rig .

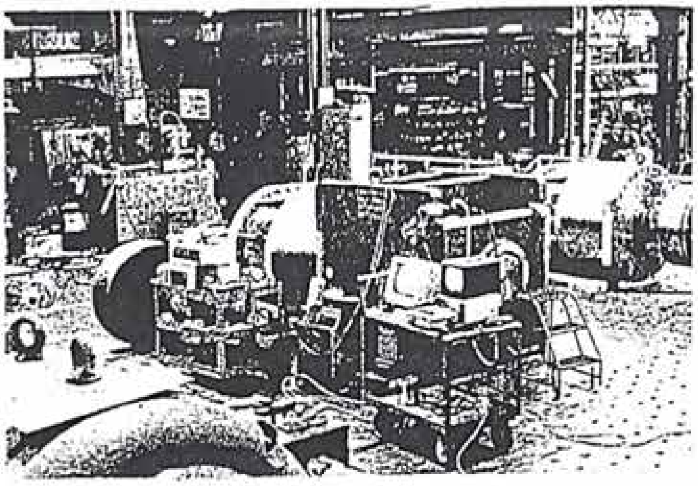

FIGURE 2. JACK-UP GEAR DRIVE TEST STAND

An actual segment of rack was installed in the test stand to mesh with the pinion, see Figure 3. This rack segment reacted the statically applied tangential load and loaded the test stand housing in a closed loop configuration and self contained the extremely large tangential force of up to 2,100,000 lbs.

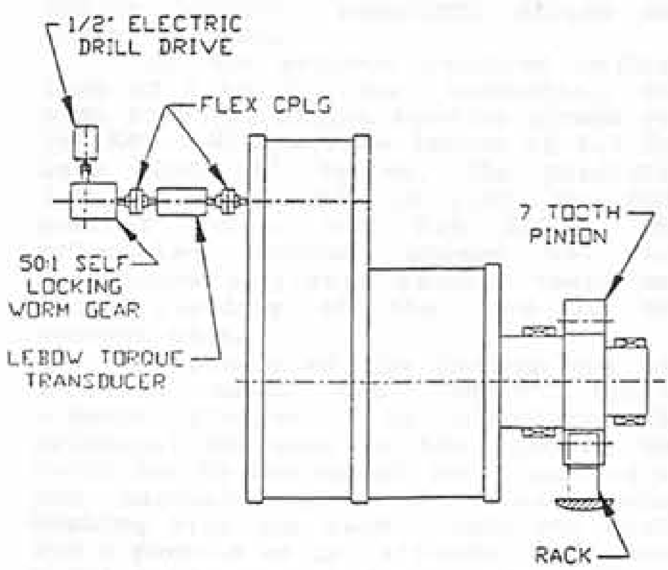

FIGURE 4

Figure 4 shows a schematic of the test drive train. The input shaft of the test gearbox was coupled to a Lebow Torque Transducer which was driven by a 50:1 ratio, self locking worm gear drive. The worm drive input was then rotated by a large 1/2 inch drive industrial drill for system loading. An input torque of 350 lb-ft at the jack-up gearbox generated the 2,000,000 lb. tangential output force via the five parallel shaft gear meshes and one planetary stage for a total 1765:1 gear drive ratio. The output force takes into account the frictional mesh and bearing losses. The system winds up like a large torsional spring requiring 38 full turns of the input shaft once all backlash is removed to obtain 1,600,000 lbs. tangential force on the 7 tooth pinion. The self locking worm drive held the input torque at any desired level and prevented unwinding of the geartrain. The Lebow Torque Cell was connected to a digital display calibrated to read input torque directly in foot pounds

The 7 tooth pinion and racks were instrumented with strain gauges at, and very near the calculated critical stress area of the tooth root fillets. Four gauges were laid at each edge of the face width so that any maldistribution of load across the face width, due to misalignment, could be assessed, see Figure 5. The strain was measured with a Vishay Strain Recording Indicator.

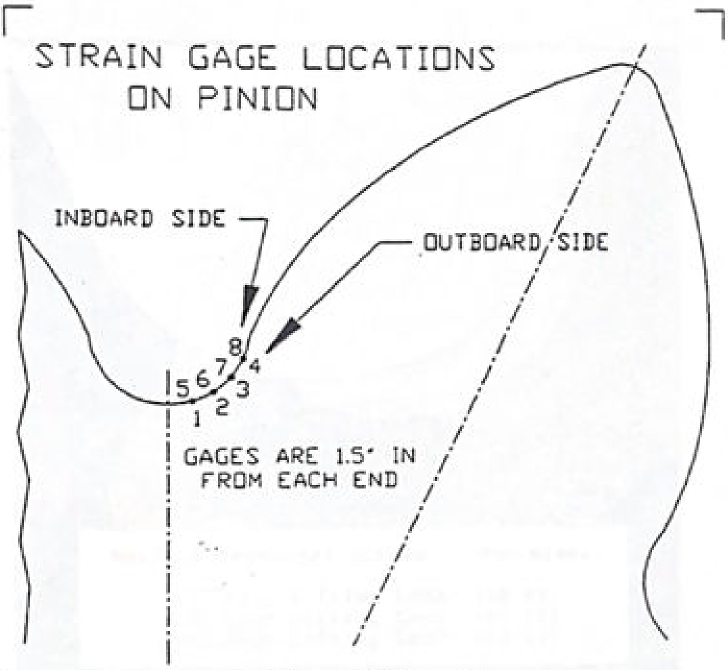

FIGURE 5

PINION MATERIAL, MANUFACTURING PROCESS AND HEAT TREATMENT

The 7 tooth output pinion was made of 17CrNiMo6, a popular European carburizing steel, which has high hardenability. The manufacturing process for the pinion consisted of flame cutting the basic profile with an allowance for finishing. The fillets were then bored and the final profile was shaper cut. The pinion was then carburized, hardened and tempered. The actual hardness traverse from a tooth is shown in Figure 6.

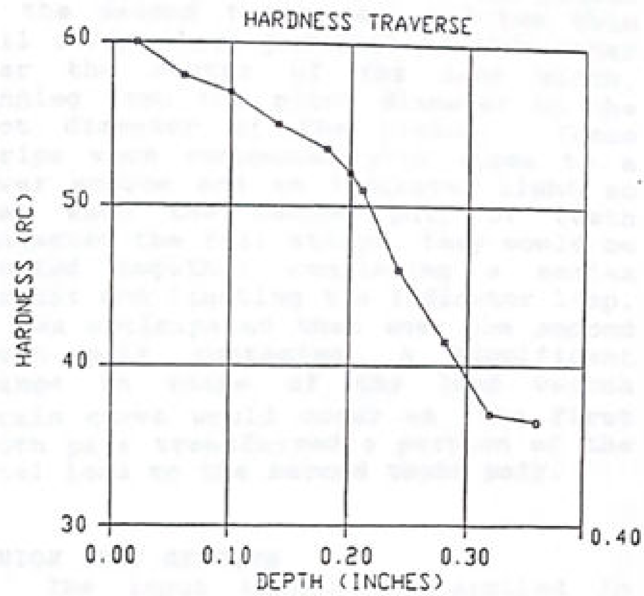

FIGURE 6

PINION LOADING, PREDICTED RATING AND STRESS ANALYSIS

At the maximum required holding load of 1,600,000 lbs. tangential, the AGMA 2001 [9] pinion bending stress was 145 KSI. With a life factor of 2.7 for less than 10^3 cycles, the predicted Service Factor (SF) is 1.21. The AGMA contact stress was 528 KSI. The calculated contact stress was not considered realistic because there was local yielding of the rack in the contact zone.

