J. MEYER

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LIFE HISTORY OF USS PLAINVIEW (AGEH-1) HYDROFOIL POWER TRANSMISSION SYSTEM

DAVID W. TAYLOR NAVAL SHIP RESEARCH AND DEVELOPMENT CENTER

Bethesda, Maryland 20084

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by
Henry W. Schab

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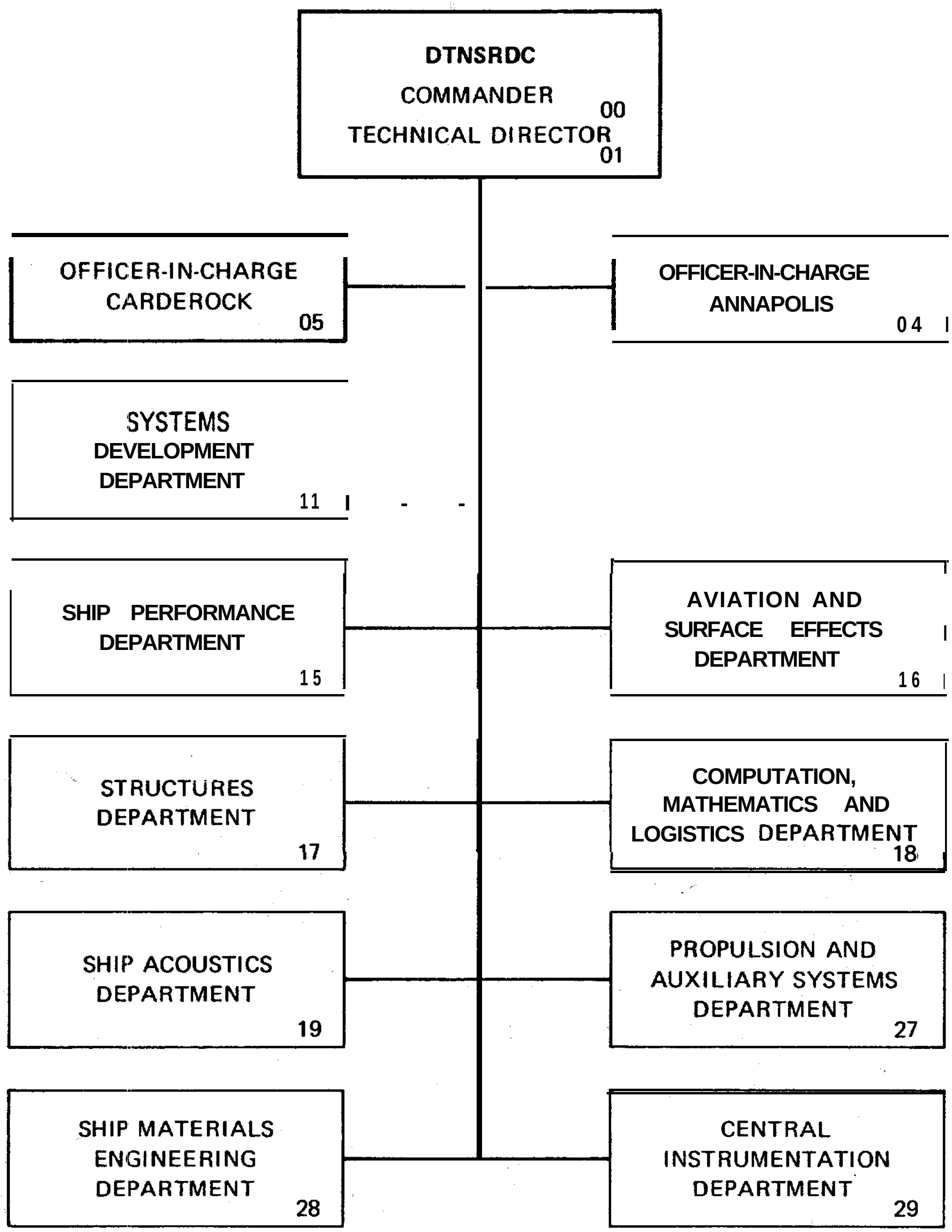

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This report records the history of the design, manufacture, shop and development tests, craft installation, operations, inspections and modifications of the large bevel gears, pinions, and associated components of the USS PLAINVIEW (AGEH-1) foilborne power transmission system from its inception in 1962 to the premature deactivation of the craft in 1978.

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Of the 288 hours of foilborne operation, 268 hours were at 60 percent rated power and 20 hours were at 90 percent power. After the deactivation of the AGEH, the transmission systems were removed, inspected, and stored. The final inspection of the gears, pinions, bearings, and couplings indicated a relatively healthy system which has many more hundreds of hours of operational life for some future application.

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LIST OF ABBREVIATIONS

CEVM - Consumable electrode vacuum melt

cpm - Cycles per minute

d.c. - Direct current

D&L - Diehl and Lundgaard, Incorporated

dia - Diameter

Dwg - Drawing

°F - Degrees Fahrenheit

ft/lb - Foot/pound

ft/min - Feet per minute

fwd - Forward

gal - Gallon

GE - General Electric Company

horiz - Horizontal

hp - Horsepower

hr - Hour

HYSTU - Hydrofoil Special Trials Unit

HZ - Hertz

IB - Inboard

ID - Inside diameter

in/sec - Inch per second

lb - Pound

lb/in - Pound per inch

LS - Low speed

LSCC - Lockheed Shipbuilding and Construction Company

m - Meter

mfg - Manufacturer

min - Minute

mo - Month

NAVSEA - Naval Sea Systems Command

NAVSEC - Naval Ship Engineering Center

OB - Outboard

OD - Outside diameter

psi - Pounds per square inch

rms - Root mean square

rpm - Revolutions per minute

RAV - Restricted availability

SI - Single idler

S/N - Serial Number

Stbd - Starboard

TIR - Total Indicator Reading

Vert - Vertical

yr - Year

ABSTRACT

This report records the history of the design, manufacture, shop and development tests, craft installation, operations, inspections and modifications of the large bevel gears, pinions, and associated components of the USS PLAINVIEW (AGEH-1) foil-borne power transmission system from its inception in 1962 to the premature deactivation of the craft in 1978.

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ADMINISTRATIVE INFORMATION

This study was sponsored by the Naval Sea Systems Command (NAVSEA)* under Task Area S0337001, Task 01700 and was administered by the Advanced Hydrofoil Systems Office, David W. Taylor Naval Ship Research and Development Center (DTNSRDC) under Work Unit 1150-002.

ACKNOWLEDGMENT

The author wishes to gratefully acknowledge the technical contribution of Ted Csaky, Code 2721, DTNSRDC.

INTRODUCTION

This report records the history of design, development, test, installation, operation, inspections, and modifications to the USS PLAINVIEW (AGEH-1) foilborne transmission system.

Information has been obtained from the ship builder's reports of tests and operations; the gear manufacturers' development and test reports and design criteria; the support contractors' inspection and modification reports; the Navy Technical team's evaluation, inspections, and reports and the craft logs.

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*Definitions of abbreviations used are given on page vii.

A technical history of the high power, bevel gear "Z" drive, propeller transmission system and documentation of the lessons learned will be valuable for future Navy ship projects that contemplate this type of power system.

DESCRIPTION OF USS PLAINVIEW (AGEH-1) TRANSMISSION SYSTEM

The arrangement of the port and starboard sides of the propulsion system are mirror images so that the relationship among the foilborne transmission system components on one side is the same as those on the opposite side.

The principal elements of the transmission system are shown in Figure 1. Power from the port and starboard main propulsion gas turbine engines is transmitted through the engine drive shafting, the single reduction gearbox, the single reduction gear output shafting, the upper strut bevel gearboxes, the strut vertical shafting, the lower strut bevel gearboxes, to the pod auxiliary equipment and finally to the propeller shaft assemblies.

Table 1 gives the weights of different components of the transmission system.[1*] The total weight of the port and starboard systems is almost 18 tons.

Table 2 gives the design characteristics of the bevel gears on the transmission system.

INITIAL CONTRACT REQUIREMENTS FOR GEAR DEVELOPMENT

The purpose of the spiral bevel gear development program was to provide the capability for design and manufacture of spiral bevel gears for the foilborne transmission system.

The design and development effort of the gears was undertaken by the General Electric Company (GE) under contract from Grumman Aircraft Engineering Corporation (Grumman), the prime contractor for design of PLAINVIEW.

This contract called for GE to accomplish the following:[2]

1. Design spiral bevel gearings and associated hardware for the transmission system.

2. Develop the metallurgy and manufacturing techniques necessary to produce the bevel gears.

3. Design and manufacture test equipment to run load tests on the gears and bearings.

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*A complete listing of references is given on page 93.

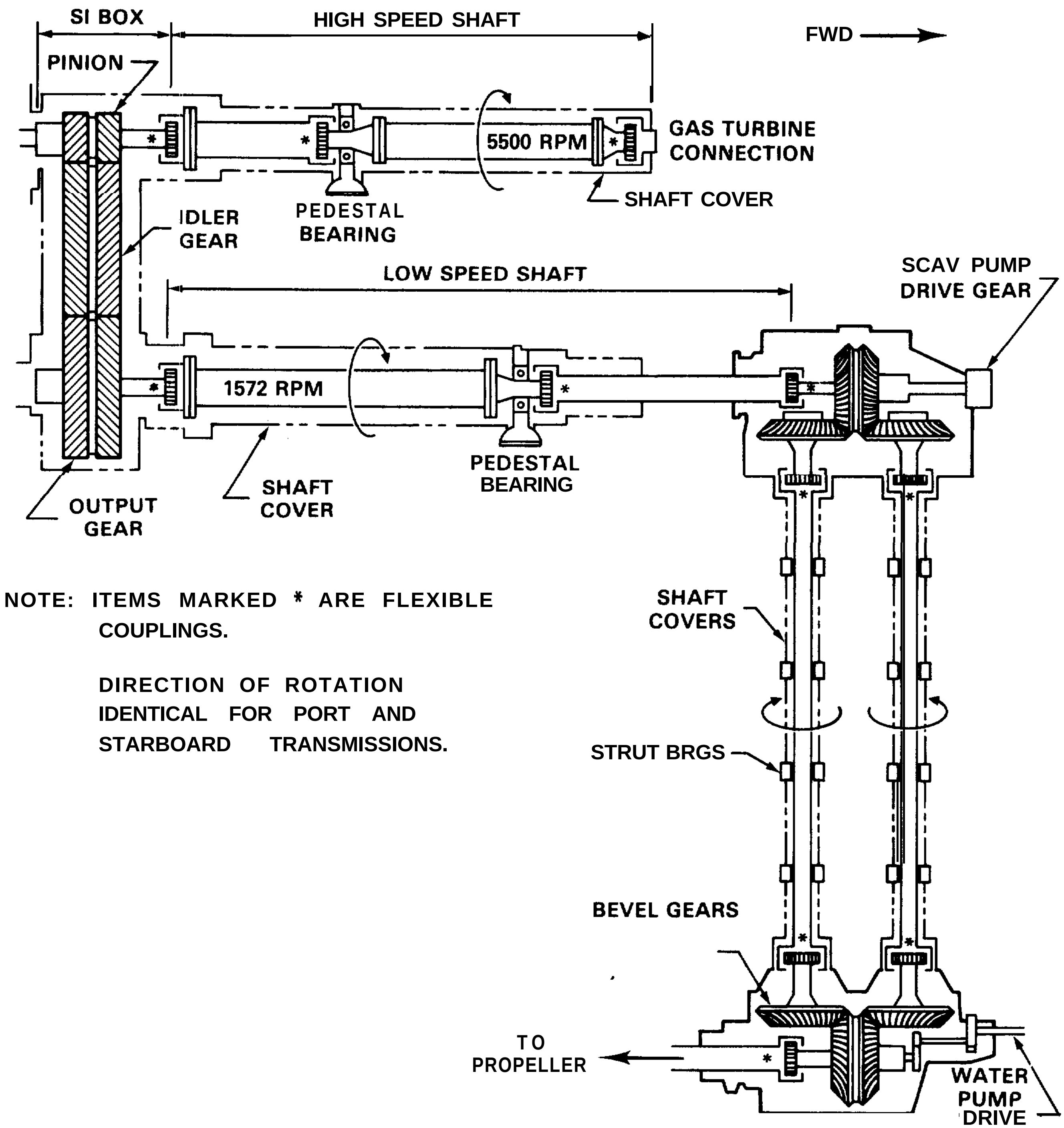


Figure 1 - Schematic of Foilborne Transmission System (One S -de)

TABLE 1 - COMPONENT WEIGHTS OF FOILBORNE TRANSMISSION SYSTEM

Transmission Components	Weight (lb)
Engine Drive Shaft	1,274
Single Reduction Gear Unit	6,088
Single Reduction Gear Output Shaft	1,933
Upper Bevel Gearbox Assembly	3,534
Lower Bevel Gearbox Assembly	3,208
Strut Shafting Assembly	1,626
Total Weight per Side	17,663
Total. Weight per Ship (Port and Starboard)	35,326

4. Run load tests on the gears and bearings to determine design characteristics of the system.

In September 1962, Grumman produced the "Specification for Main Power Transmission for AGEH Hydrofoil Research Ship."[2] This specification described the requirements for the design, manufacture, and test of the foilborne main propulsion drive system which included all the power transmission machinery between the output splines of GE Model 7LM1500PC101 turbo shaft engines and the propellers; not included were the propellers.

Pertinent details from the specification are:

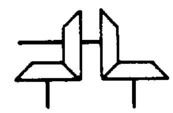
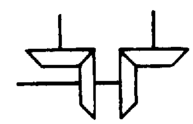
1. A target weight for the entire drive system of 46,250 lb.

2. Vibration pick-ups mounted on each gearbox "in critical places and directions."

3. Overall drive system so designed that overhaul interval shall not be less than 1,000 hr.

4. Gears and bearings in the struts and pods designed for a minimum life of 2,000 hr for bearings and 30,000 hr (approximately 20 yr) for gears when operated for 20 percent of the time at take-off power and 80 percent of the time at maximum continuous power. (Take-off power per engine is 17,500 hp at 5,500 rpm and maximum continuous power per engine is 14,000 hp at 4,950 rpm.)

TABLE 2 - SPIRAL BEVEL GEAR CHARACTERISTICS
(FOILBORNE SYSTEM)

	Upper	Lower
Input Horsepower (Continuous Rating)	14,000	14,000
Arrangement		
Horsepower/Mesh	7350 (52.5%)	7350 (52.5%)
RPM In/Out	1414/1387	1387/1414
Pitch Diameters In/Out (in.)	25.5/26.0	26.0/25.5
Number Teeth N_p/N_g — N_p = number of pinion teeth, N_g = number of gear teeth	51/52	53/51
Ratio (N_p/N_g)	1.0196:1	0.9807:1
Diametral Pitch (in.)	2.0	2.0
Face Width (in.)	5.405	5.405
Bevel Gear Diameter, Maximum (in.)	26	26
Face Contact Ratio	2.37	2.37
Spiral Angle (degrees)	30	30
Pressure Angle (degrees)	20	20
Material	AISI 9310	
Heat Treat	Carburize, Rc58-63 Case, Rc30-38 Core	
Input Torque (lb-in.)	327,409	333,850
Tangential Load (lb)	25,680	25,680
Unit Load (lb-in.)	4751	4751
Bending Stress (lb-in.^{2})	28,290	28,290
Compressive Stress (lb-in.^{2})	136,690	138,630
Scoring Index	18,096	17,920
Pitch Line Velocity (ft/min)	9441	9441
Lube Oil	Mobile RL-285C	(MS 2190TEP)
Gearbox Weight (lb)	4005	3735
Weight Ratio (lb/hp)	0.286	0.266

NAVY SPONSORED BEVEL GEAR TESTING

CONTRACT WITH GENERAL ELECTRIC

Early in 1964 GE became a subcontractor for the transmission system to the PLAINVIEW builder, Lockheed Shipbuilding and Construction Company (LSCC). During this period, GE was performing load tests on the bevel gears for PLAINVIEW. Early BUSHIPS correspondence files show that GE was experiencing difficulties with the bevel gears. GE stressed that the required pitch diameter of 26.125 in. was larger than any that had ever been designed and manufactured in the United States. Until then, the largest spiral bevel gear design handled by GE had a 20-in. pitch diameter. The Navy awarded GE a separate contract for extensive testing of the PLAINVIEW bevel gears; issued early in 1964, contract NObs 90348, work was to be completed by June 1964.

The contract called for development testing on a bevel gear test unit that was representative of those to be used in the PLAINVIEW foilborne transmission system. The gears were to be tested for:

1. 50 hr at 100 percent speed, 75 percent load.
2. 100 hr at 100 percent speed, 100 percent load.

Testing was not completed by the specified date; another agreement was issued by the Navy on 27 June 1964, increasing the contract funding and extending the completion date to 31 December 1964. Again the work was not completed by the specified date. GE was given three more extensions; the last extension, signed on 25 January 1966, required no completion date but did require the following additional tests and information:

1. 100 hr at 1,720 rpm and 328,000 lb-in. torque.
2. 50 hr at cruise rpm and 383,000 lb-in. torque (20 percent above cruise torque).
3. 50 hr at cruise rpm and 398,000 lb-in. torque (25 percent above cruise torque).
4. A final technical report covering all the work done in the contract.

The tests were finally completed in 1966 but for some unknown reason(s) the report was not issued until November 1969 (only after Navy insistence that the manufacturer fulfill his contract). Authored by Smith,[4] much pertinent technical data and information was missing.

TEST FACILITIES AND TEST CONDITIONS

A sketch of the bevel gear test facility is shown in Figure 2. This is a standard four square test rig; it uses a load gearbox to close the torque loop with the bevel gearbox under test, a torquing device, and a power source. A PLAINVIEW lower bevel gearbox was used to support the test bevel gears and was mounted to the base of a vertical cylinder. The test load gearbox was mounted on the top of the cylinder and connected to the bevel gearbox by two parallel vertical shafts. On one shaft, a large hydraulic torque applier was installed. Power was provided by an 800 hp d.c. electric motor connected to the bevel gearbox horizontal output shaft through a speed increasing gear set.