The pinion of the jack-up rig was analyzed using the "ANSYS" finite element program to calculate the principal stresses at the root of the tooth due to the normal force applied at the highest point of contact when meshing with the rack. Only one tooth and a portion of the adjacent tooth were modeled because of the minimal effects of the other pinion teeth, see Figure 7. The pinion was modeled using the STIF42 - 2D isoparametric solid elements from the ANSYS library. This element is used for two dimensional models of solid structures and can be used either as a biaxial plane element or an axisymmetric ring element. The element is defined by four nodal points and has two degrees of freedom at each node. A two dimensional model with a unit depth was considered adequate as there would be no out of plane effects in the pinion with it's 8.5 inch face width. Figure 8 presents the calculated principal stress at the root of the pinion tooth.

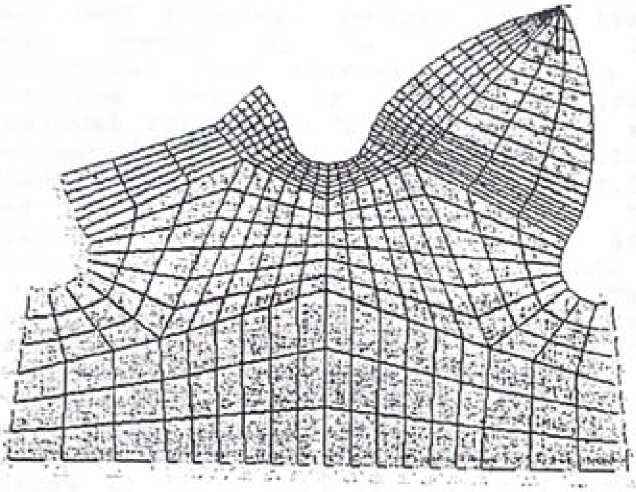

FIGURE 7 - BULL PINION MODEL AND BOUNDRY CONDITIONS

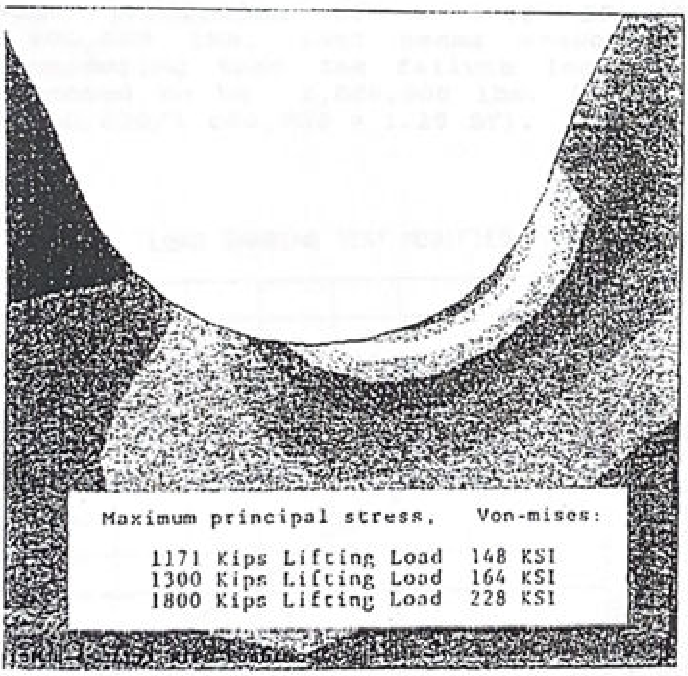

FIGURE 8 - CALCULATED PRINCIPAL STRESS AT THE ROOT OF THE PINION TOOTH

TRANSVERSE LOAD SHARING

When one pinion tooth is loaded at or near the highest point of single tooth contact, a second tooth pair is very close to contact. The test was run with the pinion and rack indexed relative to each other in such a way that a second pair of teeth had a gap measured between contacting surfaces of .063 in. on one edge of the tooth and .049 in. on the other edge. The pinion of the second tooth pair had two thin foil strips laid parallel to each other near the center of the face width, running from the pitch diameter to the root diameter of the pinion. These strips were connected with wires to a power source and an indicator light so that when the second pair of teeth contacted the foil strips, they would be shorted together completing a series circuit and lighting the indicator lamp. It was anticipated that when the second tooth pair contacted, a significant change in shape of the load versus strain curve would occur as the first tooth pair transferred a portion of the total load to the second tooth pair.

PINION TEST RESULTS

The input torque was applied in increments and complete sets of all available data were recorded. Strain Gauges 2 and 7 , which were at the peak stress points (one on each edge of the pinion tooth), are plotted on Figure 9 against output tangential force. The indicator lamp lit up at 1,000,000 lb. load indicating a second tooth pair contact. Gauge 2 showed a reduction in the slope of strain versus applied load beyond the 1,000,000 lb. load level. Gauge 7 showed less change in slope. This lower slope verified a sharing of load between 2 tooth pairs. The reduction in load of the first pair, due to load sharing, was not as great as was expected. The strain curves continued in a nearly linear fashion up to 1,800,000 lbs. and then showed a slight increase in slope. The load was incremented up to a tangential load of 2,100,000 lbs. and held for 15 minutes at this point. As the highest applied load was approached, there was a cracking noise much like the sound of ice cracking. When the load was released, there was a high amount of residual tensile strain showing on the strain gauges. Gauge 2 showed (1143 microstrain) and gauge 7 showed (1557 microstrain). It was hypothesized that yielding of the core occurred which left the case in residual tension.

A second test was run on the same tooth, similar to the one above, except that the initial gap of the second tooth pair was set at .074 in. and .097 in. The load was incremented again and several loud cracking sounds were heard while holding at 1,650,000 lbs. The strain in gauges 2 and 7 went to zero because the cracks occurred right through the gauge length. Within a few seconds the tooth failed catastrophically. The sound was much like a loud gun shot. Sparks flew out of the test fixture. This violent failure was different from what was reported in

Ref.8. Moores's testing indicated a some what gentle parting after the initial cracking. This discrepancy might be a function of the size difference between the test samples. Several other teeth were loaded up to 1,800,000 lb. tangential load several times with no cracking heard or any significant residual strain in the gauges. It was estimated that the low cycle failure load was likely to be around 2,000,000 lbs. based on the testing done. The maximum strain recorded of 8693 microstrain at this load corresponds to a tensile stress of 261 KSI. This stress is very close to the estimated tensile strength of 275 KSI for the carburized case material.

The FEM results at 1,175,000 lbs. load (just after second tooth pair contact) gave 148 KSI. The measured stress at this load was 143 KSI, which agreed reasonably well with the FEM calculations. At a tangential load of 1,800,000 lbs., the FEM results gave a stress of 228 KSI. The measured result at that load was 218 KSI. This difference can be explained by the load sharing effect among the two teeth. The AGMA prediction of a 1.21 SF at 1,600,000 lbs. load seems reasonable considering that the failure load was expected to be 2,000,000 lbs. (i.e., 2,000,000/1,600,000 = 1.25 SF).

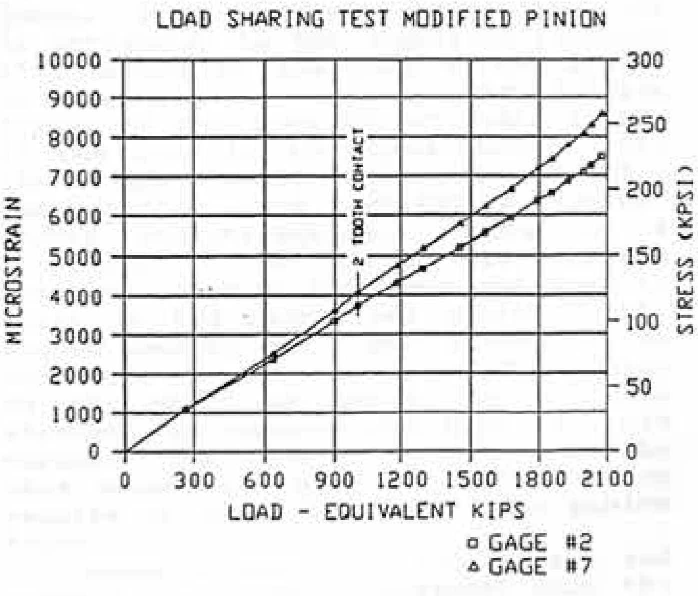

FIGURE 9

RACK TEST

In this test, the rack and the pinion were again indexed. The highest point of single tooth rack contact was obtained and this point meshed with the dedendum of the pinion as shown in Figure 10. The test was conducted as

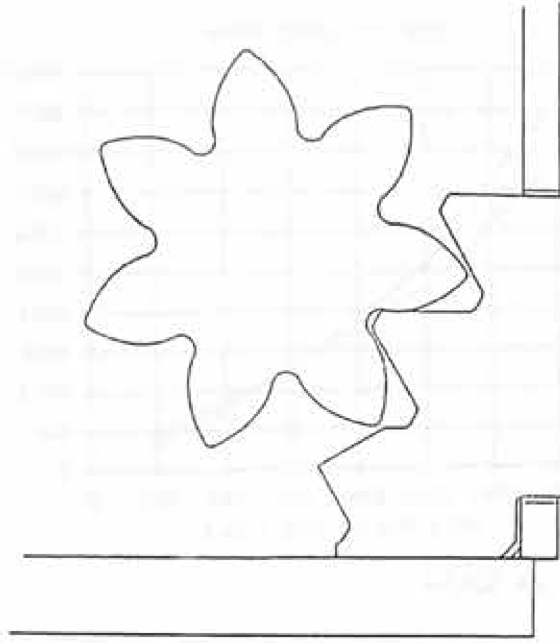

FIGURE 10

described above for the pinion. The rack segment tooth root fillet was instrumented with four strain gauges as shown in Figure 11. The fourth gauge on each side of the rack was laid on the end face of the rack rather than the root fillet. Gauges 2 & 6 were at the calculated maximum critical stress area.

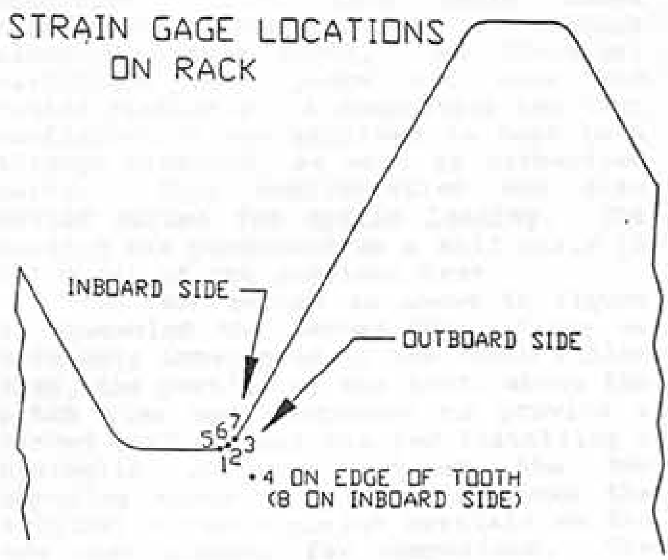

FIGURE 11

RACK MATERIAL HEAT TREATMENT AND MANUFACTURING PROCESS

The rack material was not known as it was customer supplied. The tooth profile was flame cut only. The hardness readings of the rack averaged 350 BHN on the tooth flank surface and 280 BHN two inches below the flank surface, measured on the side of the rack.

RACK LOADING, PREDICTED RATING AND STRESS ANALYSIS

The AGMA 2001 bending and contact stress numbers and the FEM bending stress at 1,600,000 lbs. tangential load were:

AGMA Bending Stress = 158 KSI
AGMA Contact Stress = 528 KSI
FEM Bending Stress = 170 KSI

The material yield strength at 350 BHN was estimated at 165 KSI.

RESULTS OF RACK TEST

The rack was loaded against the pinion tooth to a tangential load of 2,100,000 lbs. Failure of the rack had not occurred even though the tooth root fillet strain exceeded the expected yield point of 5500 microstrain or 165 KSI stress.

The rack was not loaded to higher levels because there was concern for the test stand and other elements within the gearbox. Figure 12 shows the plot of strain and stress versus tangential force. The non-linearity of this plot is attributed to the localized yielding that occurs at the peak stress areas. As the load increases, the surface fibers of the root fillet yield first. If the load is increased further, the inner fibers carry increased load until they yield, thus causing a further stress redistribution. This is a characteristic of the ductile, through hardened material that does not seem to exist in the carburized pinion tooth. Higher loading and higher stress can be accommodated with the carburized pinion to the point at which the ultimate strength is reached and then failure occurs. The lack of ductility in the case causes a crack to initiate which results in rapid failure of the pinion tooth.