Test plans called for periodic inspections of the bevel gear tooth contact pattern and overall condition of the gears by dropping the lower portion of the gear casing. Such inspections enabled detection of tooth surface distress as well as any early stages of unfavorable contact; thus, major damage could be avoided and corrective action taken.

Torque loading the bevel gears in the test rig differs from the torque loading when the gears are installed in PLAINVIEW. For both the rig and PLAINVIEW, the bevel gears are arranged within the casing so that they are rigidly fastened to each other, that is, they are back to back, each one engaged by its mating pinion. In the PLAINVIEW installation, the torques of the back-to-back gears are in the same direction (Figure 3); however, in the test rig, the torques of the back-to-back gears are in the opposite direction (Figure 4). In order to duplicate the gear reaction and bearing loads to the greatest possible extent, the hand of the spiral bevel of one of the sets of test gears was reversed. Thus, all of the gear reaction loads and bearing and casing loads are identical, except for the direction of the tangential load of the set with the reversed spiral. It should be noted that this manner of test loading does result in dissimilar deflections of the back-to-back gear shafts (compare view B in Figures 3 and 4).

This back-to-back torque speed test rig has a low power requirement because the only power required to operate this assembly is that necessary to overcome gearbox losses (about 1 percent of the tested hp/mesh).

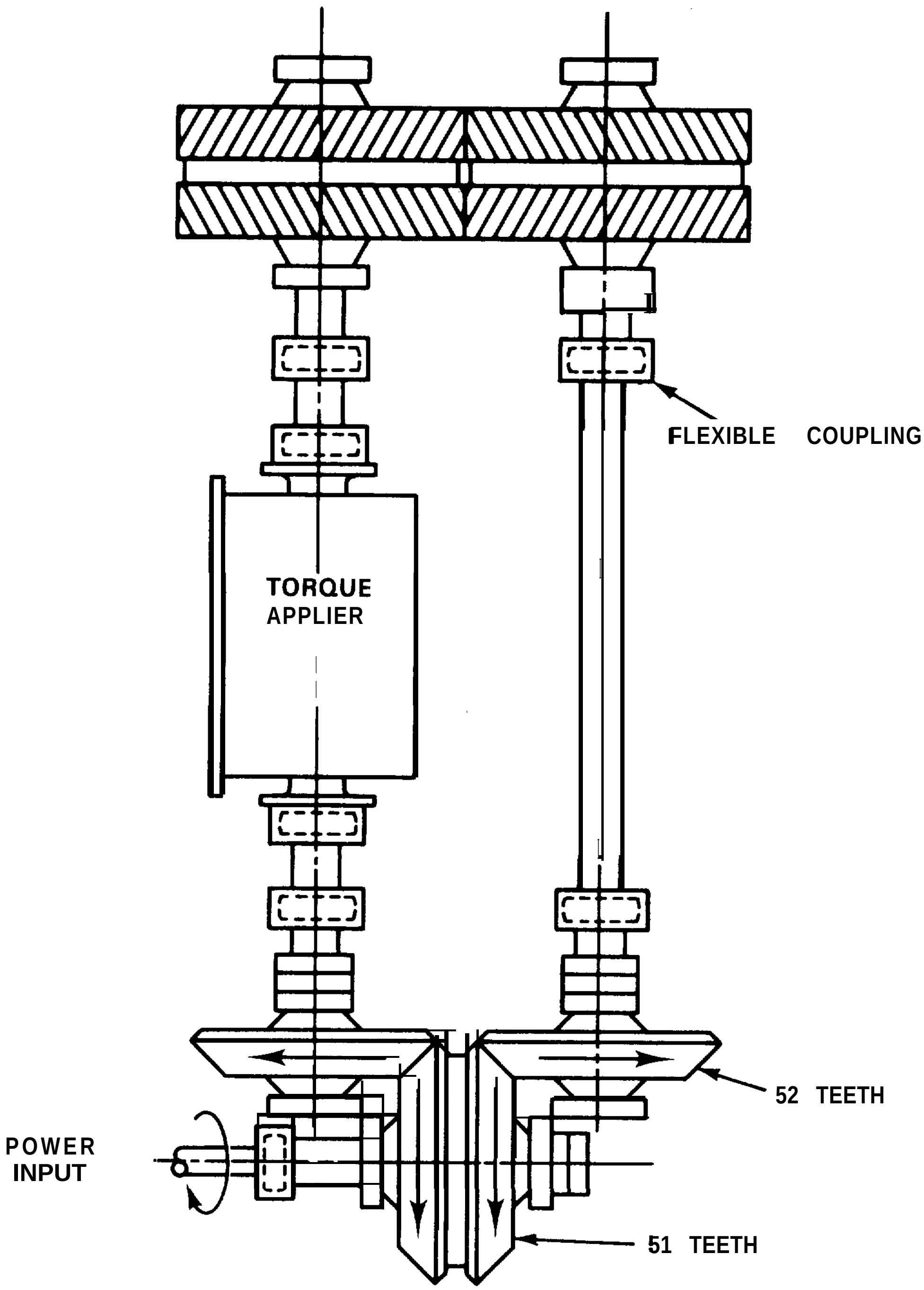


Figure 2 - Test Gear Torque Loop For Shore Base Testing of Bevel Gears[5]

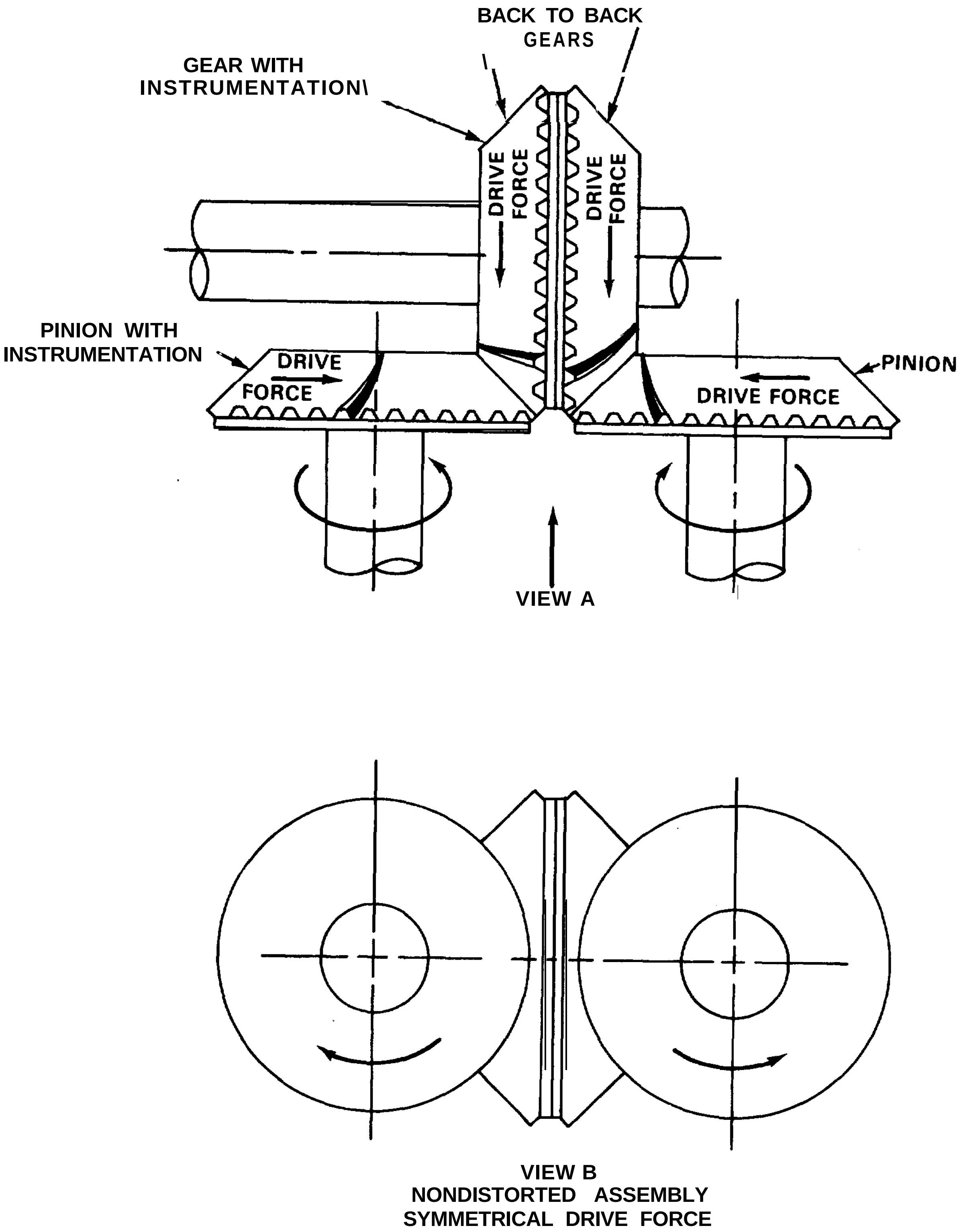
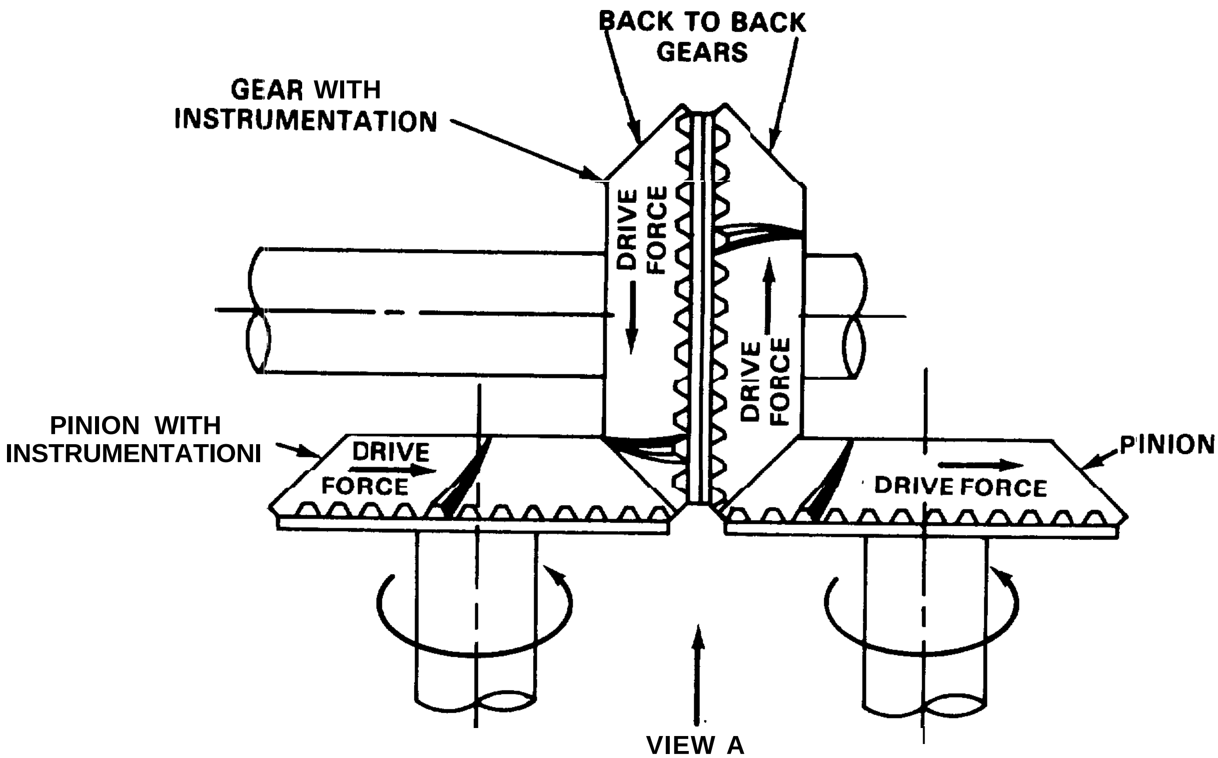


Figure 3 - Bevel Gear Static Load - Plainview Condition[4]



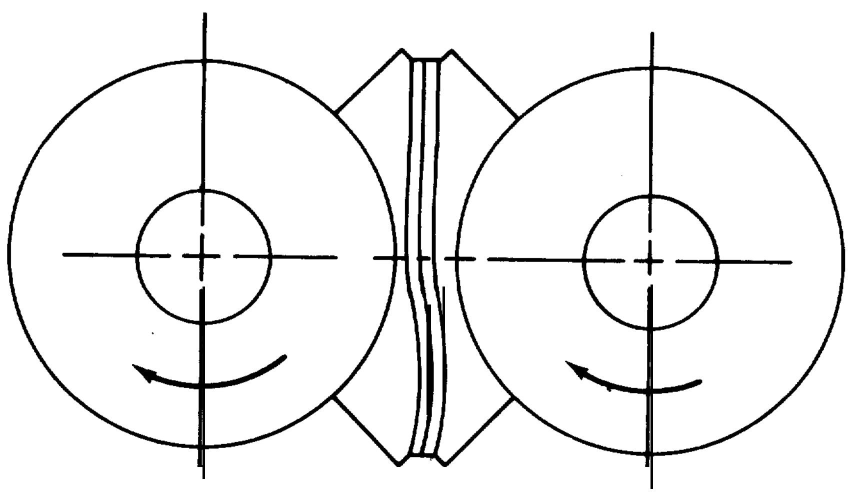


Figure 4 - Bevel Gear Static Load - Test Condition[4]

TEST GEAR MATERIALS AND GEAR SETS

All four sets of bevel gears were designed and manufactured for test purposes. Two sets were fabricated of aircraft quality AISIE9310 normalized forging and one set of consumable electrode vacuum melted (CEVM) AISIE9310 normalized forging. The fourth set, fabricated of an aircraft quality 5 percent nickel, 2 percent aluminum nitriding steel, normalized forging, was not machined into a finished gear set: there were indications that nitrided gears would not possess the load capability of case-hardened gears. (The forging is now at the Hydrofoil Special Trials Unit (HYSTLJ) at Bremerton, Washington.) Except for the CEVM materials, all of the gear blank forgings were formed into a pancake shape, then pierced, and then forged into donut-shaped rings. The CEVM material, a 16-in. cube of steel with known grain flow line orientation, was hammered with the die faces normal to the flow lines to produce a pancake; the center was then pierced, and a final forming die was used to produce a conical-shaped gear blank.

The essential difference between the aircraft quality 9310 and CEVM 9310 is that the latter requires closer quality control over such "dirty" elements as phosphorous and silicon. Table 3 gives the material and physical properties of AISIE9310.

TEST RESULTS

The AISIE9310 gears were ordered from Gleason Works in May 1962 and initial tests began in August 1963. After approximately 36 hr total test, of which over 20 hr were at full load or higher, the test was terminated in February 1964 due to a tooth failure. Full torque load in these tests of 360,000 lb-in./mesh was equivalent to an 80 knot ship with a propeller speed of 3,130 rpm; these test conditions simulated the four engine (70,000 hp) 80 knot ship that PLAINVIEW was originally designed to be.

In addition to the tooth failure, there were problems as well with the bearings. The inner races of the large heavily-loaded roller bearings (located on the horizontal gear shaft and at the bottom of the vertical gear shafts) had rotated on their shafts and showed signs of fretting corrosion. The bearings were MR 234 roller bearings with a rated radial load of approximately 30,000 lb each. These bearings had been installed with the bearing manufacturer's recommendation of a 0.0006 to 0.0014 in. interference fit of the inner race. Before the next tests

TABLE 3 - PROPERTIES OF AISIE9310 STEEL[4]

Chemistry	Ladle Analysis (percent)
Carbon	0.08-0.13
Manganese	0.45-0.65
Silicon	0.20-0.35
Phosphorus (maximum)	0.025
Sulphur (maximum)	0.025
Chromium	1.00-1.40
Nickel	3.00-3.50
Molybdenum	0.08-0.15
Physical Properties	
Tensile Strength	135,000 psi
Yield Strength (minimum)	100,000 psi
Hardness	30-38 Rc
Reduction of Area	60 percent
Elongation	16 percent
Impact, Izod	93 ft/lb

were scheduled, the scored shafts were plated up to 0.002-0.003 in. interference fit and fitted with steel keys made of mild steel which were later replaced with steel keys of 32-38 Rc material between the shaft and slots milled into the ends of the race. Identical alterations were also made to the four PLAINVIEW shipboard gearboxes.