Gauge 2 read 3527 microstrain and gauge 6 read 2590 microstrain when the loads were released on the rack, indicating that high levels of residual tensile strain were present, and that yielding had taken place.

The contact zone of the rack where the pinion meshed with it, showed localized plastic deformation. The band of axial deformation was 2" wide and had an average depth of about .075 in. The permanent set due to the localized contact renders normal Hertzian stress analysis invalid.

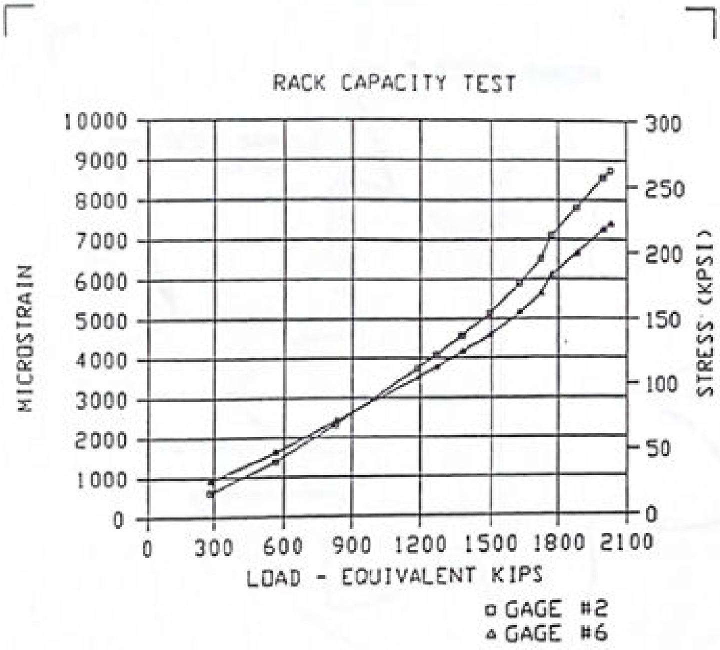

FIGURE 12

TEST 2

VERY LOW CYCLE BENDING FAILURE TEST (6" CP)

TEST OBJECTIVE AND METHOD

The second test was conducted to determine what load would cause catastrophic failure of a through hardened pinion tooth. An identical carburized test piece was made and tested similarly. A completely new test configuration was utilized to test both through hardened, as well as carburized parts. This configuration was also better suited for cyclic loading. The testing was performed on a half scale (6 CP) model of the previous test.

The test pieces as shown in Figure 13 resembled the letter "H". Since we were only interested in the tooth fillet area, the portion of the tooth above the pitch line was contoured to provide a curved surface suitable for installing a hydraulic cylinder between the two opposing teeth. Figure 14 shows the original 7 tooth pinion overlaid on the new test pieces, for comparison. The figure shows that the test profile was identical to the 7 tooth pinion in the root fillet area. Figure 15 is a photo of the actual test setup. The test pieces were two inches thick. Two identical teeth were loaded at the same time with the setup as shown. Failure occurred on the weakest tooth of the pair. The pointed protrusion adjacent to the test teeth was left to be a backup to retain the loaded tooth which breaks off with a high energy release. Two tests were planned for each test sample. One test was to load the piece to failure with incremental loads and return to zero load after each load step. The purpose of the return to zero was to measure permanent deflection of the entire tooth and to check for residual tensile stress at each load increment.

Permanent tooth deflection was measured using Bently Nevada Proximity Probes and recording the DC voltage output which is proportional to the probe air gap with the tooth. The main item of interest was to determine where permanent set started, not necessarily the magnitude of set.

The second test applied cyclic loading at the desired stress level to determine a point in a (low cycle) stress cycle curve. This test was performed on the other half of the "H" shaped test sample. The test load was established to far exceed the highest loading on an actual 7 tooth pinion. The entire setup was covered with a round enclosure for safety reasons, to contain the failed teeth. Tooth root fillets were strain gauged in the center of the relatively narrow face width.