With the teeth profile modified to ensure improved tooth contact, a new set of bevel gears was fabricated from AISIE9310 and tests started again in June 1964. Testing was terminated after approximately 48 hr at 100 percent torque load, at a speed of 1565 rpm (100 percent of 50 knot propeller speed), because of a loud noise coming from the test gearbox (vibration increased from 0.5 to 1.5 mils). Examination of the gears revealed a broken tooth in the 52 tooth forward input gear which

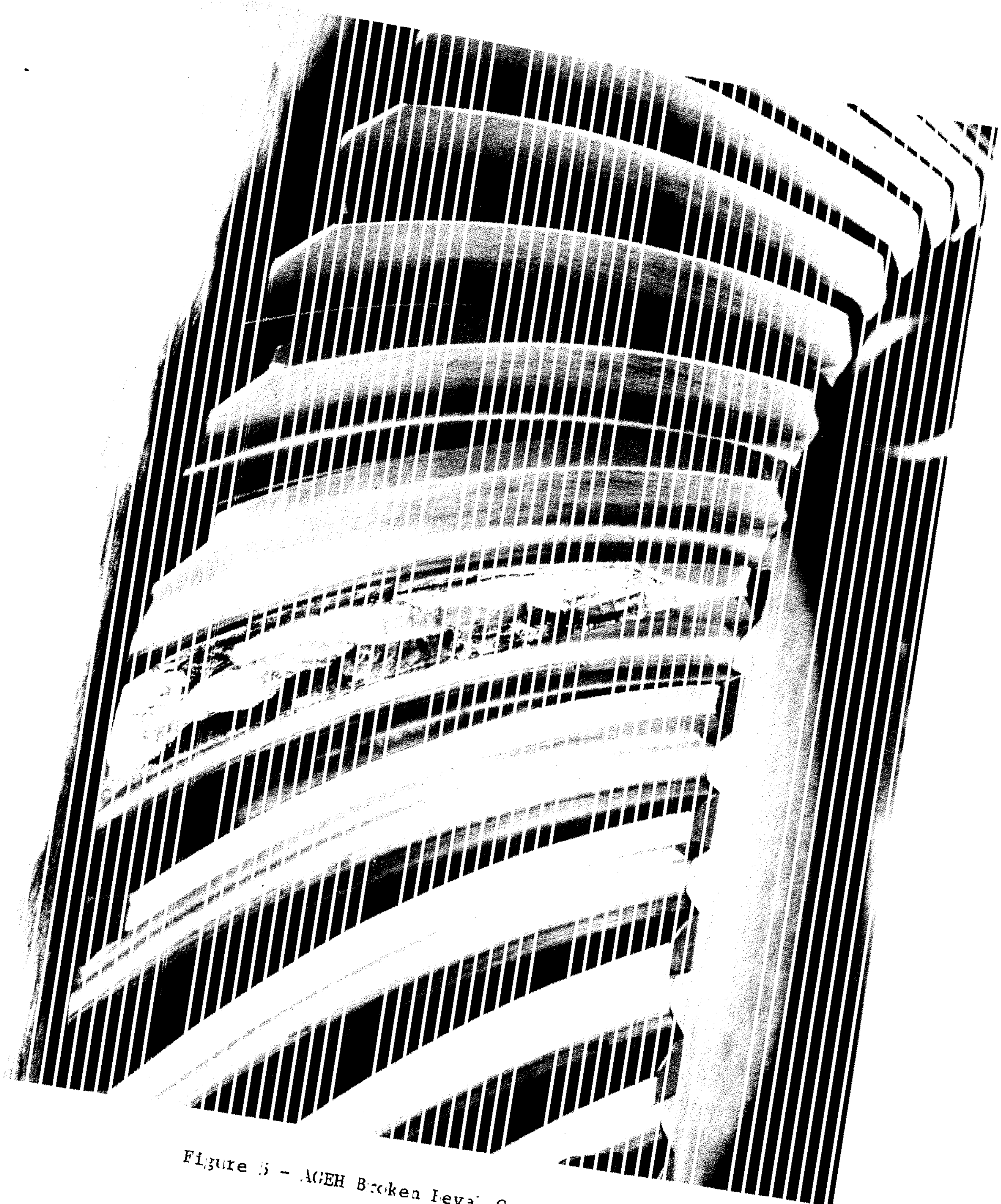


Figure 5 - AGEH Broken Bevel Gear Tooth

rotated about the vertical axis (Figure 5). Further examination showed that another tooth was badly cracked and held into the gear blank only because the crack had not yet penetrated to the surface at each end. A chemical analysis of material taken from the failed gear showed the following percentages of impurities:

Materials	Percent of Impurities
Carbon	0.13
Manganese	0.64
Phosphorus	0.009
Sulphur	0.019
Silicon	0.40
Chromium	1.20
Nickel	3.30
Molybdenum	0.10
Vanadium	0.03

Although these impurities would have been acceptable for aircraft quality gears, they were considered unacceptable for the heavily loaded design under test.

The two fractures were very similar and are examples of "case crushing," an uncommon type of tooth failure that is caused by collapse of the carburized case. In effect, the carburized case acts like "thin ice," breaking up under compressive loads (the designed case depth was 0.060 in.); when the case cracks, damage is not limited to any specific area of the tooth profile but extends over most of the addendum and dedendum regions. The subsurface cracks apparently approach the tooth surface normal to the tooth profile.

Based on the two failures, Willis[5] made the following recommendations for producing a new set of gears:

1. Change gear material from AISIE9310 to CEVM 9310, the result being a tighter specification of the chemical properties of the steel.

2. Improve fiorging techniques in the gear blank forming process.

3. Increase the case-hardened depth, after finish grinding, to 0.100-0.120 in. to strengthen the areas where maximum subsurface stress occurs.

4. Increase operating backlash to develop optimum tooth contact under rolling torque load.

5. Increase the tempering temperature to 305°F and holding time temperature to 4 hr after case carburizing; this is done to relieve residual stresses.

In July 1964, the CEVM AISIE9310 gears were ordered from Gleason Works and a year later, in June 1965, tests were started again. In April 1966, having successfully met the contract test requirements of 250 hr at 100 percent torque load and 100 hr at 115 percent torque overload, these tests were completed. Figures 6 and 7 show teeth patterns of lower starboard gears, tested at various load conditions, prior to installation in PLAINVIEW.

The gears were inspected by the magnaflux process; there was no apparent failure or distress noted in the teeth. However, after the last magnaflux inspection, at the end of the 100 percent load tests, observations showed that gear webs had developed longitudinal cracks; they traveled from bolt hole to bolt hole (Figure 8). Twelve cracks in the web flanges were found between bolt holes, and one crack not located at a bolt hole.

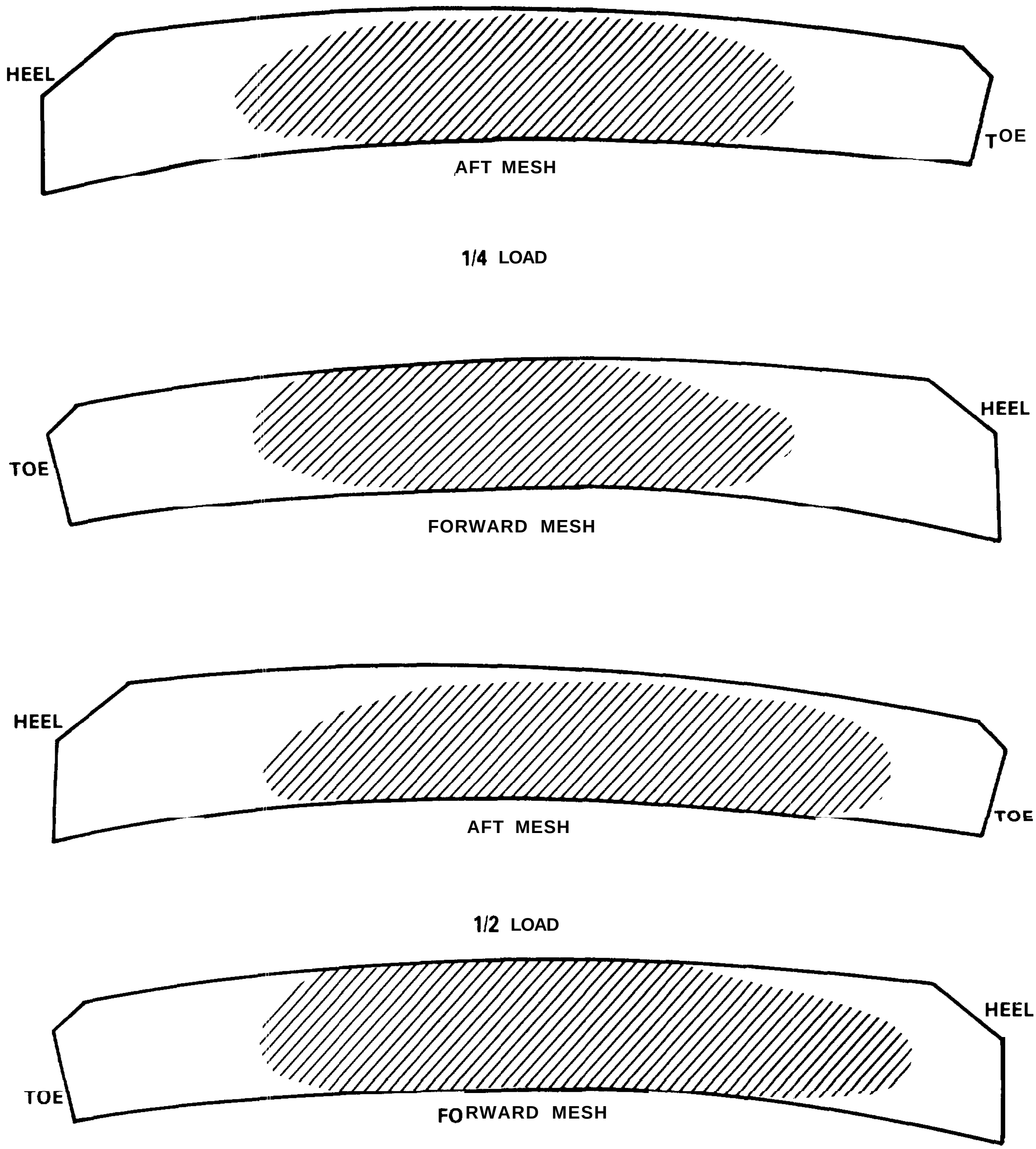
According to the gear inspection report,[3] test gears underwent the following loads when the flange cracks were discovered:

Torque (lb-in.)	Horsepower	Hours	Cycles
320,500	17,500 at 1720 rpm	250	25×10^6
366,285	20,000 at 1720 rpm	100	10×10^6

It was believed that the cracking was associated with fretting corrosion: fretting had been observed after all the first three development test runs. In an attempt to minimize the effects of fretting, lubricant was added to the shims in the gear-shaft assemblies. The web and hub mounting areas also showed signs of fairly intensive fretting.

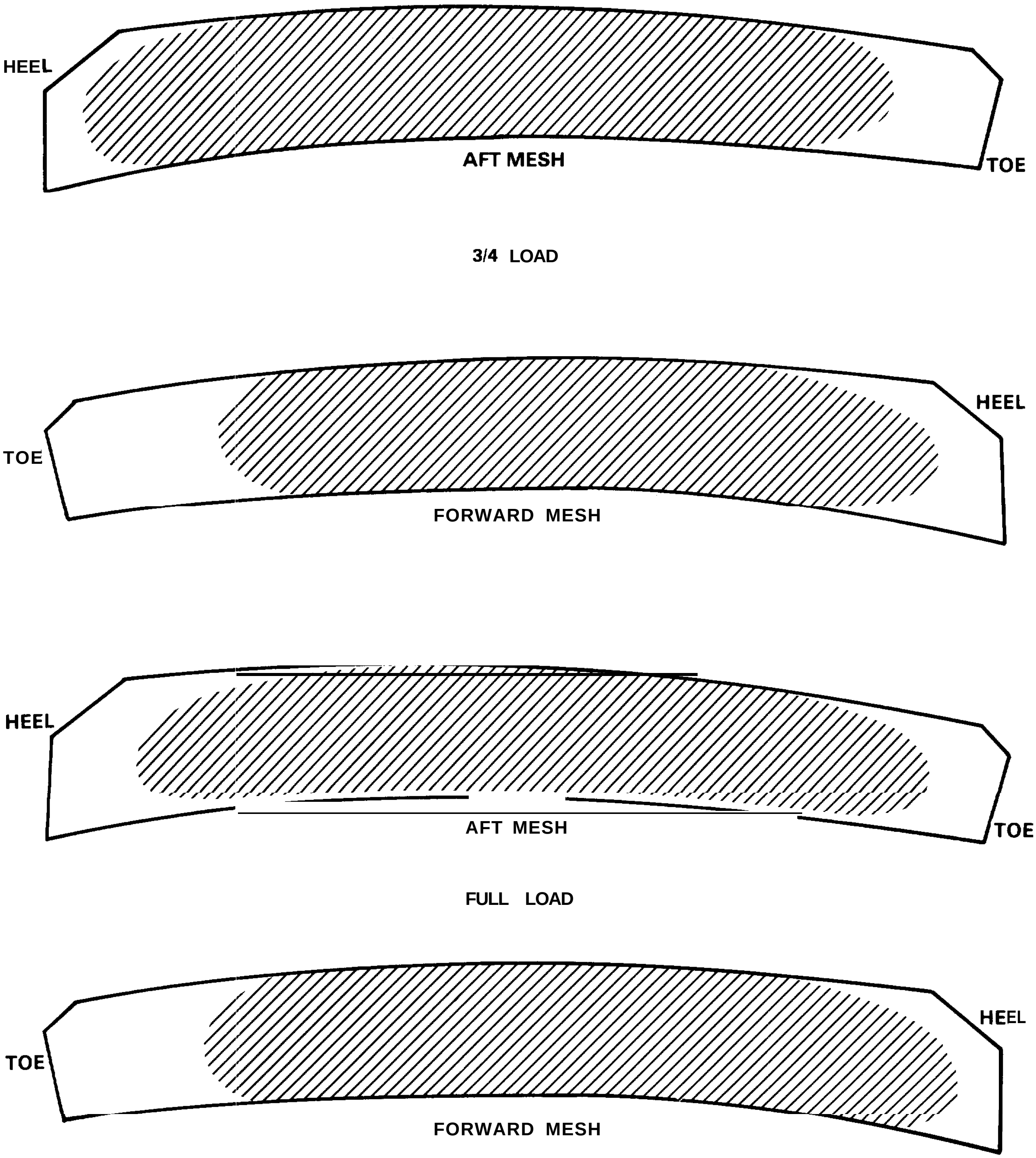
Fretting corrosion is known to cause cracking of steel at stress levels as low as 8,000 psi, regardless of the physical properties of the material. In general, fretting has been a common problem in lightweight high power transmission systems wherever two surfaces are bolted or pinned together and exposed to high vibratory energy.

Subsequent analysis, reported by Smith,' indicated that the flange face fretting and cracks were at least partly due to the method of testing. Figures 3 and 4 show the unsymmetrical force reaction tending to bend the output gear



Note: Patterns made before the 1966 bearing failures.

Figure 6 - Tooth Contact Pattern: Lower Starboard Gear Box, at One-Quarter and One-Half Load



Note: Patterns made before the 1968 bearing failures.

Figure 7 - Tooth Contact Pattern: Lower Starboard Gear Box, at Three-Quarter and Full Load

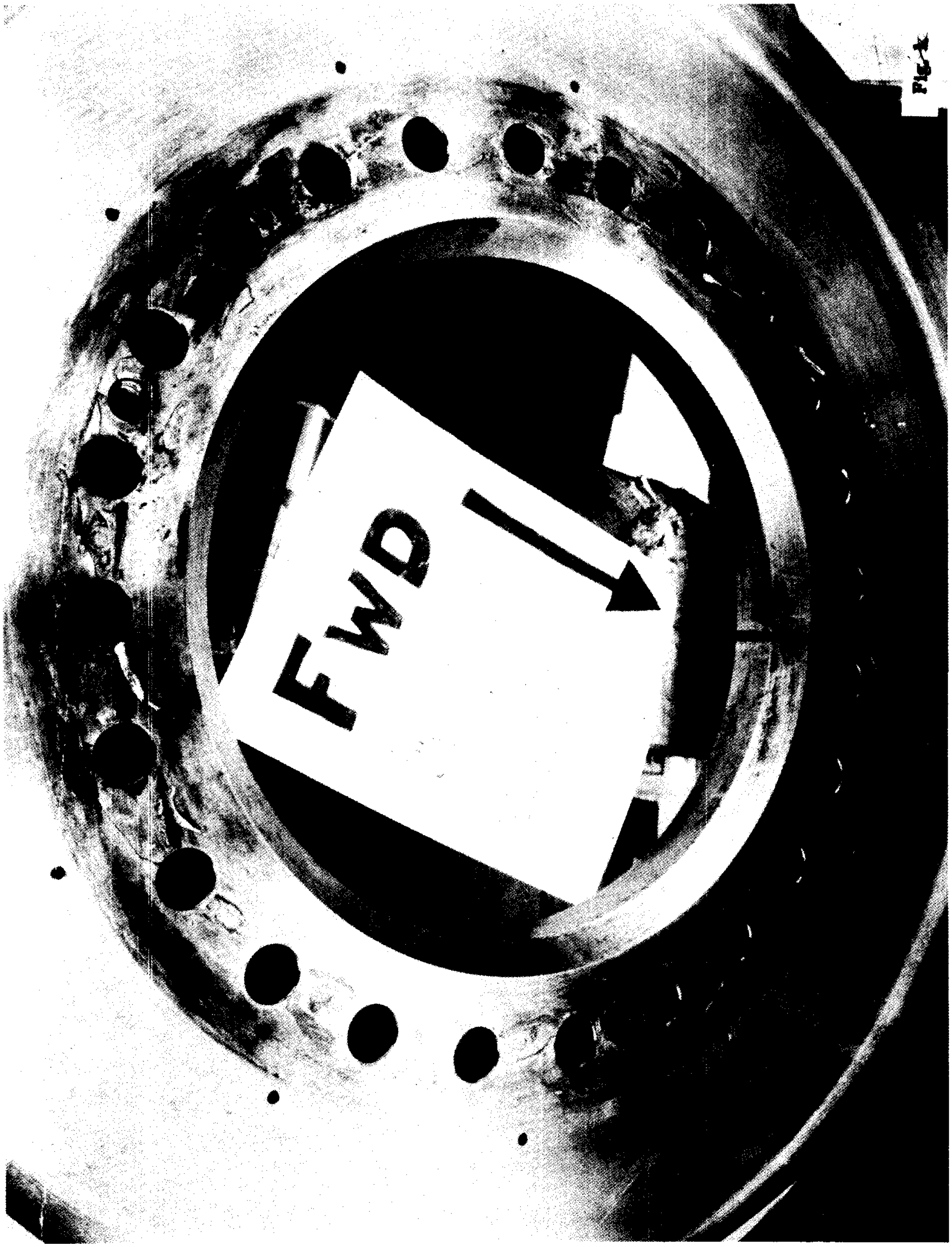


Figure 8 - Test Gear with Cracks in Gear Web

flange and gear assembly. This force pattern, which can be cyclical, produces a traveling wave as the gears pass through mesh. Small motions of deflection can cause sliding between the gear flange and the shim, thus causing fretting. It was surmised that the ship installation and operation would not produce such a similar traveling deflection wave through mesh; it was assumed that the forces would be symmetrical with no resulting deflection.