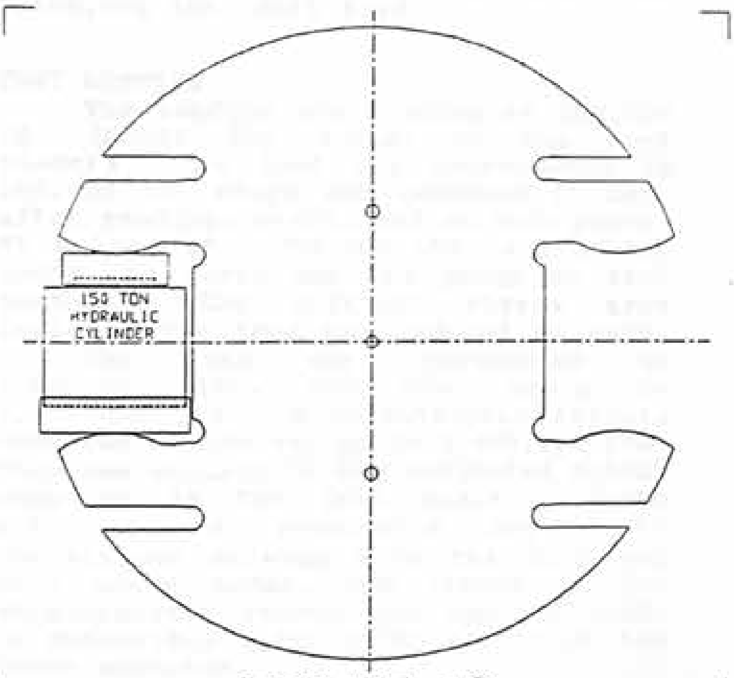

FIGURE 13

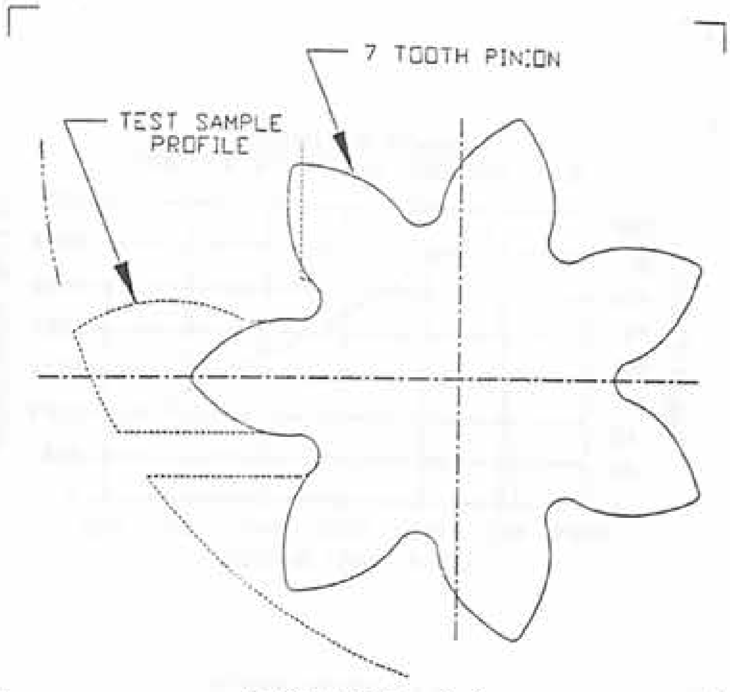

FIGURE 14

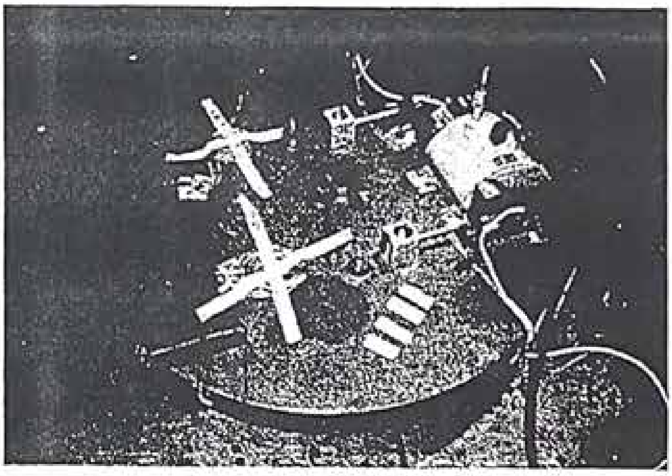

FIGURE 15 - VERY LOW CYCLE BENDING FAILURE TEST SET UP

MATERIAL, HEAT TREATMENT AND MANUFACTURING PROCESS FOR CARBURIZED TEST PIECE

The carburized test piece was made of 4320 steel. It was carburized, hardened and tempered. The case hardness was 58 RC with an effective case depth of .110 in. This depth was one half the depth of the full scale tooth. The entire profile was NC milled after band sawing out the large areas between teeth. A tensile test of the material gave a yield point of 180 KSI and an ultimate strength of 190 KSI.

LOADING AND STRESS ANALYSIS

The half scale model of the pinion used for testing was also analyzed using the finite element program 'ANSYS' in order to duplicate the stress pattern from the full scale model and calculate the load requirements for testing. The assumptions made for modeling were similar to those for the original model. However, the mathematical model was based on the test fixture geometry. The load location for the test model was determined arbitrarily by the location of the Enerpac hydraulic cylinder and it's contact point with the test specimen. The loading at this point was applied to obtain a stress pattern which is similar to the original full scale model. An applied load in the half scale model of 101,700 lb. was calculated to generate the same fillet bending stresses on the full scale 7 tooth pinion as did the 1,100,000 lb. force. This became the scale factor for the testing.

This ratio of loads was maintained so that the full and half scale tests could be correlated. Equivalent load on a full size pinion was really of interest. (The full scale 7 tooth pinion had a measured test strain of 3800 microstrain at 1,100,000 lb. load. The half scale test piece had a measured strain of 3850 microstrain with the 101,700 lb. scale load equivalent of 1,100,000 lbs. full load.

TEST RESULTS

The loading was started at 500,000 lb. (46200 lb. actual on the test piece). The load was incremented in 100,000 lb. steps and returned to zero after readings stabilized at each point. At a load of 1,700,000 lbs. a cracking sound was heard and the gauge on each tooth in the critical stress area failed. The load was reduced to zero.

The load was incremented to 1,800,000 lbs. and then again to 1,900,000 lbs. A catastrophic failure occurred on the way up to 2,000,000 lbs. This was similar to the estimated actual capacity in the full scale 7 tooth pinion. A reasonable degree of correlation existed with the full and half scale model. See Figure 16 for strain/stress versus the applied load. No measurable plastic deflection of the tooth occurred.

The cyclic test at lower loading on the carburized pinion was aborted because an excessively high load was accidentally applied during the setup which caused an immediate crack which precluded the planned cyclic testing.

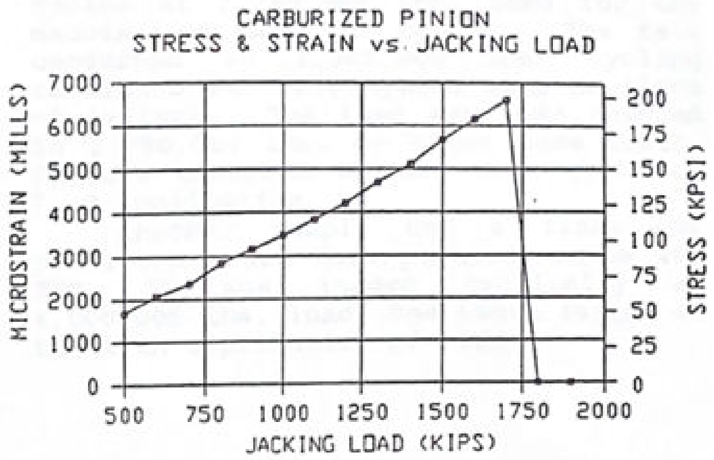

FIGURE 16

THRU HARDENED TEST SAMPLE MATERIALS, HEAT TREATMENT AND MANUFACTURING PROCESS

The through hardened part was made of 4340 steel. It was heat treated after milling the profile and tempered at $1000^{o}F$ to 400 BHN. It was hardened after the milling of the profile.

LOADING AND STRESS ANALYSIS

The loading and stress analysis were identical to the carburized part. The only change between the two test pieces was the material and the heat treatment.

SUMMARY OF RESULTS

Initially it was not known where a through hardened part would fail. It was estimated that a maximum test load of 4,000,000 lbs. would be sufficient to cause failure. The loading was incremented to 1,700,000 lbs. which generated fillet strains of 6000 microstrain, equivalent to 180 KSI bending stress. Yielding was expected to start at this point. Above this applied load the residual strain began to increase rapidly when the load was reduced to zero. The loading continued to the 4,000,000 lb. level which was the limit of the test equipment. This load was in excess of twice the failure load of the carburized part. See Figure 17 for Strain/Stress versus Load. This was the 36th cycle of load application of increasing magnitudes.

Measurable permanent set began to occur at 2,500,000 lbs. See Figure 18.

A set of .006 in. occurred at 4,000,000 lbs. Since a single load had not failed the part it was decided to start cycling the part at the 4,000,000 lbs. scale, (370,000 lbs. actual) level. On the 35th cycle at this load a catastrophic failure occurred.