According to GE, production schedules for PLAINVIEW equipment made it necessary to interrupt development testing of the CEVM gears in spring 1965. This interruption occurred, however, after the case-crushing failures and, thus, gears identical to the CEVM type were delivered to the ship. The gears were delivered before the completion of the 350 hr of test. GE originally requested that development work begin a minimum of 18 mo ahead of the ship hardware contract. Contract personnel awarded these contracts only 6 mo apart; this resulted in parallel procurement and manufacture of both test and ship hardware. As a consequence of the long development and procurement cycle of a new complex system, this contract scheduling hindered development of the CEVM gears. Thus, shipboard gears were fabricated and delivered to the ship before testing was completed and, thus, before the problem of flange cracking even became apparent.

The later discovery of the mounting flange cracks led to a series of stress and deflection analyses to determine the source of alternating stresses which could cause such cracking. Analyses, however, indicated low stress levels relative to the material involved.

It was, therefore, decided to instrument a gear and pinion assembly with strain gages and then apply a torque load to the system to obtain actual stresses for a more accurate comparison with the material capability.

One of the test gear assemblies from the earlier tests was instrumented. Both faces of the mounting flanges of one pinion and one gear were instrumented with rectangular rosette (three-element) strain gages located above and between alternate pairs of bolt holes.

The gear assembly was mounted in a special test stand and torque applied in both parallel attitude (normal loading for ship installation) and opposed attitude (ship torque test condition) at the following percentages of the assumed cruise torque value 314,330 lb-in./mesh: 25, 50, 75, 100, and 115 percent.

The strain gage tests, like the analyses, showed low stress levels; this could not explain the cracking which was assumed due to high stress levels. However, fatigue cracking requires cyclic stresses which apparently were associated with the conditions created in the third load test. This also is believed to have produced the interstice fretting that was observed. As previously mentioned, fretting corrosion is known to cause cracking in steels at low stress levels (reported to be as low as 8,000 psi). With this circumstantial evidence, it was concluded that periodic inspection of the gears for evidence of fretting corrosion would be required during the life of these gears so that any problem could be dealt with immediately.

In July 1966, NAVSEA and NAVSSES representatives visited GE to inspect and discuss the cracked test gears since PLAINVIEW was scheduled for trials early in 1967.[6] (Trials using the foilborne transmission actually started in October 1967.) At the time of the aforementioned casualty, GE had one ship strut transmission available. Of immediate concern was whether or not this strut transmission should be delivered to the ship or be held up for modifications which could then result in further delay.

After a detailed inspection and discussion with GE personnel, two alternative proposals were suggested:

1. Construct the gearbox to be perfectly rigid, thereby eliminating relative motion which was the cause of the fretting.

2. Accept the relative motion between mating surfaces and attempt to reduce resultant fretting through other means.

With regard to the present manufacturing capabilities, the latter approach was considered the most adaptable to PLAINVIEW. According to NAVSEA, the proposed modification to the gears would do the following:[6]

1. Replace steel shims between gears and shaft with "fretting resistant" silver-plated bronze shims.

2. Shot-peen mounting surfaces of gears and shafts, thereby reducing tendency to crack by inducing a residual compressive stress.

3. Coat gear flange face and locating bore with Teflon. (This is generally an effective lubricant to retard fretting.)

4. Install body-bound fitted bolts between gear flanges to reduce slippage and movement. (Since the mounting surfaces are lubricated, torque will be transmitted almost entirely by the bolts.)

5. Fill space between gears with damping material such as lock foam and rubber.

It was estimated that these modifications of the existing gears would delay the ship approximately 8 mo.

Based on the 350-hr gear tests, GE advised NAVSEA that the ship could safely operate with the existing or unmodified gears for 250 hr foilborne at full power, after which transmission overhaul "would be imperative."[6] According to NAVSEA, GE was unwilling to guarantee much more than 250 hr of full power operation even with the modifications. It was also tentatively decided, with NAVSEA concurrence, that the unmodified gears would be installed in PLAINVIEW, and a set of modified gears would be manufactured and tested prior to PLAINVIEW completing 250 hr foilborne operation.[7] It is of interest to note that NAVSEA never released funding to build the modified gears.

SHIPBOARD EXPERIENCE AND PROBLEMS

EARLY PROBLEMS AND MODIFICATIONS

The unmodified CEVM gearbox assemblies procured for PLAINVIEW were shipped to LSCC in mid-19865 and installed in the struts; one assembly was spin tested and inspected during this year. Following this spin test, the assemblies remained idle; neither was flushed with lube oil until October 1967 when the foilborne trials began. Thus, for 2 1/2 yr, the gear-bearing shaft systems sat with no protection against corrosion. Several failures of the pod scavenge pump in February 1968 led to a general inspection which revealed salt water had entered the lower transmission unit; this was the result of improper sealing around the propeller shaft assembly after the spin test inspection.

During subsequent short time foilborne operations, salt water entered both port and starboard lube oil systems. An inspection of the gear assemblies in May 1968, triggered by a failure of the pod scavenge pump, disclosed damage to the large roller bearings and scoring of the forward mesh of the port lower gear. Of the 16 roller bearing races in the four gear assemblies, 12 races had broken after less than 10 hr flying. All races had failed in the key slot fillets. In each case, the crack appeared to start at one corner of the race retention key slot and then to propagate diagonally across the race until the race failed in hoop tension. The cracked corner was the one loaded in tension by the key which tried to keep the race from turning. Water corrosion was also evident in all fractures. GE stated that the ship was operated for "some hours" in the hullborne mode under foilborne power and that approximately 50 percent overload was imposed on the transmission system; bearing failures were attributed to these conditions,[4] LSSC, however, in subsequent correspondence took issue with this finding, citing the lack of both power and torque data; nevertheless, LSCC admitted to the "probability" of torque overloads on the transmission. (During inspection, it was noted that all bearing sets which carried broken races displayed slight wear by fretting.)

Shafting and gearing were removed from the craft and returned to the factory where detailed inspection showed:

1. No evidence of fretting.

2. No evidence of cracking after all gear and pinion mounting flanges, shims, and shaft flanges were magnetic particle inspected.

3. Evidence of light scoring of the port-lower bevel gear teeth; however, it was concluded that the scoring must have been due to vibration which itself was the result of the cracked bearings. (The scoring was not severe and several experts advised that any attempt to remove'the scoring would be more harmful than leaving it alone. They indicated that normal operation should polish out the scoring.)

After a thorough analysis of the failed parts, GE proposed the following modifications:

1. Plug the gear shaft stubs (under the bearing race) which had been "bottle-bored" to reduce weight. This would make the shaft a solid section, thereby increasing its stiffness under the bearing race. This would reduce the compressive strain of the shaft and maintain the high interference fits necessary for roller bearing race retention. "Bottle-boring" of the shafts to reduce weight had produced shaft bore contour which generally matched the outer diameter (OD) profile. It was felt that this practice contributed to the fretting problem since shaft rigidity and stiffness was reduced in areas where maximum rigidity is necessary. (Bottle-boring of these forged shafts reduced the weight of each bevel gearbox by approximately 280 lb.)

2. Increase the key slot corner radii from 0.010 to 0.030 in. to 0.050 to 0.080 in. In addition, two radial grooves of 0.250 in. radius would be added to each side of each key slot to relieve stress concentration.

3. Reduce the original fit interference from a nominal 0.0038 in. to a nominal 0.0029 in.

4. Fill with epoxy the key slot area as a seal against water entry into the gearbox; the epoxy would also seal the key assembly against water contact at key slot fillets.

During negotiations between GE and LSCC, NSSC stated they had only been advised of the damaged bearing problems and requested that LSCC provide operational data which might be pertinent to analysis of the transmission system failures and proposed fix. In an interim report, LSCC provided the following information:[9]

1. Total turbine running time

   Port - 10 hr 32 min

   Starboard - 10 hr 40 min.

2. Total foilborne time - 32.5 min.

3. Typical values for thrust torque in various modes of operations (Table 4).

TABLE 4 - TORQUE AND THRUST FOR VARIOUS
FOILBORNE MODES OF OPERATION

Operational Mode	Turbine (rpm)		Torque (in.-kips)		Thrust (kips)
	Port	Starboard	Port	Starboard	
Hullborne	3,200	3,300	81	78	37
Before Flight	2,900	2,800	61	46	20
Take-Off	5,050	5,050	165	148	59
	5,000	4,900	148	122	48
Foilborne	4,700	4,700	120	101	40
Steady State	4,700	4,600	108	98	39

While the data in Table 4 are from a flight on 30 March 1968, LSCC indicated that they were typical of all readings made on several voyages.

Smith[4] made the following recommendations on ship operations and gearbox inspections:

1. Operate the ship for a total of 100 hr at cruise level loading.

2. Remove one unit of spiral bevel gear assembly after 100 hr of operation and conduct a complete inspection for flange fretting, bearing race condition, and tooth integrity.

3. Repeat No. 2 after 500 hr of operation if there is no evidence of failure.

4. Make the following design changes if flange fretting is noticed after 100 or 500 hr:

a. Add thick clamp rings under bolting at each gear and pinion flange to stiffen assembly against cyclic deflection.

b. Change mounting flange bolts from "clearance" to "fitted" type.

REASSEMBLY AFTER MO'DIFICATIONS

In September 1968, SLJPSHIPS 13 called for Navy technical assistance "to establish a knowledgeable Navy basis for evaluation of gear box reassembly and future performance."[8] A team of gear experts from DTNSRDC and NAVSEA assembled at GE (Seattle) to witness the reassembly of the spiral bevel gearboxes, discuss inspection procedures, and obtain experience for future Navy controlled inspections and reassemblies. The sets were being reassembled after modifications had been made to the bearings that incorporated stress relief grooves and insertion of "plugs" in the shafts under the bearing races. During the visit, the two lower gearboxes had been "buttoned up;" one upper box was in the process of reassembly, and one was still to be reassembled. The team concluded that the assembly procedures were satisfactory and made the following recommendations:

1. No-load contact patterns would be obtained during all future inspections since the assembly procedures using no-load tooth contacts patterns are satisfactory.

2. Inspection of the gears should be made after the next 3 or 4 hr of foilborne operation to include a tooth contact check and observation of tooth surface condition and bearing conditions.

3. The technical manual should be expanded to include more complete disassembly and assembly procedures, complete with tooth contact diagrams and methods of correcting poor contact.

4. Temperature and vibration levels should be monitored during operation to supply a baseline for proper operations.

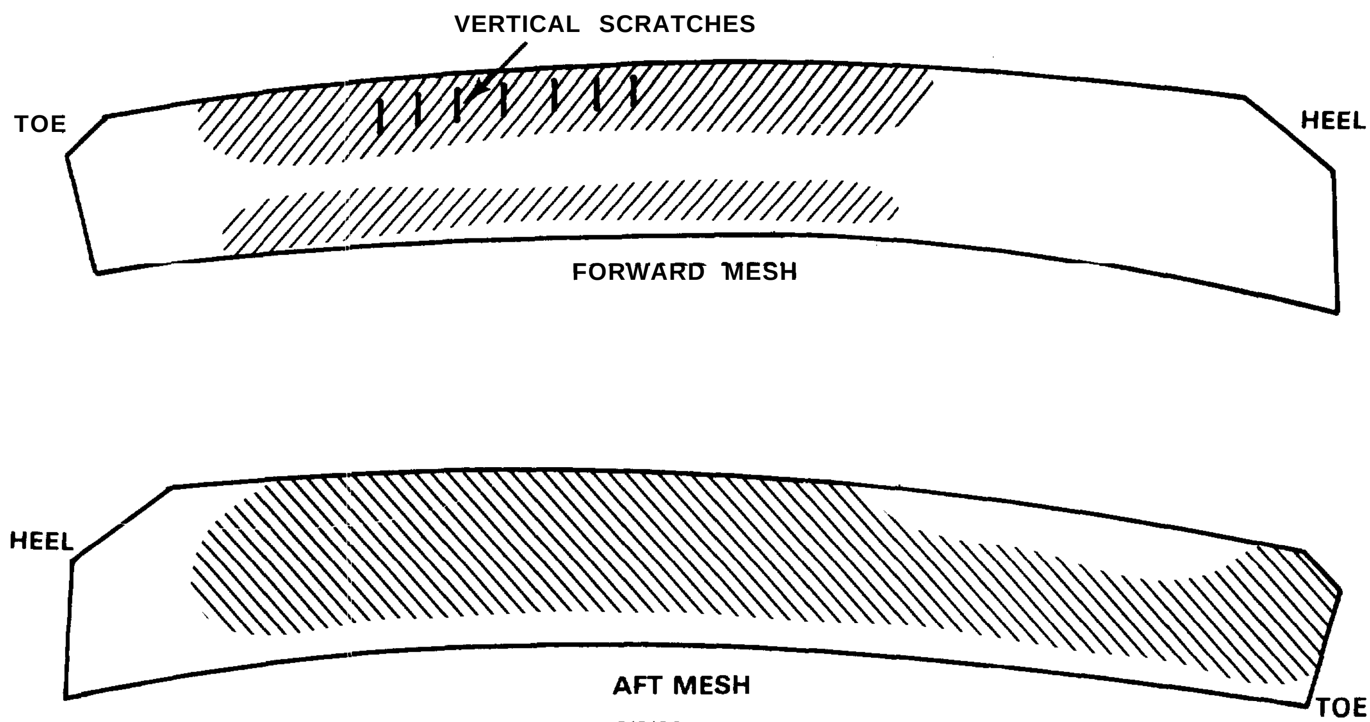
As Samuel[10] and Schneider[11] report, the team concluded that the modifications of the bearing inner races and the "plugging" of the shafts were an improvement, but a close watch was to be maintained for any incipient failures. The team also concluded that the backlash in the gears must be a minimum of approximately 0.020 in. In addition, a long list of technical questions were addressed to both GE and LSCC (Appendix A). The team found that the gearboxes reassembled with the new bearings and plugged shafts did not produce satisfactory contact patterns when the original shims were used to position the gears. To obtain the proper contact, shim thicknesses were changed, thus changing the backlash on some of the gears. However, all the backlash readings were between 0.025 and 0.030 in. although it was noted that stamped on one of the lower gearboxes were the following backlash figures: 0.025 in., 0.038 in., 0.033 in., and 0.046 in.; and stamped on the upper boxes was 0.024 to 0.035 in. It was suggested that the differences might have been dimensional ones among the bearings and/or permanent gearbox distortion.

INSPECTIONS OF SHIPBOARD INSTALLATION

In December 1968, SUPSHIP 13 and GE personnel inspected the PLAINVIEW gears which had operated for approximately 5 hr. An unusual "double wear" pattern was detected on the port lower gearbox (Figure 9). The upper starboard gear showed a high "toward toe" no-load contact pattern which, as reported by Schneider,[13] was the best that GE said could be obtained following installation of the new bearings (from August to October 1968). Figures 6 and 7 show tooth patterns on the starboard lower gears for comparison.

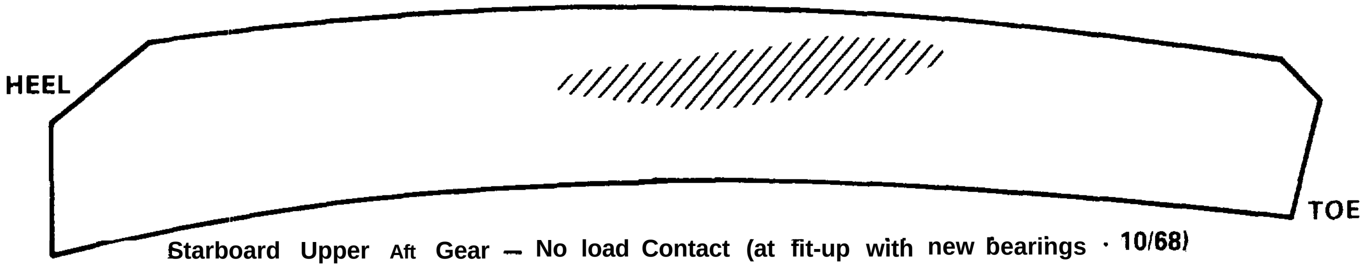
In February 1969, a DTNSRDC technical representative inspected the gearboxes at the request of SUPSHIP 13. Approximately 8 hr of foilborne operation has been accumulated prior to this inspection. The double wear pattern on the port gears showed no increase or deviation from the December 1968 inspection (Figure 10). Apparently, a "polishing" action between mating surfaces was taking place and a "run-in" had been achieved. The starboard gear tooth patterns deviation was not considered serious but for long term operation, a closer scrutiny was recommended.

An inspection of the bevel gears was made in June 1969 after approximately 14 hr of foilborne operations.[12] Three of the four gearboxes were examined, both upper gearboxes (port and starboard) and the lower port box. In both the upper gearboxes, good tooth contact was observed and no wear pattern was visible; thus,

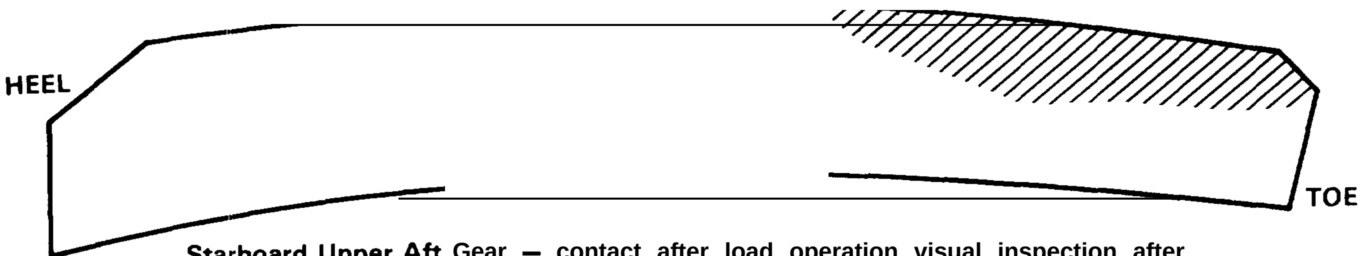


Note: Tape taken from GE during inspection on 12/3/68 after 5 hours operation.