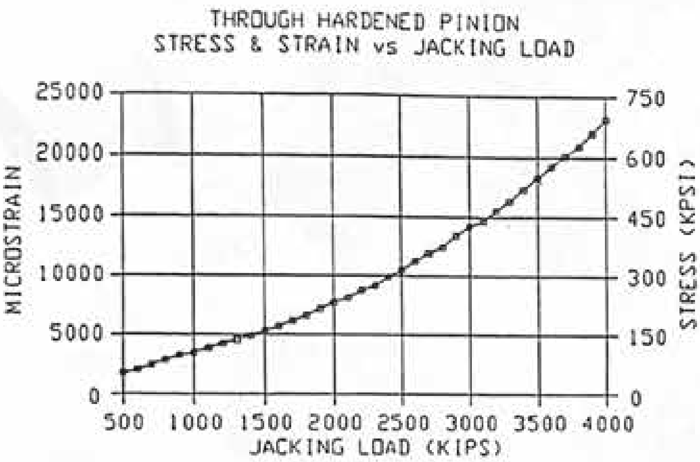

FIGURE 17

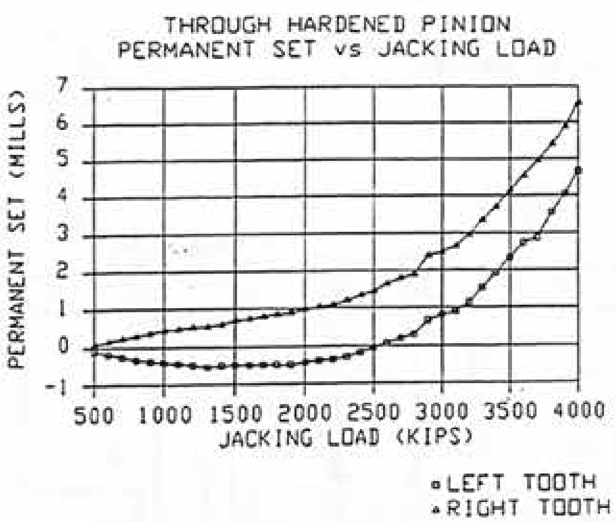

FIGURE 18

ENDURANCE TEST

The second set of teeth in the through hardened test piece was now loaded at the 1,365,000 lb. load level to determine the cycles to failure. The actual 7 tooth pinion has a total operating life requirement of only 2300 cycles at a 1,171,000 lbs. Miners Rule effective load. The required number of cycles at 1,365,000 lbs. load for the machine is only 319 cycles. The test conducted at 1,365,000 lbs. cycling continued for 3070 cycles with no signs of failure. The load was then doubled to 2,750,000 lbs. on these same teeth. Failure occurred on the 14th cycle of load application.

Another sample had a flame cut profile and was through hardened to 405 BHN. It was loaded cyclically at 4,000,000 lbs. load. One tooth failed on the 19th application of load.

TEST 3

LOW CYCLE DURABILITY, WEAR, BENDING AND REVERSE BENDING TEST

TEST OBJECTIVES AND METHOD

The final phase of the testing was conducted to evaluate a through hardened spur planet gear versus the carburized planet gears used in the final reduction stage for the jack-up gear drive. This stage was configured as a planetary epicyclic. Durability, strength and wear characteristics were desired from this test.

A third test stand was manufactured from a standard, double reduction gear housing which was reconfigured to be a self contained back to back, or four square type of test stand, see Figure 19. Internal loop loading was accomplished by external torque arms, loaded against each other with jacking screws. The loop torque was verified by strain gauging of the inner quill shaft which was calibrated with a known external force. The system was rotated by the remaining input shaft at approximately 6 RPM. See Figure 20 for the actual layout with the heat treatment of the components identified. One of the center idlers was through hardened. The other idler was carburized. An idler gear is subject to reverse tooth bending, much the same as is a planet gear. This arrangement was utilized to simulate a planet gear. The gears were splash lubricated with Mobil 636 SHC oil and were run with a nominal operating temperature of $80^{O}F$.

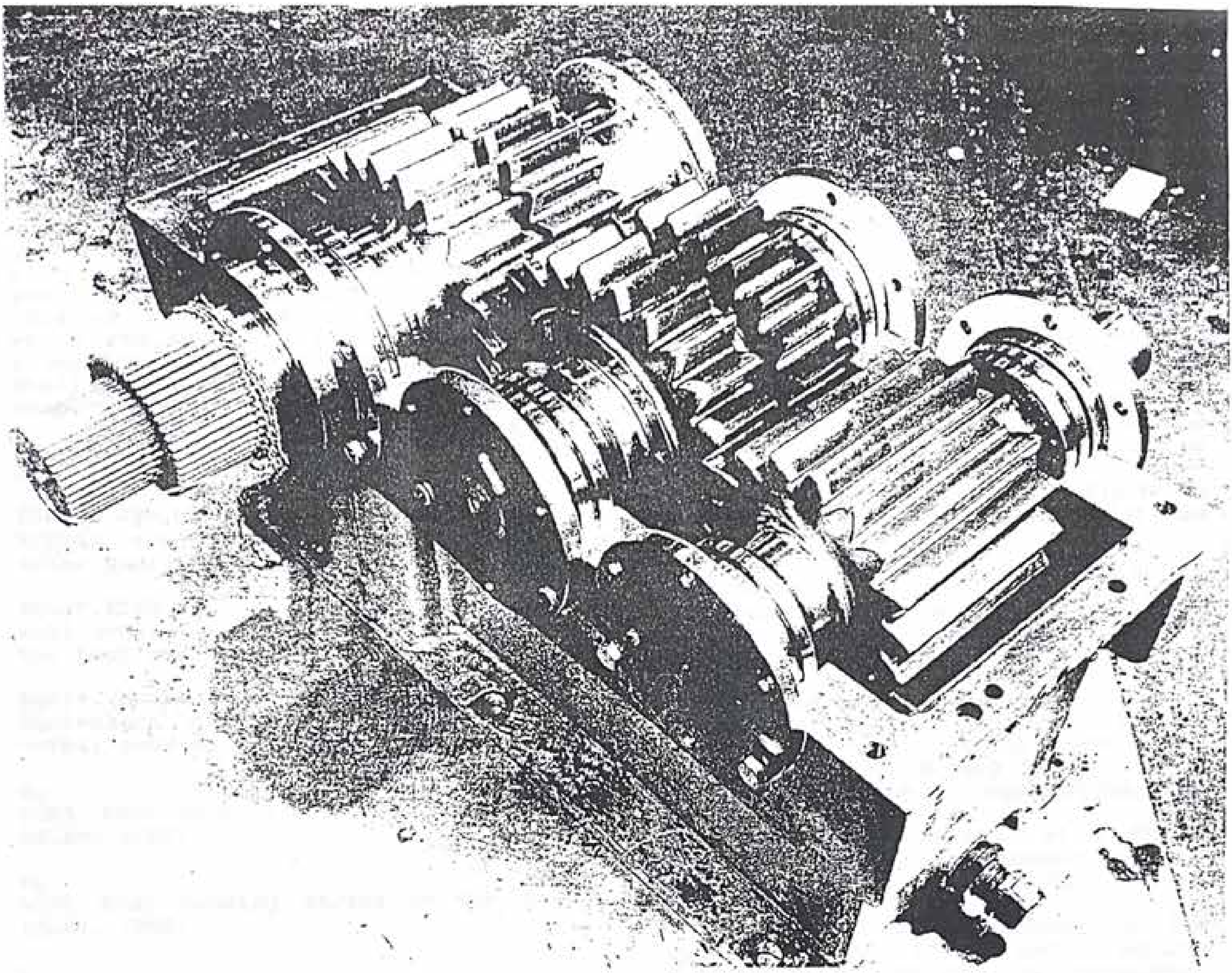

FIGURE 19 - RECONFIGURED FOUR SQUARE TEST STAND

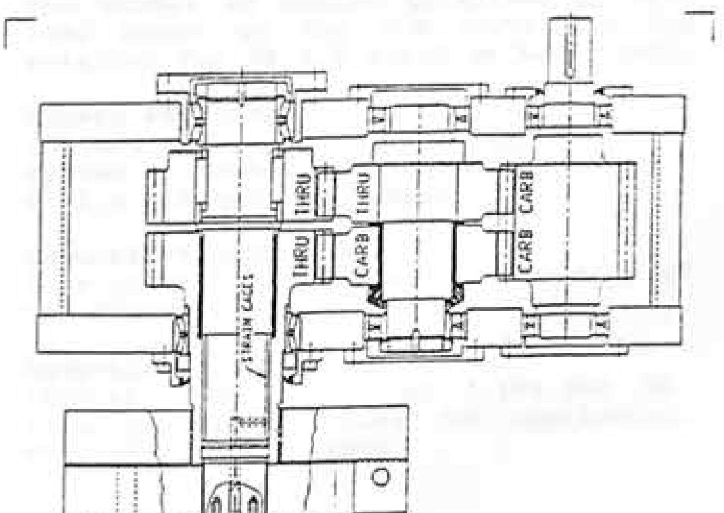

FIGURE 20

MATERIALS, HEAT TREATMENT AND MANUFACTURING PROCESS

The carburized idler (planet) was made of 4320 steel. It was carburized, quenched and tempered to a final hardness of 58 RC. This also applies to the right side, or input pinion shaft. The through hardened planet was made from 4340 steel, liquid quenched and tempered to 388 BHN. The two torquing pinions on the left side of the test stand were also through hardened, the same as the idler above. All of the carburized elements and the through hardened planet were ground. The other members were hobbed only. This arrangement evaluated a carburized-carburized mesh, a through hardened-through hardened mesh, and a carburized-through hardened mesh simultaneously at extremely high loading. Again, the total life requirements in actual service for this application are extremely low, requiring only 6500 cycles of the planet gear at an effective load of 1,171,000 lbs. output pinion tangential force.

LOADING, RATING AND STRESS ANALYSIS

The loading is expressed in terms of the full size actual jack-up gear for consistency and comparison purposes.

The test simulated various operating conditions of the jacking gear. The three basic load conditions were the Miner's Rule equivalent loading of 1,175,000 lbs. tangential force in the 7 tooth output pinion for 177 hours, 125% of the preload jacking condition which was equal to 1,625,000 lbs. and 2,050,000 lbs. simulating the maximum stall torque of the motor. Table 1 is a summary of all the testing performed.

The following is a more detailed explanation of each item evaluated in the table:

Planet Cycles
Actual revolutions of the planet or idler gear in the test.

Equiv.KIPS
Full scale equivalent load simulated by the test conditions.

Equiv. Hours
Equivalent number of hours run at normal jack-up rig speeds.

S_c
AGMA 2001 contact stress on the test idler. (PSI)

S_b
AGMA 2001 bending stress on the test idler. (PSI)

K
"K" factor, which is a relative measure of tooth compressive loading.

Allow Cycles N_f
The number of cycles permitted at this load based on the S-N curve for the material for SF 1.0 based on AGMA 2001.

Miners Fraction
(Actual cycles at test load)/(Allowable cycles at that load) based on AGMA 2001 SF 1.0 allowable stresses.

Cumulative Life Used
This number represents the summation of the Miners fraction of life used.

Equivalent Life Fraction
(Actual test cycles at 1,175,000 lb. load)/(Required cycles for application at 1,175,000 lb. load)

SUMMARY OF RESULTS

The testing was concluded when the through hardened planet teeth exhibited spalling over approximately 50% of their contacting flanks. This member has survived 46.5 times the required cycles and at the specified loads. Pitting on this member did not start until after the twentieth test condition in Table 1. The carburized planet was well polished but did not pit. The through hardened planets demonstrated metal rolling over at the tips and roots at the fifth test condition in Table 1. It was initially thought that excessive wear would be the primary failure mode. The gears were disassembled after the sixteenth test condition in Table 1 to perform geometry checks to measure the wear. The actual measured wear on the through hardened planet was about .003-.004 in. as compared to about .001 in. for the carburized members. The wear continued at this rate for the duration of the testing. There was no tooth breakage or fatigue cracks initiated for any of the components of the test. See Figure 21 for condition of the teeth after the last test.

CONCLUSIONS

1. The tests demonstrated that the through hardened gearing had at least a 2:1 margin on breakage capacity from statically applied bending loads compared to carburized gears.
2. Through hardened gears yield locally in the critical stress areas and redistribute load toward the center of the tooth until equilibrium is maintained.
3. Carburized gears fail at a stress level that is in good agreement with the ultimate strength capability of the carburized case.
4. Although wear occurred in the dynamic testing, it was not a failure mode for gears with this low rotational speed. The through hardened gear, did not have the pitting resistance of the carburized gear. The pitting life of the through hardened gearing proved adequate for the application and capable of at least 40 times the expected duty cycle requirements before spalling becomes significant. Failure in this application would be considered to be the point where tooth breakage occurs.

RECOMMENDATIONS

Further testing should be performed to develop proper statistical variation of the measured characteristics. Additional testing should also be performed to determine whether the same characteristics apply to shock loading conditions as well as statically applied loads.