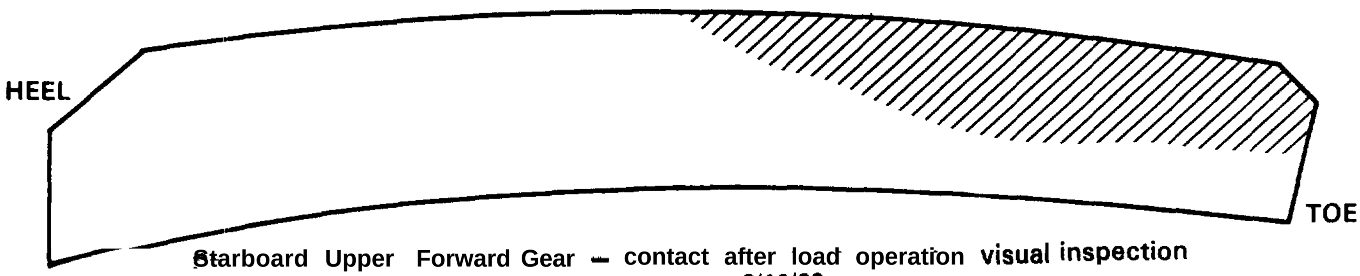
Figure 9 - Tooth Contact Pattern: Port Lower Bevel Gear, at 5 Hours of Operation



Starboard Upper Aft Gear — No load Contact (at fit-up with new bearings · 10/68)



Starboard Upper Aft Gear — contact after load operation visual inspection after 8 hours · 2/10/69



Starboard Upper Forward Gear — contact after load operation visual inspection after 8 hours 2/10/69

Figure 10 - Tooth Contact Pattern: Upper Starboard Aft and Upper Starboard Forward Gears, at 8 Hours of Operation

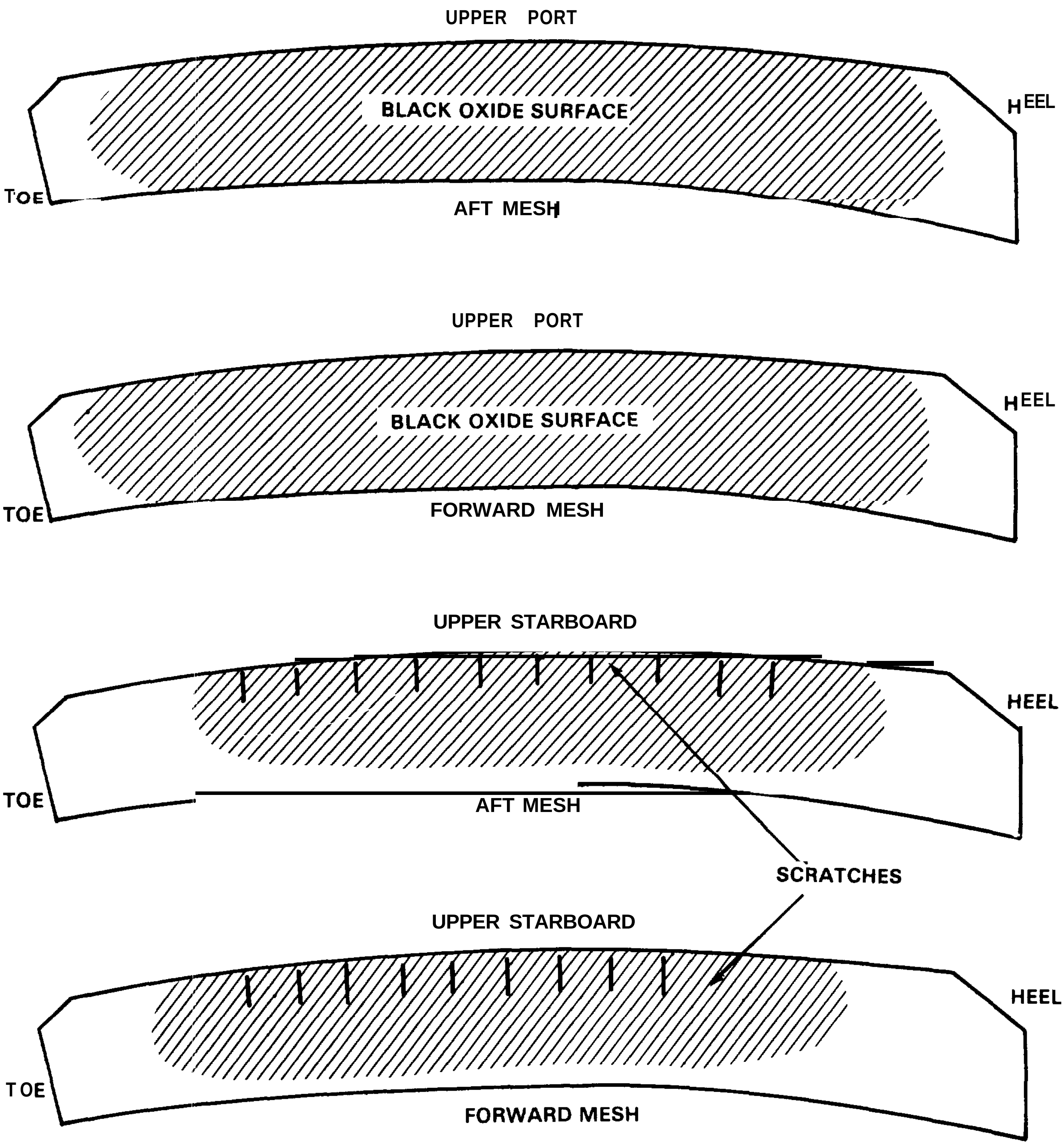
it was concluded that gear operation was in the region of hydrodynamic lubrication (Figure 11). The lower port gear again showed some metal-to-metal contact over approximately 50 percent of the contact surface; this same pattern was observed in the mating pinion. However, there was no progressive deterioration observed. A question was raised about the determination of the proper backlash for the gears. Earlier, GE had attributed some of the test gear failure to improper backlash, thus indicating that backlash was critical. A GE assembly drawing showed that the recommended backlash was approximately four times larger than that recommended by Gleason Works Company. GE was contracted for clarification.

During inspection, the backlash between the forward lower port gear and its pinion was measured at 0.051 in. The assembly drawing called for backlash of 0.024 to 0.028 in. between a production gear and master gear. Since a master gear has a theoretical tooth thickness which usually cannot be duplicated exactly in a production gear, the backlash in assembly of production gears may be 0.048 to 0.056 in. A GE technical representative was also present and gave the gears a "clean bill of health."

During this same inspection period, measurements were made on the transmission drive shafts (both port and starboard) between the bearing pedestal and the SI gearbox. There appeared to be a large discrepancy between the port and starboard shaft alignment with the starboard shaft indicating greatest misalignment. Maximum design and installation misalignment was 0.130 total indicator reading (TIR). While the port side was within design limitations, the maximum measured on the starboard shaft was 0.331 TIR. There is no record to indicate what action was taken to resolve the misalignment.

In October 1969, DTNSRDC proposed an inspection procedure for the PLAINVIEW bevel gears.[14] The procedure was based on the assumption that fretting corrosion had caused failure in the early test gears and that the same could occur to the operational gears. Although ship gears had, until that time, only accrued approximately 14 hr, it was considered important that a close watch be kept for any indications of fretting corrosion.

Because of the potential problems caused by gear fretting, as outlined by GE prototype tests, NAVSEA was contacted about the feasibility of obtaining spare gears under warranty provisions of the procurement contract.[7] In October 1969, NAVSEA proposed two possible courses if a high probability of gear failure existed:[15]



Note: Inspection after 14 hours of operation • 6169.

Figure 11 - Tooth Contact Pattern: Upper Port and Starboard Gears, at 14 Hours of Operation

1. Inspect gear webs before or upon reaching 250 hr of foilborne operation. In the event of a deficiency, the warranty provisions would be invoked and replacement and costs made contractor (LSCC) responsible.

2. Initiate procurement of a replacement set without conducting an inspection. (Since this procurement would have to be paid for out of limited R&D funds, this action was never taken.)

The next inspection took place in early April 1970, after the craft had accumulated a total of about 25 foilborne hours, It was also determined that until then the gear system had been operating only at about 60 percent of maximum power. Taking into account these facts and the short period of time since the previous inspection, no great changes in gear tooth pattern were expected.

The double contact pattern on the forward mesh in the lower port gear appeared to be the same as previously seen. This indicated that the combination of hydrodynamic and boundary lubrication was capable of maintaining the load imposed on the gears. Except for the lower port, no metal-to-metal contact was apparent on any teeth, and the black oxide was still present (Figure 12). It was estimated that the average contact surface was about 85 percent of the available surface with the center shifting toward the toe, except for the lower port forward and aft pinions and the starboard lower forward pinion where the contact pattern was running off the toe. However, it was concluded that since the average contact pattern covers so much of the tooth surfaces, adjustments were not necessary at this time and that at the next inspection period, backlash adjustments would be made if the contact patterns were changing noticeably.

During this period there was an opportunity to inspect the helical gears in the SI gearbox. On both gearboxes, port and starboard, the contact pattern was clearly visible along the whole tooth face width. In the port box, the contact was concentrated in the pinion addendum while on the starboard gears, the portions of pinion addendum and dedendum showed contact pattern. It appeared that oil film thickness on these gears was less than that of the bevel gears. The SI gears appeared to be functioning properly. No teeth patterns were recorded; visual observations appeared to suffice.

As reported by Csaky[16] a visit by a DTNSRDC representative to both GE and Gleason Works was made in May 1970 to discuss past and present features and modifications to PLAINVIEW bevel gears and to discuss the possible manufacture of

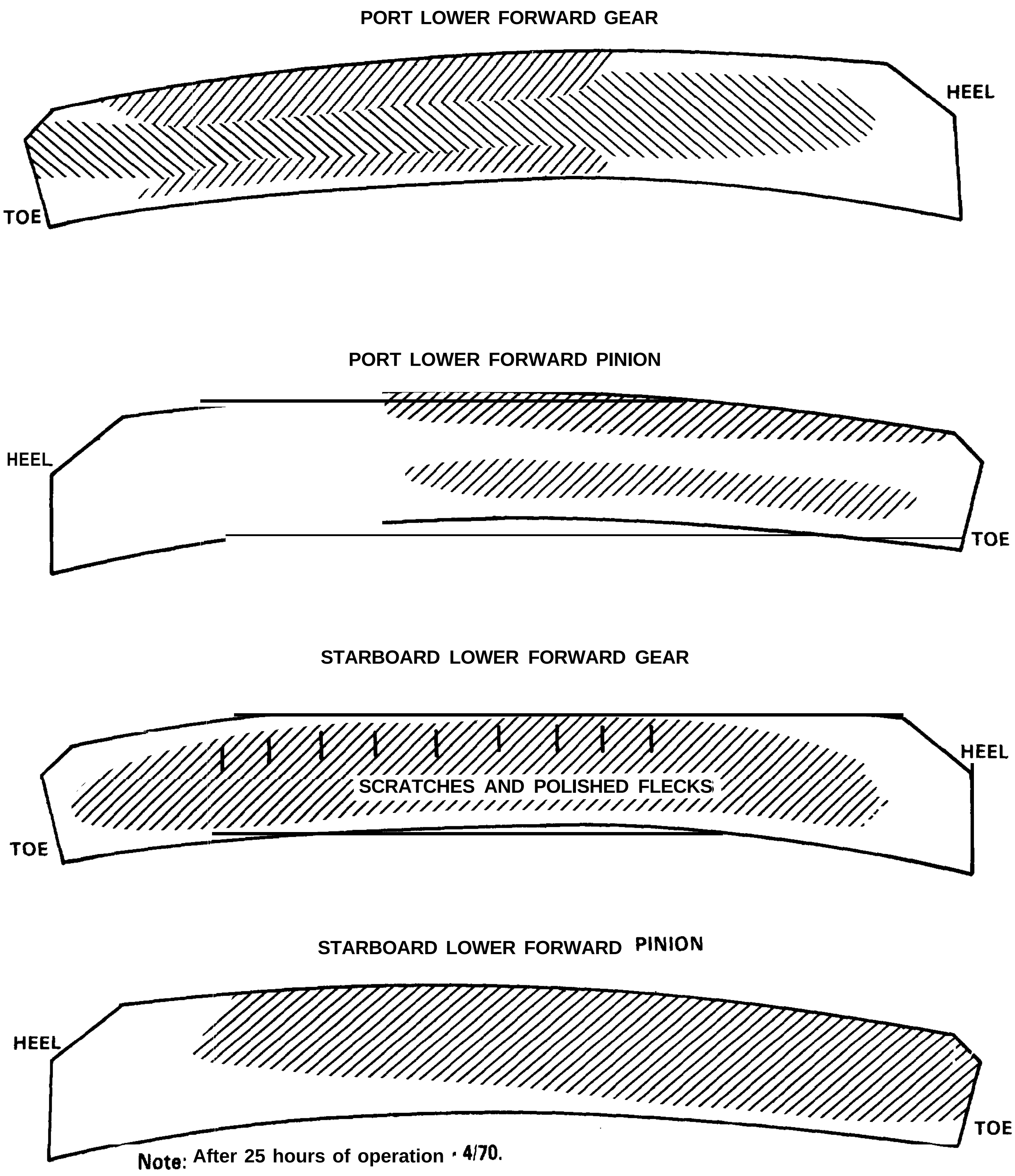


Figure 12 - Tooth Contact Pattern: Port Lower Gear and Pinion, at 25 Hours of Operation on April 1970

spare gears. Gleason Works, producers of the basic gears (tooth geometry, gear cutting, grinding, and testing), emphasized again that the proper contact pattern under no-load is the prerequisite to satisfactory operational conditions and is even more important than backlash. It was stated that, at first, a gear is ground to have a central toe contact pattern; afterwards it is subjected to deflection testing under 1/4, 1/2, 3/4 load, and full load. Each time, the tooth contact pattern is observed and photographed. With increase in load, the contact pattern spreads toward the heel (Figure 13). If the contact pattern at each of the mentioned loads is not satisfactory, the gear is corrected by grinding. The correction is made by using different diameter grinding wheels. When a no-load pattern on a corrected gear is taken, the resulting pattern may be different from the initial one. However, this latter pattern should be used for the assembly and reassembly operations. Typical contact tapes taken after gear corrections were made showed the length of the contact pattern to be about 1/3 of the total face width and a shift off the width center toward the toe.

Discussions were held in May 1970 with GE representatives who indicated that the minimum backlash should be 0.025 in. with "no" maximum value. DTNSRDC claimed that the instructions in the Technical Manual on Gear Reassembly No. 4-2-5-2, Item 24, issued by LSCC were too general and, therefore, not adequate. GE agreed but claimed that they had worked out the reassembly procedures in detail and had provided sketches of contact patterns and complete instructions to LSCC. LSCC quoted that because of "disagreements with regard to cost of editing an amendment to the existing Technical Manual," the Manual was not issued in the form proposed by GE. GE forwarded to DTNSRDC a copy of their instructions which had been sent to LSCC; a comparison with LSCC instructions revealed that the latter had been condensed and failed to include the sketches of contact patterns. These are an important feature of the gearbox reassembly, and they should have not been omitted.

During these same discussions, the subject of fretting corrosion of the gears disclosed inconsistencies and contradictions about the causes of corrosion. The conclusions of the Bevel Gear Development Test, as reported by Smith[4] in 1969, emphasized the fact that fretting corrosion could only occur in the test gear configuration. In May 1970, GE altered its position and concluded that various conditions could contribute to the development of fretting corrosion under actual

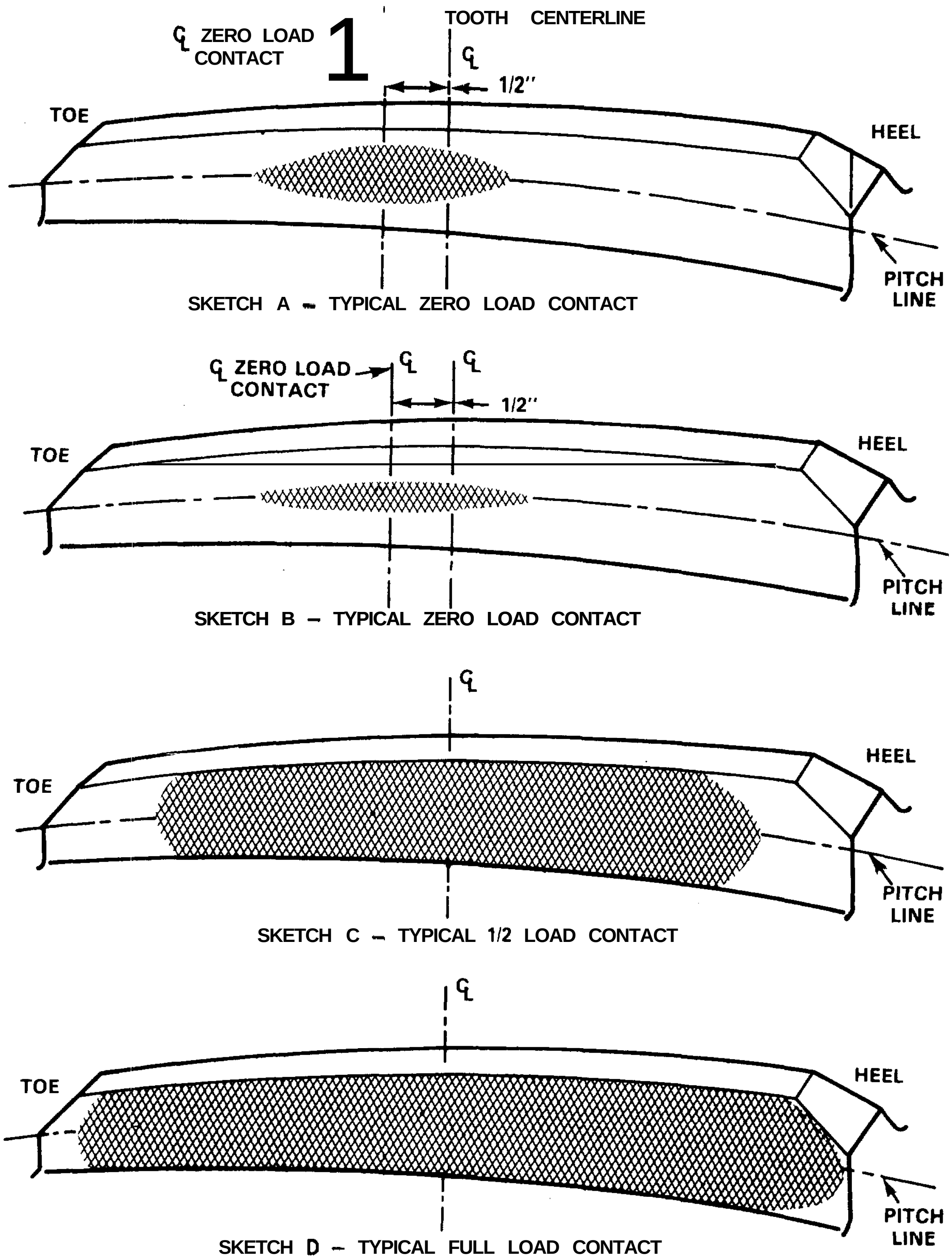


Figure 13 - Tooth Contact Patterns: Standard Bevel Gear Under Different Loading Conditions

operational conditions. As reported in Csaky,[16] GE and DTNSRDC agreed that the following considerations, modifications, and changes would improve the life of future gears:

1. Incorporate fitted bolts in gear mesh assembly to prevent "slippage" or oscillation between gear and gear flange.
2. Increase surface finish smoothness of joint faces.
3. Increase thickness of gear web to increase natural frequency.
4. Maintain close control of bolt tension in installation.
5. Provide better distribution of bolt compressive stress over contact area.
6. Electroplate areas subject to fretting corrosion action.