ACKNOWLEDGEMENTS

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							CARBURIZED PLANET Sac = 225000				THROUGH HARDENED PLANET Sac = 168230 (388BHN)			
TEST NO.	PLANET CYCLES	EQUIVALENT		STRESS			ALLOW. CYCLES Nf	MINER'S FRACTION OF LIFE	CUMUL LIFE USED	EQUIV LIFE FRACT	ALLOW. CYCLES Nf	MINER'S FRACTION OF LIFE	CUMUL LIFE USED	EQUIV LIFE FRACT
		KIPS	HRS	Sc	Sb	K								
1	34	820	0.9	246142	57143	1769	2010943	0.00	0.0	0.0	11179	0.00	0.0	0.0
2	47	1091	1.3	283945	76044	2354	156816	0.00	0.0	0.0	872	0.05	0.1	0.0
3	338	1186	9.2	296075	82680	2560	74293	0.00	0.0	0.1	413	0.82	0.9	0.1
4	1228	1135	33.4	289653	79132	2450	109907	0.01	0.0	0.2	611	2.01	2.9	0.2
5	515	1137	14.0	289881	79257	2454	108374	0.00	0.0	0.3	602	0.85	3.7	0.3
6	1710	1147	46.6	291126	79939	2475	100387	0.02	0.0	0.5	558	3.06	6.8	0.5
7	440	1147	12.0	291220	79991	2477	99811	0.00	0.0	0.5	555	0.79	7.6	0.5
8	1618	1138	44.1	290013	79329	2456	107499	0.02	0.1	0.7	598	2.71	10.3	0.7
9	635	1144	17.3	290749	79732	2469	102740	0.01	0.1	0.8	571	1.11	11.4	0.8
10	3	1984	0.1	382981	138341	4283	750	0.00	0.1	0.9	4	0.75	12.2	0.9
11	210	1563	5.7	339896	108965	3374	6317	0.03	0.1	1.3	35	5.98	18.1	1.3
12	347	1542	9.4	337559	107472	3327	7145	0.05	0.1	1.9	40	8.73	26.9	1.9
13	673	1601	18.3	344005	111616	3456	5097	0.13	0.3	3.6	28	23.76	50.6	3.6
14	245	2050	6.7	389297	142942	4426	560	0.44	0.7	9.1	3	78.65	129.3	9.1
15	1163	1626	31.7	346696	113369	3510	4435	0.26	1.0	12.5	25	47.16	176.4	12.5
16	2081	1625	56.7	346565	113283	3507	4465	0.47	1.4	18.4	25	83.83	260.3	18.4
17	979	1651	26.7	349360	115118	3564	3868	0.25	1.7	21.6	22	45.53	305.8	21.6
18	1346	1665	36.7	350850	116102	3595	3585	0.38	2.1	26.4	20	67.55	373.4	26.4
19	612	1660	16.7	350252	115707	3582	3696	0.17	2.2	28.5	21	29.78	403.1	28.5
20	1591	1651	43.3	349300	115078	3563	3880	0.41	2.7	33.7	22	73.76	476.9	33.7
21	1958	1652	53.3	349471	115191	3566	3847	0.51	3.2	40.2	21	91.58	568.5	40.2
22	734	1653	20.0	349580	115263	3569	3825	0.19	3.4	42.7	21	34.54	603.0	42.7
23	1102	1660	30.0	350313	115746	3584	3685	0.30	3.7	46.5	20	53.78	656.8	46.5

TEST DATA SUMMARY

TABLE # 1

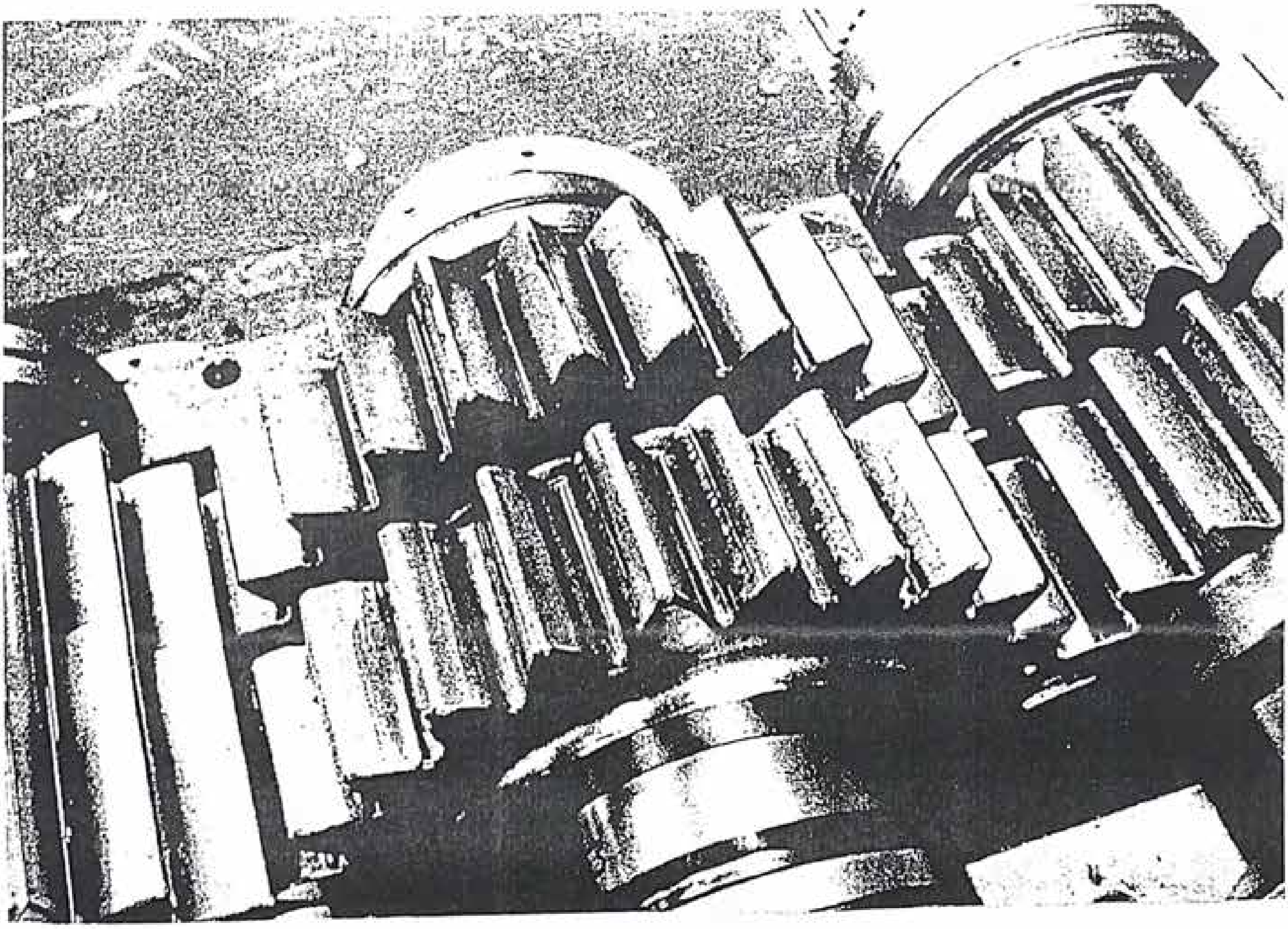

FIGURE 21 - TEST STAND TOOTH CONDITION