Gear Box Inspections

On 21 July 1971, the foilborne transmission lower starboard gearbox was opened for inspection.[17] General observation showed sea water lying in low places within the housing; rust was beginning to form in a number of places such as the aft ring gear, bearing outer race, and bolt heads. The bearing ball and roller paths, however, showed no signs of rust. Fresh oil was poured on all gears, bearings, and shafts within the gearbox to help flush out the water and reduce corrosion. Lube oil samples taken from the oil tank showed no sea water present. Gear contact wear patterns for the starboard lower gear revealed little change from the April 1970 inspection.

The next inspection of the bevel gears took place in September 1971, after a total of 33 foilborne hours has been accumulated.[18] This was an unscheduled event which took place because of sea water flooding the starboard strut-pod after a voyage on 8 September. (A scheduled inspection had been set for after 35 foilborne hours.) As reported by Csaky,[18] the foil anchor pin access plate in the watertight area had developed a leak around a fastener. (The bolted access plate was replaced with a welded plate for an interim fix.) The flooding also caused lube oil scavenge pump motor failures in both the lower and upper gearboxes. The upper box pump failure'resulted from flooding of the upper gearbox when the struts were retracted. It was feared that the water had penetrated the lower gearbox during this flooding; this proved to be true. When the lower gearbox was opened, several pints of water were removed after which the gears were "washed down" with several buckets of oil. The present configuration of the access cover plate for the gearbox is difficult to seal. It is virtually impossible to construct an effective sealing gasket which must contain 36 bolt holes and be over 6 ft in length and approximately 1 1/4 in. wide (Figure 14).

It was found that several teeth were spotted with a light layer of rust which could be easily wiped off. All bolt heads were also lightly rusted. On the outer race of the roller bearing, on the vertical shaft, and on the face of the bearing cage a layer of rust was clearly visible. The water was removed from the casing and the teeth were cleaned thoroughly. The bolt heads were degreased and sprayed with a zinc primer. Fretting corrosion on the gear flanges and bolt surfaces was suspected. Because it was practically impossible to check for flange face fretting, one bolt was removed and inspected. Microscopic spots of corrosion were detected on the bolt body on 120 degrees of arc at about the midpoint of the bolt length (Figure 15). This was proof that fretting was taking place on the flange surfaces.

The condition of gear teeth in the aft and forward gear (lower starboard gear) and on the aft and forward pinion (upper starboard gear) were found to be similar to the condition after 25 hr of operation: no metal-to-metal contact was observed. On the contrary, some spots of Red Dykem on tooth ends still remained, thus confirming the presence of oil film between teeth in action.

The next gear inspection took place in March 1972.[19] This was prompted after water was again found in the starboard strut system after a foilborne operation. (Operational time on the gear system was approximately 50 hr.) The crew observed a noticeable list to starboard after the craft had been on the hullborne mode for about 1/2 hr. When the starboard foil was raised, a vast amount of water drained from the strut. After returning to port, the starboard bevel gear system was drained of more than 300 gal of oil mixed with about 30 gal of water. The gear system was refilled again with clean oil and sprayed by the oil jets to help wash out any remaining oil/water mixture. A second draining produced over 5 more gallons of water so the gear system was filled and sprayed a second time and drained. No water was detected this time.

The bevel gears in the lower gearbox appeared to be in very good condition. There were a few small superficial rust spots on the teeth that were easily wiped off, and some rust was seen on one of the roller bearings in the lower box. Again, only small portions of the bearing race were visible.

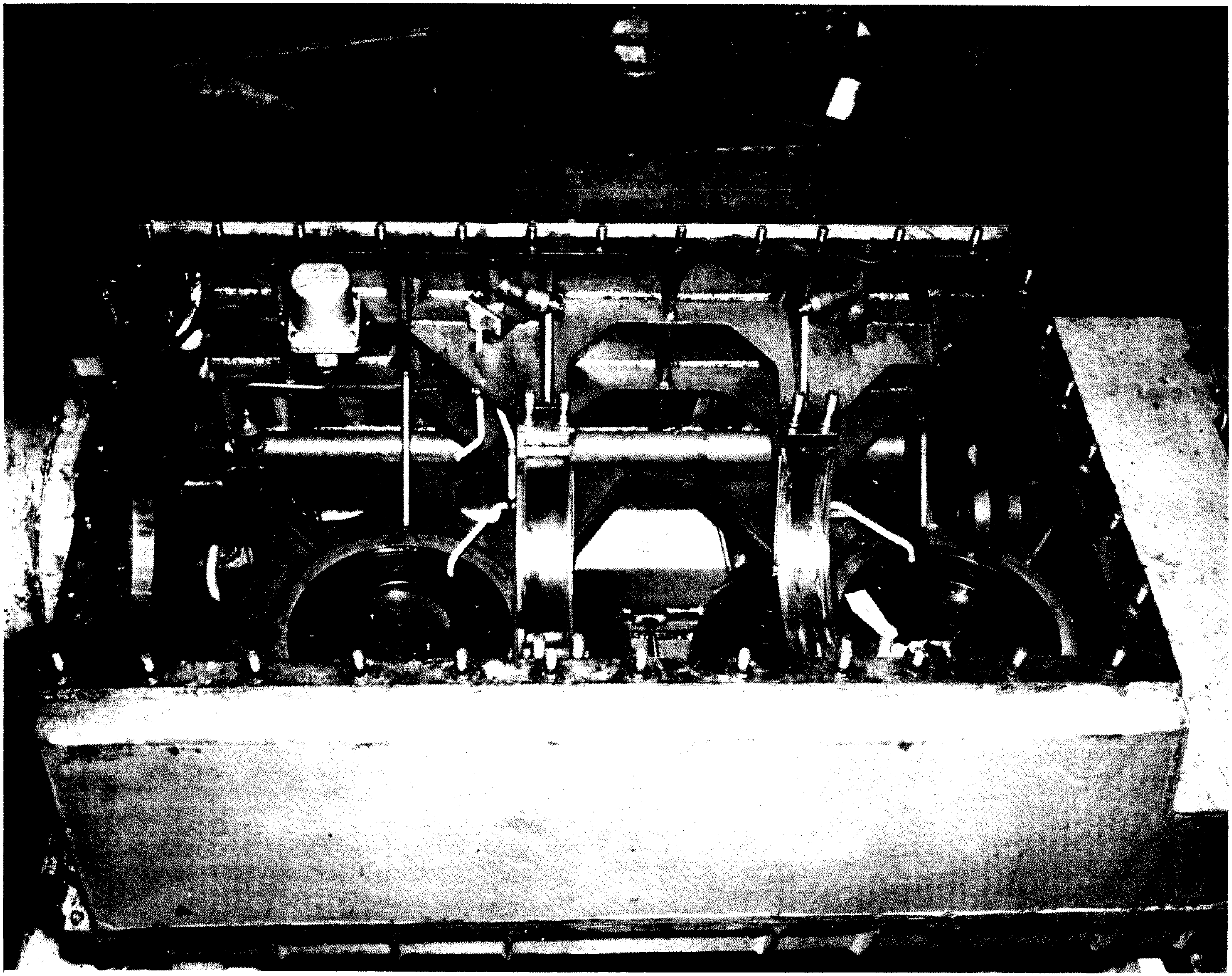


Figure 14 - Open Gear Box

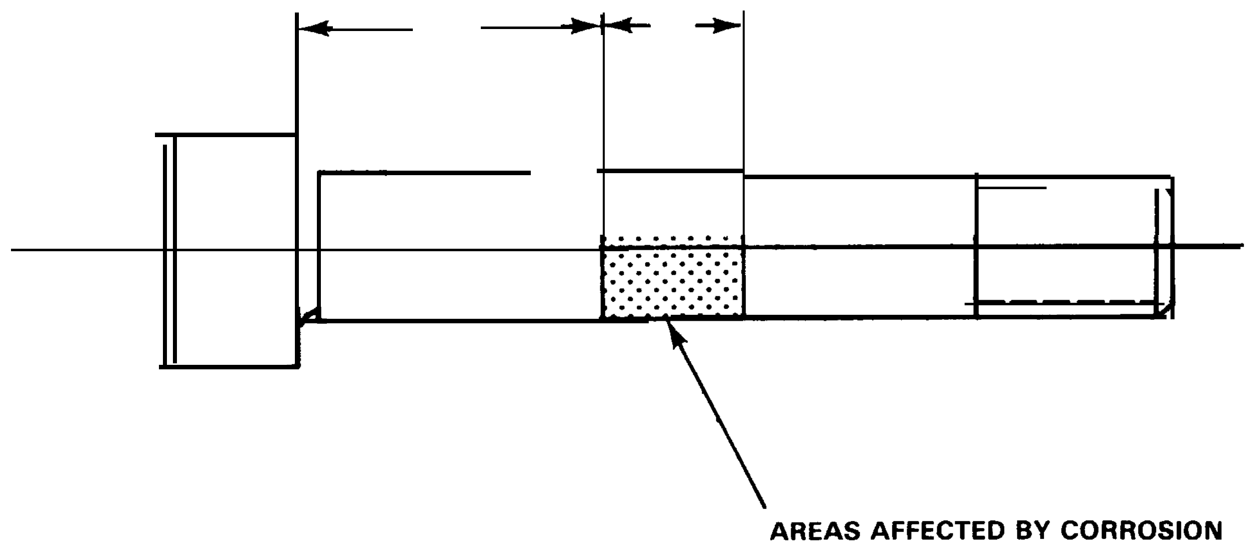


Figure 15 - Coupling Bolt With Corrosion Areas

Water was found as usual in the various compartments formed in the gearbox by the stiffeners. (Note: This is a design deficiency which requires correction.)

Single Idler Gear Inspection

As described in a Boeing memo report[17] and Liljegren,[20] the foilborne transmission systems ST gearboxes were opened in January 1972 for inspection by Boeing and GE representatives. The starboard pinion and idler gears were in excellent condition. The port pinion showed some wear through the zinc phosphate coating to the nitrided case, particularly on the left-hand (aft) helix. Tooth contact patterns were made of both helices (Figure 16). No significant wear on the idler gear was noticeable; however, the finish on the gear teeth appeared coarser than the pinion and the starboard idler gear. The low speed (LS) gear was in good condition. Total foilborne time at this inspection was in excess of 50 hr.

The SI gearboxes were opened again in February 1972 for inspection by Boeing personnel.[21] The starboard pinion and idler gears were in excellent condition, showing no change from the previous inspection in January. The port pinion showed very little additional wear nor was there any noticeable change detected on the idler gear. The LS gear was not inspected. Total foilborne time was approximately 57 hr.

Another inspection of the SI gears was made in March 1972 and detailed by Csaky.[19] A total of 72 hr has been accumulated to this date. The gear teeth showed signs of scuffing with the port gear showing the most wear. The zinc phosphate coating "wearing away" was noted as in previous inspections although the wear did not seem to be serious at this time. It was pointed out that the port SI gears had the root of the teeth on the pinion hand honed to correct an evident error in fabrication. It was recommended by DTNSRDC that a replacement pinion gear for the port SI gearbox be ordered as soon as possible. It was also urged that an inspection of the pinion and idler gears be made after each 10 hr of foilborne operation so that the wear rate of the pinion could be monitored. A sudden increase in wear could cause extensive damage and thus expensive repairs.

In June 1972, the PLAINVIEW transmission system, consisting of the upper port and starboard bevel gears and the SI gearbox, was inspected after 118 hr of foilborne operation. Summaries of the inspection are given by Csaky[22] and in a Boeing memo.[23] All gear sets and visible bearings appeared to be in good condition.

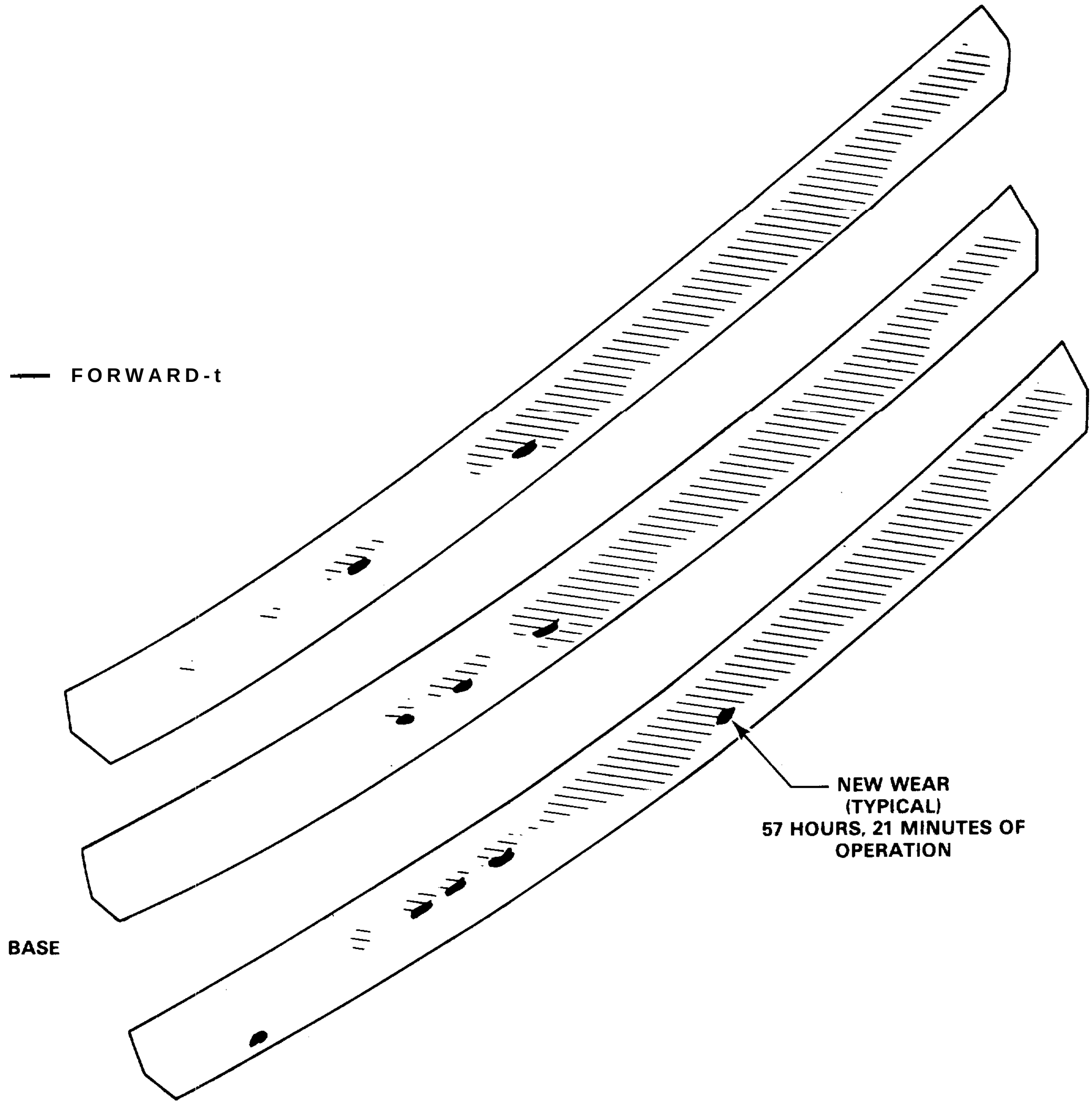


Figure 16 - Port Single Idler Gear Teeth and Wear Pattern
(Left Hand Helix)

The port bevel gears showed signs of metal-to-metal contact. The oxide coating was worn through in a 1/8 to 1/16 in. wide band along the pitch line of the teeth exposing bright metal. A quart of water was trapped in the gearbox in spite of a very low water content analysis in the lube oil. This mixture apparently splashed about during retraction as gear faces had salt spots present which, when wiped away, left a dark mark on the oxide coating. The starboard gears showed signs of past corrosion including surface rust on the edge of several bearing races. The condition of the teeth was good.

The port pinion of the SI gear system showed signs of fine pitting. The Red Dykem applied after the March inspection still filled most of the old fine pits but new fine pitting had appeared. It was noted that this pitting should lessen with time if it represents a normal condition. The wear of the phosphate coating was essentially unchanged.

On the starboard pinion a 3/32 in. diameter chip of coating was missing at the center of each tooth. It was suspected that a piece of hard material had passed through the mesh, producing the noted marks and possibly the recent doubling of vibration output monitored on this pinion. Both the port and starboard output gear showed excellent tooth contact over the load face.

It was recommended that the SI pinion gears be examined approximately every 50 hr of foilborne operation to keep a close watch on progressive or rapid pitting, wear, or coating loss.

The last voyage of PLAINVIEW, prior to being laid up for extensive repairs and overhaul, occurred on 2 January 1973. However, the last foilborne operations had taken place on 11 December 1972 at which time approximately 195 foilborne hours had been logged.

GEAR COUPLING PROBLEMS

From May to August 1968, when LSCC was reinstalling the foilborne transmission after repair to PLAINVIEW, it was discovered that the port and starboard input coupling assemblies, joining the power turbine and SI gearbox, were pitted, corroded, and rusted.[9] This two-piece, three coupling shaft assembly (sometimes referred to as high speed shaft) (Figure 1) is supported midway by an antifriction pedestal bearing and is designed to transmit 17,500 hp at 5,500 rpm. Close inspection of three other similar couplings shows similar problems, but to a much lesser degree; on disassembly of the port turbine shafting (the section from the turbine to pedestal bearing), similar signs of deterioration were observed on the forward coupling. LSCC stated that excessive heat caused the damage; for example, the starboard aft coupling had molten metal extruded from the teeth, leaving deeply pitted tooth surfaces on both the exterior and interior teeth.

At the same time, the Navy notified LSCC that the shaft misalignment exceeded the specification. The port and starboard shaft couplings, located on the SI gearbox shaft and pedestal bearing shaft, were misaligned; the port eccentricity was 0.125 in. and the coupling face run out was 0.027 in., while the starboard eccentricity was approximately 0.250 in. and the coupling face run out was 0.019 in.

Since there were no spare couplings and they were not readily obtainable, LSCC convinced the Navy that operations could be continued with the damaged couplings since the coupling teeth did not transmit enough load per tooth to cause a gross failure, In addition, LSCC promised that tests would be undertaken to determine the cause of failure of the couplings. It was planned to operate the craft for a very limited time (about 1 hr) to obtain data from accelerometers and displacement measuring devices placed at each foundation point. The couplings would then be reopened and inspected. Future corrective action was to be based on the results of -this inspection.

Although the Navy expressed concern over the use of several damaged main transmission couplings during scheduled sea trials, the trials took place in November 1968 and the couplings performed satisfactorily. While foilborne operations lasted approximately 2 hr, there was no evidence of additional wear in the couplings. LSCC stated that further nonroutine inspection of the couplings was unnecessary and, thus, the spline gear couplings would continue to be utilized until replacement parts were received and installed. This would be done subsequent to Preliminary Acceptance Trials.

Alignment measurements were made using extensiometers, accelerometers, and a laser unit.[9] Two kinds of relative movements between the turbine and SI gearbox became evident, a continuous oscillation movement and a fixed movement, the amplitude of which is related to such dynamical factors as applied power, flying height, and rudder movements.

All the accelerometers and extensiometers showed vibration movement: the gross fixed movements were between the pedestal bearing and SI gearbox. The starboard side showed greater vibratory and gross movement than port but not sufficiently different to indicate abnormalities. There were also indications that the SI gearbox, which is mounted to the overhead structure, rotated toward the pedestal bearing when power was applied from the engines, All movements were directly applicable to the result primarily of power application although flying height was responsible for movement too. As both actions normally occur together, it was difficult to determine the percentages of responsibility of each. For one 40-knot flight, it appeared that the SI gearbox and bearing pedestal came together 18 to 22 mils, depending on whether port or starboard side was being measured. By using the laser technique, it was determined that the SI box also exhibited rotary motion about its upper forward edge. Depending on the exact location of the center of rotation, the starboard SI box rotated between 15 and 20 min of arc toward the pedestal bearing.

Some measurements were made to determine the misalignment between the turbine and SI gearbox. During a foil down position, face and radial deflection readings were taken of the relative position of the SI gearbox coupling from the pedestal bearing coupling. In a vertical plane position, face alignment varied by 11 min; and, in the horizontal plane, the misalignment was 8 min. A similar situation existed on the port side. It was further concluded that under flight conditions, the transmission misalignment tends to correct itself from the cold alignment measurements at dockside.

LSCC stated the couplings could have failed for any one or combination of the following reasons:

1. Misalignment in flight.
2. Improper oil supply.
3. Excessive hullborne speed with main transmission power.
4. Coupling sleeve bolts too short.

It appeared that the failure could have been attributed, in part, to poor lubrication. Althoug'h lube oil is sprayed from a nozzle, the spray to the coupling teeth is blocked by two oil retaining rings. The ID of the ring which is located close to the teeth is smaller than the ID of the gear teeth so that the oil cannot be sprayed directly into the teeth mesh area.

The coupling designed by GE may be described as a quasi-flexible coupling because only the tips of the teeth are crowned. The angular misalignment capacity of this coupling was reported to be ±1/4 degree, due mainly to backlash between the teeth.

New GE couplings were installed in the transmission system sometime in late 1968 or early 1969.

In March 1970, after the craft had accumulated 25 foilborne hours, coupling failure again was discovered after excessive vibration had been detected. One coupling located on the pedestal bearing end of the high speed shaft, connecting the engine drive shaft to the SI gearbox (port side), was damaged beyond repair. There was severe surface damage to the hub external teeth and extensive pitting on the internal teeth. Signs of extruded molten metal were clearly visible on several teeth; as reported by Csaky,[24] evidence of the same kind of failure was observed on two other couplings though to a lesser degree.

The inspection indicated that failure was caused by insufficient misalignment capacity of the couplings, The static misalignments required a coupling capable of taking distortion of approximately ±1 degree; along with dynamic misalignments, the coupling capacity should be at least ±1 1/2 degrees.[24]

It was decided to replace the original GE-type couplings with a redesigned coupling that would tolerate larger misalignment and provide better lubrication. Subsequently,, Zurn self-lubricated flexible couplings were ordered for specific locations in the transmission train. These gear-type couplings have fully crowned teeth (in all three planes) and permit angular misalignment up to ±1/4 degree, although the manufacturer claims higher limits. (In some applications, fully crowned splines have been successfully operated with even as much as 3 degree misalignment.) Zurn couplings use a self-contained high viscosity oil lubricant (Lubriplate 8); in contrast, the original GE couplings used a low viscosity gear oil (2110 TH); it was believed that under the high heat conditions produced by excessive misalignment the heavier oil would help keep a film on the coupling teeth. The damaged couplings were replaced with available spare GE couplings until the new Zurn couplings were available.

In August 1971, after 1 to 2 hr of foilborne time, the GE coupling (located at input to the SI gearbox on the high speed shaft) failed. Shortly thereafter, three Zurn-type gear couplings were installed in the foilborne transmission systems, one each at the SI gearbox input in both port and starboard systems and one at the port low speed pedestal bearing position.

On 6 September 1972, both port and starboard Zurn couplings failed after about 107 foilborne hours. Failure occurred in the couplings at the SI gearbox on the high speed shafting. The cause was assumed to be the result of shaft misalignment, which produced failure of the lubrication seal retainers in the couplings and subsequent loss of the lubricant.

The following summarizes actions taken in an attempt to resolve the coupling failures. In July 1971, a two-phase contract was issued to Zurn Industries to (1) review and propose modifications to the foilborne transmission system and (2) design and develop working drawings for the selected concept. In April 1972, Zurn was contracted to supply both high speed and low speed shafting and couplings, with the low speed shafting having a disconnect feature; in this way, the main engines, with their connected hydraulic pumps, could be operated with the propellers disengaged. However, it was discovered that the couplings and shafts would not fit through the strut-attached shaft, and the Zurn design produced excessive overhang weight to the LM 1500 gas turbine along with insufficient axial flexure to accommodate thermal growth.[25]

Diehl and Lundgaard, Inc. (D&L), was therefore contracted, in January 1974, to redesign the high speed coupling shaft system so that the existing GE couplings could be incorporated with the modified Zurn couplings. D&L also established the alignment procedures for the port and starboard transmission systems.[26]

OVERHAUL INSPECTIONS AND MODIFICATIONS

In order to verify the condition of the transmission system and to assure integrity of the drive system during the long overhaul period, inspection of the gears and bearings was undertaken in May and August 1973 by DTNSRDC, Boeing, and D&L personnel. Reports were issued by Boeing,[27] D&L,[28] and Csaky.[29] The disassembly included separating all four struts from the hull and placing them on a barge. The attached propellers could be rotated 360 degrees and, thus, all gear teeth could be examined. It was difficult, however, to inspect gear teeth on the vertical shafts because they were obstructed by structural parts of the gearboxes.

Table 5 tabulates inspection results, comments, and recommendations by the three different inspection parties. Most of the observations and comments are quite similar and consistent.

TABLE 5 - INSPECTION REPORT OF PRINCIPAL GEAR SYSTEMS ON USS PLAINVIEW, 195 HOURS FOILBORNE TIME
(300+ Hours on Entire Transmission System)

	Bevel Gears				SI Gears		Comments/Observations
	Upper		Lower				
	Port	Starboard	Port	Starboard	Port	Starboard	
DTNSKDC (13 Aug 1973)	Backlash measured to he 0.040 in. This is within limits. No pitting or scoring.		Double tooth contact pattern noted on forward mesh. Similar condition noted in 1969.		Tooth pattern contact is good. Brownish deposits seen in gearbox. No pitting or scoring on helical gears.	No excessive coating removal. Some teeth wear starting to show especially on output gear.	All bevel gear teeth in good condition. Some rust spots present which were easily removed. Black oxide coating seen on teeth flanks. Contact pattern covering about 90 percent of total area. Inspection of bevel gears made thru open cover. Recommend complete disassembly of gears and hearings for more detailed inspection. Recommend better surface finish to stbd idler gear by honing. See reference 29 for more details.
Boeing (13 May 1973)	Considerable salt residue in gearbox. Salt spots on teeth. Gearset wiped off and fogged with "523."	No change since last inspection.		Good tooth contact pattern. Only minimal signs of con-o-sion seen.	Moderate amount of water in sump. No corrosion seen. Brownish color noted on gearset. Slight progression in wear of load face coatine on pinion.	Slight random coating breaks. Mixture in sump consists of about 30 percent water.	Transmission systems not in operation for 3 mo prior to inspection. All gearsets and visible bearings in good condition. Last operation of transmission had been with Mobilarama 523 (rust preventative lubricant). Port lower bevel gearbox was inaccessible due to lack of staging. Both output shaft pedestal hearings need replacement because of static corrosion on halls and races. Both sthd gear couplings need replacement because of pitting in center of male crown tooth contact area. See reference 27 for more details.
Diehl & Lundgaard (13 Aug 1973)	Rust spots on teeth. Backlash 0.040 in. Brownish deposits from lube oil in gearbox. No pitting, scoring, cracks.	Rust spots on teeth. Signs of penetration of black oxide coating.	Brownish deposits from lube oil in gearbox.		Pinion/idler tooth contact pattern is good. Output gearbox has rusty look. Fwd hearing running near 190°F too hot. No pitting, scoring, cracks, etc in gear teeth.	Pinion/idler tooth contact pattern is good. Output gear shows about 0.001 in. to 0.002 in. wear along root and at ends of teeth caused by contact with nitrided idler.	All four bevel gearboxes inspected thru open cover on gearbox. In all cases the running tooth contact patterns appear to be 80 percent or better. All contact patterns are acceptable and no corrective action is recommended. Recommends removing and completely disassembling one gearbox for close inspection. Correct stbd SI idler gear surface finish. Scavenge pump drive shaft (in upper stbd gearbox) sheared off. See reference 28 for details.

An extensive series of inspections were undertaken between April 1974 and August 1975; the purpose was to scrutinize the entire transmission system and recommend repairs and modifications for inclusion in the repair and modification contract which was then being formulated.

BEVEL GEARBOXES

One of these inspections, made by D&L on 29 April 1974, involved removing first the port upper bevel gearbox from the strut and then the horizontal shaft. After disconnecting the gear flange bolts, the forward gear was separated from its mounting surface. At that time, the aft gear could not be separated from its mounting surface, apparently because of a tight fit on the pilot diameter.

Inspection showed little fretting or corrosion on the face of the shaft flange, gear flange, or spacer shim. The absence of mounting face fretting on the inspected gear was considered significant since, as reported by Smith,[4] this was the area of greatest fretting damage on the prototype gears. Nevertheless, both the shaft OD and gear ID surfaces showed extensive fretting. It should be noted that this was the first time one gear flange had been removed from its shaft since its initial installation in 1965.

Although only one gear and its mounting surfaces was inspected, there was a suspicion that the other gears were subjected to the same fretting corrosion. This was verified by the 19 December 1974 inspection when the other three bevel gear units were removed from the struts and completely disassembled.[31] The horizontal shafts were removed from the gearboxes and the gears and bearings removed from the shafts. All the shaft inspections displayed moderate fretting on the gear pilot surfaces (Figure 17). The percentage of fretted area on the 16 surfaces varied from 10 to 90 percent with depth of fretting ranging from 0.007 to 0.020 in. The fretting on the base surface of the gear pilot was significantly less severe than on the shaft (Figure 18). On the port lower bevel gears, corrosion depth varied from 0.001 to 0.012 in.. as measured in random areas. Figure 19 shows the port lower gear carrier; fretting corrosion is evident on the gear pilot surfaces.

In January 1975, Boeing made a detailed inspection of the three other gearboxes and reported the following general conditions existing in all four of the gearboxes:[32]

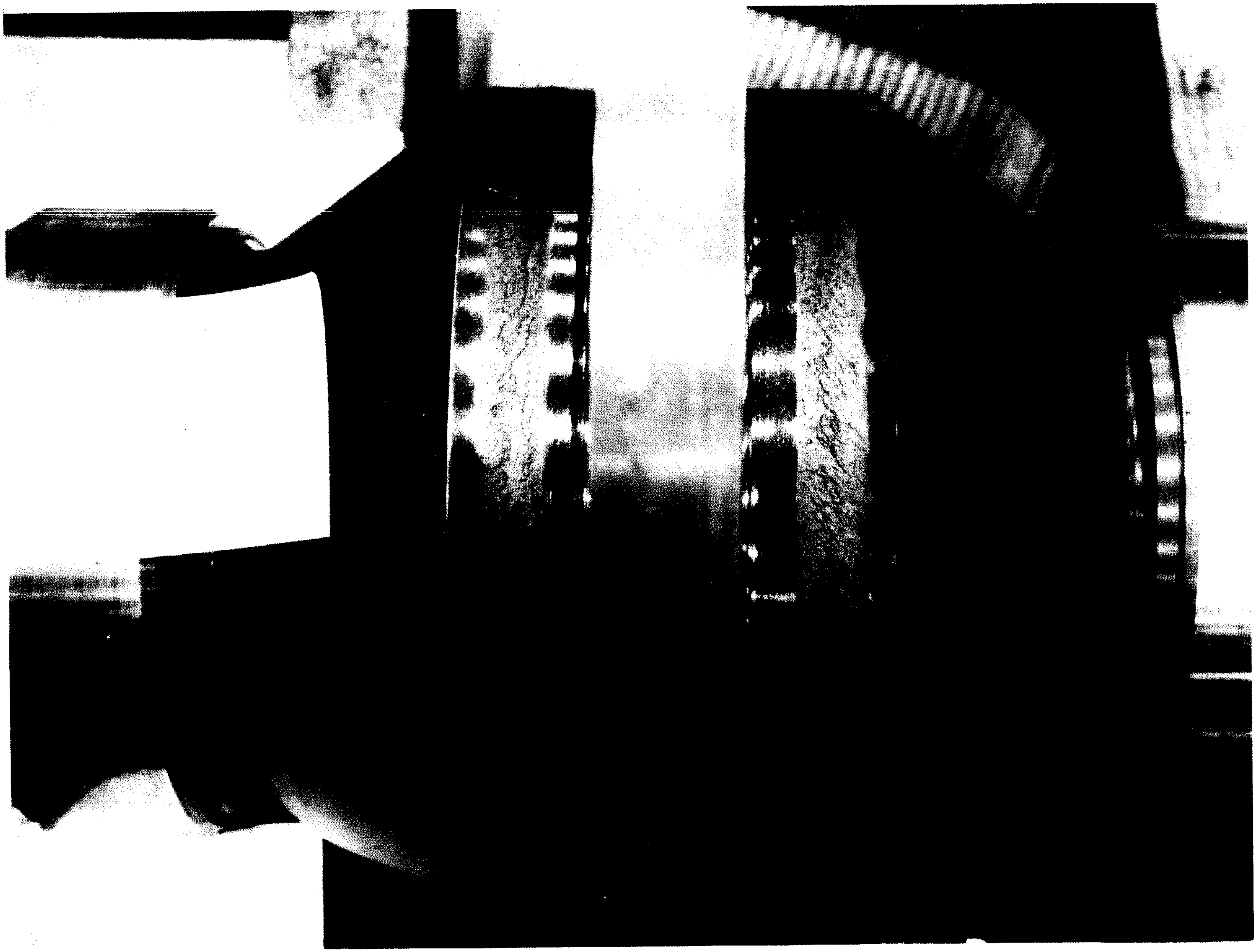


Figure 17 - Upper Gear Shaft with Fretting Corrosion

Figure 18 - Ring Gear Fretting Corrosion

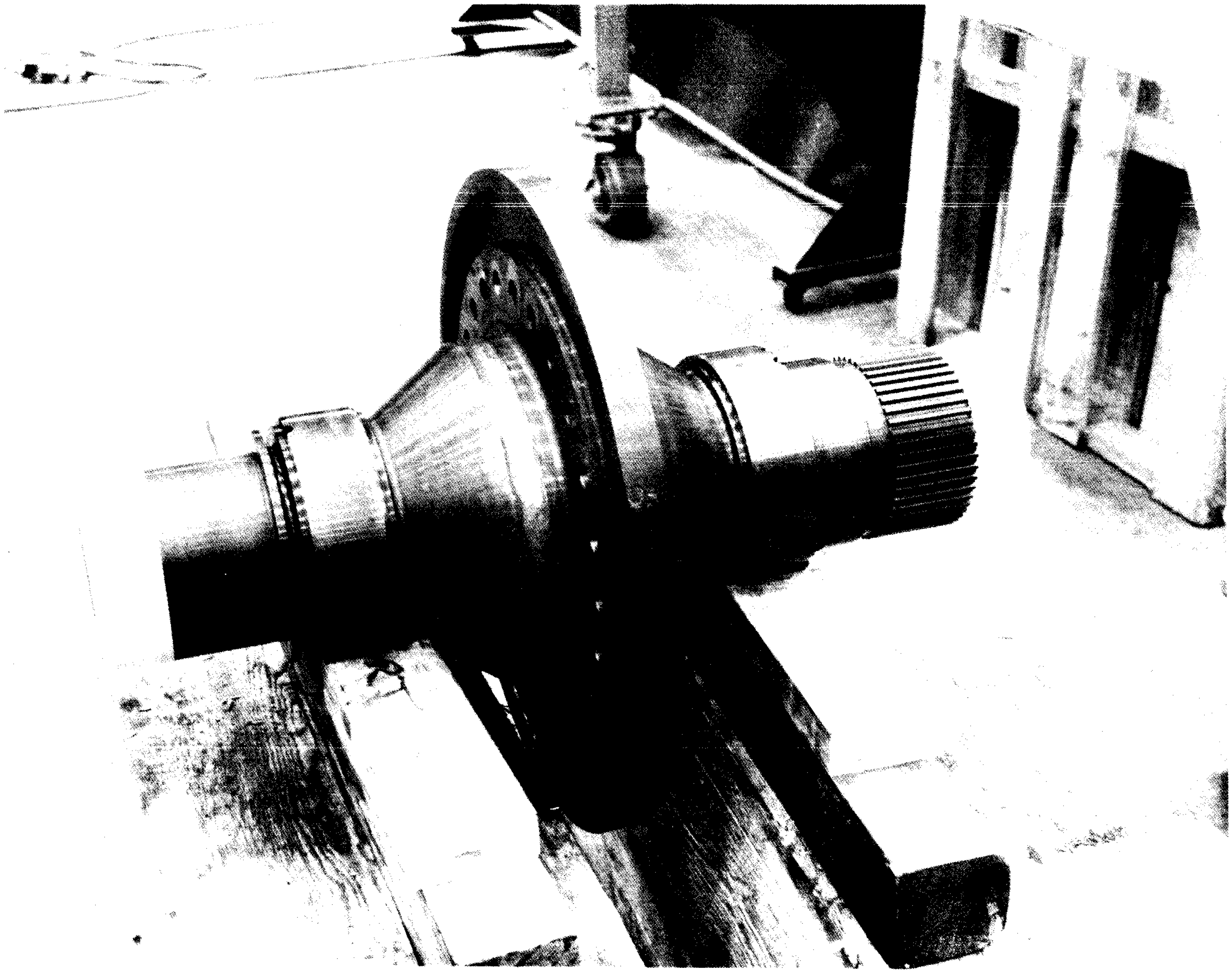


Figure 19   Port Lower Gear Carrier Unit

1. Rust on various parts of the gearbox housing, enough to warrant corrective treatment.

2. Rust spots on most of the bevel gears.

3. Rust on the inner and outer end of the vertical gear shafts.

4. Water in the gearbox housing.

5. Foreign material on various areas of the gearbox housing.

6. Fretting corrosion on the horizontal drive shaft of all four units, at the gear mounting surface and the gear register surfaces.

7. Signs of discoloration, rust, pitting, loss of plating, and oil drain holes plugged with sludge on all bearings.

PROPELLER AND STRUT SHAFT

In addition to the gear and bearings inspections, the propeller shaft system and the strut shaft assemblies were inspected and reports issued by D&L[33,34] and Boeing.[35] Results showed that the propeller shafts and bearings were in good condition with very little work required for their reuse.

As reported by Boeing, in the detailed inspection results and recommended actions, the vertical drive shafts were in relatively good condition considering their exposure to water leakage.[35] The shaft guides suffered severe corrosion, however, with enough pitting in the "O" ring grooves to compromise the seal. Shaft splines showed moderate corrosion while all the gear coupling splines evidenced corrosion pitting and chipping of the teeth at the outer ends. In addition, all the shaft bearings had been subjected to corrosion and galling from lack of lubrication. With one exception, all of the oil holes in the ball seats were plugged with dirt; the oil hole free of dirt previously had its diameter increased from the original 0.18 in. to 0.708 in. The detailed inspection results and recommended actions and disposition for these systems[35] are recorded in Tables 6 and 7.

BEARINGS

Inspection of the large 230 and 234 type roller bearings (over 6.5 in. bore and 12 in. diameter) led to a recommendation to replace or repair all the bearings. There are four 234 type bearings and two 230 type bearings in each gearbox, making a total of 16 of the former and 8 of the latter. In addition, there are 3 duplex ball thrust bearings in each gearbox, making a total of 12. One duplex set is mounted on each horizontal and each vertical gear shaft.

TABLE 6 - INSPECTION OF PORT AND STARBOARD PROPELLER SHAFT ASSEMBLIES

am

	Port	
Propeller Shaft	Minor axial scoring all around forward bearing journal caused during bearing inner race removal.	Hand work to clean up journal as follows: 1. Use fine emory cloth in a shoe-shine, circumferential stroke. 2. Take care to maintain a uniform amount of polish all around. 3. Polish should only remove proud material along scores. Do not attempt to remove all trace of score marks, as this will reduce fit of inner race.
Thrust Bearing	Minor rust stains on outer race.	This bearing is serviceable. No action recommended.
Forward Radial Bearing	Minor rust stains on outer race, inner race, and some rollers.	Considering light load and moderate speed, this bearing is acceptable for further use. No action recommended.
Aft Radial Bearing	Inner race has deep (0.02 in. to 0.03 in.) corrosion bands which correspond to roller contacts all around. Outer race and rollers are badly pitted all over.	1. Replace with new bearing. 2. Determine cause of corrosion and correct.
Propeller Shaft Seal Assembly	1. Inner seal carbon ring has several corner nicks along outer and inner edges. None appear to penetrate all the way across the face, although many have a slight discoloration which may indicate an erosion leak path across. 2. The inner seal runner appears to be in excellent condition with only the slightest wear step. 3. Outer seal carbon ring has only one or two outer edge nicks. 4. Outer seal runner appears to be in good condition	In view of aft radial bearing corrosion, it appears that this seal has been leaking. On this basis, we recommend replacement. However, the inspection shows it to be in better shape than the starboard seal. Therefore, we would also consider overhauling this seal as follows: 1. Re-machine both carbon rings to improve surface finish. 2. Re-machine both seal runner faces to remove wear step.

TABLE 6 (Continued)

Item	Damage Found	Disposition
	Starboard	
Propeller Shaft	Moderate axial scoring all around forward bearing journal caused during bearing inner race removal.	Hand work to clean up journal as follows: 1. Use fine emory cloth in a shoe-shine, circumferential stroke. 2. Take care to maintain a uniform amount of polish all around. 3. Polish should only remove proud material along scores. Do not attempt to remove all trace of score marks, as this will reduce fit of inner race.
Thrust Bearing	Entire bearing is in excellent condition except for some new scattered corrosion stains.	This bearing is serviceable. No action recommended.
Forward Radial Bearing	1. Very minor rust stains on both races. Roilers in good condition. 2. Bore of inner race has moderate axial scores corresponding to those on starboard shaft.	1. This bearing is serviceable except for inner race bore scores. 2. Hand work bore in accordance with instruction for shaft given above.
Aft Radial Bearing	1. Both races and rollers have scattered minor rust stains. 2. Inner race has 2 in. wide heat damaged area which appears to have been caused by excessive use of heat during disassembly.	This bearing is serviceable except for heat damage on inner race. Replacement is recommended if a spare bearing is available. If a spare bearing is not available, consideration could be given to the continued use of this bearing as follows: 1. Check hardness in heat damaged area. If less than 60 Rc, replace inner race. 2. If hardness is 60 Rc or better, re-finish inner race roller path area to a surface finish of 16 nns or better.
Propeller Shaft Seal Assembly	1. Inner seal carbon ring has a few corner nicks at inner and outer edges of face. Inner edge has a 0.06 in. wide worn band all around which corresponds to a hole in the seal runner plating. 2. Inner seal runner has a 0.001 in. to 0.003 in. wear step plus a small hole in plating. 3. Outer seal carbin ring has two small areas of surface erosion on face which could be leak paths. 4. Outer seal runner has 0.001 in. wear step and has several areas of erosion and/or de-plating in seal area.	Replace entire seal, or overhaul seal assembly as follows: 1. Re-machine inner carbon ring face. 2. Re-plate and re-machine inner seal runner face. 3. Re-machine outer carbon ring face. 4. Re-machine outer seal runner face. If surface damage does not clean up, re-plate and then re-machine face.

TABLE 7 - INSPECTION REPORT OF PORT AND STARBOARD STRUT SHAFT ASSEMBLIES

Item	Damage Found	Disposition
Strut Shafts	The shafts are in generally good condition. There are no signs of journal scoring and no deterioration of splines. The black oxide coating appears to have been removed by hand work in a band 8 in. to 10 in. long at each journal location. This was probably done during assembly of the shaft to the struts. Almost all of these areas have some mild corrosion attack, particularly in a band just above the bearing. Since these shafts are extremely vulnerable to loss of fatigue properties from surface corrosion, I believe that the black oxide should not have been removed. If possible, a black oxide coating should be restored wherever it has been removed.	1. Clean all over with solvent. 2. Hand work polish journals using fine emory cloth to remove corrosion stains and to restore surface finish. 3. Investigate possibility of applying new black oxide coating where coating has been removed. 4. Corrosion attack above bearing indicates long term exposure to moisture pockets above bearing. This might be prevented by daily or even weekly use of the lube system pumps to flush moisture away.
Flexible Couplings	All have an accumulation of rust and corrosion products, but have no evidence of significant tooth deterioration.	1. Clean all over with solvent.
Sleeve Bearings	None of the bearings show any sign of heat damage or wear. Several have gouges caused by entry of the shaft splines during installation.	1. Do NOT remove from spherical seat. Clean up both seat and bearing with solvent. 2. Light hand work using bearing scraper to remove any proud metal around gouges. This work should be done by someone experienced with scraping sleeve bearings. 3. Investigate re-assembly procedures which will prevent further damage to bearing surfaces.
Spherical seats	These are generally in excellent condition. Since they are stainless steel, there has been little or no corrosion other than staining from corrosion of outer housing. Several of the spherical seats are very stiff and their ability to self-align is questionable.	1. Clean up all over with solvent. 2. Any seat which cannot be aligned by hand force should be disassembled (male from female) and hand polished to free up its self-aligning feature. Carefully polish male part evenly all over its spherical surface using fine emory cloth in a shoe-shine motion.
Bearing Housings	These are all badly corroded all over.	1. Vapor blast all over to remove rust and old protective coatings. Limit blast exposure on inside machined surface to prevent loss of fit. 2. Re-apply zinc phosphate (Endurion) coating all over. 3. Paint* all exterior surfaces. Oil or grease inside surfaces prior to assembly.
Shaft Housings	These are mildly corroded on outside surfaces with some scattered mild corrosion on the inside surface. The most serious corrosion appears to be at the "O" ring seats.	1. Any shaft cover with significant inside rust should be vapor blasted all over and zinc-phosphate coating re-applied. Paint* as below. 2. All other shaft cover's can be treated thusly: • Solvent clean. . Wire brush outside and "O" ring seating surfaces. . Paint* outside surfaces and "O" ring seats. . Oil spray inside surfaces after painting outside.

*Use a good, oil resistant, corrosion resistant paint.

Visual inspection of the thrust bearings revealed a general state of corrosion, evidenced - to varying degrees - as pitting and/or discoloration on the balls and outer races. (The inner races were not inspectable without bearing disassembly.) All bearings from both starboard gearboxes showed very minor effects of corrosion compared to those from the port gearboxes. Table 8 summarizes the observations from the inspection,

As discussed in reports by D&L[36] and Boeing,[37] inspection revealed that all the roller bearings in the gearboxes had been subjected to different degrees of corrosion damage. It was recommended that all roller bearings should be replaced to assure 100 percent reliability for future operations. However, due to long lead times, high initial cost, and few available spares, alternative plans had to be substituted. (Reports by D&L[38,39] and Boeing[40,41] detail additional inspections, installation techniques, clearances and fit, replacements, exchanges, and final installation status of the bearings of the foilborne transmission gearboxes.)

SUMMARY OF MAJOR MODIFICATIONS

The major work accomplished to produce a zero time and reliable foilborne transmission system for PLAINVIEW included the following:

1. All 24 roller bearings in the gearboxes were replaced. Compared with the original 41 mm diameter rollers, the new 234 type bearings have 32 or 36 mm diameter rollers.

2. On the four horizontal gear shafts, the fretted material was machined away to a depth of 0.060 in. so that the final shaft diameter is now not less than 10.875 in.

3. The gear retainer bolts were machined to 0.820-0.0005 in. and the matching holes were reamed 0.820-0.0005 in. diameter; this was done to reduce fretting and retain realignment and concentricity. Bolt material is AISI 4140 or 4340 and hardened to 33-38 Rockwell C.

4. The ring gear register surfaces were hand worked to remove fretting material.

5. Gear couplings have a narrow range for misalignment capacity. In the PLAINVIEW installation, this misalignment capacity for both the original GE and Zurn type is approximately the same, ±0.23 degree. It is believed that the latter will hold up longer because its higher viscosity oil tends to maintain a better oil

TABLE 8 - BEVEL GEARBOX BEARING INSPECTION

Location	GE Dwg No.	Mfg No.	Mfg S/N	Outer	Rollers	Inner
				Port Lower Gearbox		
Horiz Aft	961B982	MRC MR234E1	3	120° arc of rust stains.	Good.	Good.
Horiz Fwd	961B982	MRC MR234E1	7	Single scattered rust lines nearly all around. Can get 90-120° clear of rust. Barely acceptable.	Most good, one roller with bad rust line.	Good.
Vert Aft Lo	962B982	MRC MR234E1	4	Good.	Good.	Many rust stain lines. Some with depth. Marginally acceptable.
Vert Aft Hi (Reject)	961B981	MRC R230E	05	90° arc of rust stains. One bad pit due to fatigue or particle.	Good.	Fair - OK.
Vert Fwd Lo	961B982	MRC MR234E1	2	Fair.	Fair.	Single rust stain line at many roller locations all around. No apparent depth - OK.
Vert Fwd Hi	9618981	MRC R230E	09	Good.	Good.	Single rust stain line at each roller location all around. No apparent depth - OK.
				Port Upper Gearbox		
Horiz Aft	961B982t	SKF 45691	207	Good. One rust line.	Several rollers with deep pit line.	Fair.
Horiz Fwd	961B982	SKF 45891	SGA8L	Fair. Some single rust lines. Barely acceptable.	Some rollers with single scattered rust pitting. Barely acceptable.	Fair.
Vert Aft Lo	961B982	SKF 45891	SGA4L	Good.	Good.	Many rust stain lines. Some with depth. Marginally acceptable.
Vest Aft Hi	961B981	MRC MR230E	3	Good.	Mostly good, one roller with single pit. Replace roller if possible.	Many rust stain lines. No apparent depth. OK.
Vert Fwd Lo (Reject)	961B982	SKF 45891	SGA3L	Fair.	Fair.	Many rust stain lines. One or two deep ones nearly across. No good.
Vert Fwd Hi	961B981	MRC MR230E	012	Good.	Most good, several with medium deep rust pits. Marginally acceptable.	Minor scattered rust stain lines. some small rust pits at outboard end. OK.

Note: All bearings were replaced during RAV.

TABLE 8 (Continued)

Location	GE Dwg NO.	Mfg' NO.	Mfg S/N	outer	Rollers	Inner
				Stbd Lower Gearbox		
Horiz Aft	961B982	SKF 45696	E6	Minor scattered rust stains. Acceptable.	Good.	3 shallow score lines all around (2 IB, 1 OB). Acceptable.
Horiz Fwd	961B982	MRC MR234E1	5	Good.	Good.	Several shallow score lines at IB end. Acceptable.
Vert Aft Lo (Reject)	961B982	MRC MR234E1	8	Nearly 180° of bad rust.	Four rollers badly rust-ed.	One shallow score around IB end. Several deep rust lines across. 30° of rust patches. Reject.
Vert Aft Hi (Reject)	961B981	MRC R230E	08	Nearly 180° arc of rust 1/2 across or better.	Medium rust stains, some with depth.	Minor rust pits nearly all around. OK.
Vert Fwd Lo (Reject)	961B981	MRC MR234E1	6	Good.	Good.	One deep score line all around at IB end. Other-wise looks good. Reject.
Vert Fwd Hi	961B981	MRC R230E	06	30° arc of rust stain nearly all across.	Good.	One shallow score line all around at IB end. One patch of rust stain 1/2 in. x 1 in. in area. Marginally OK.
				Stbd Upper Gearbox		
Horiz Aft (Reject)	961B982	SKF 456961A	SGA6L	180° arc of rust stains. 75 percent across.	Several rollers with deep rust pits - reject.	Good.
Horiz Fwd	961B982	SKF 456961,~	SGA5L	Good.	Good.	Good.
Vert Aft Lo	961B982	SKF 456961A	SGA2L	Single, scattered rust lines nearly all around all across. Question-able.	One or two lines of rust nearly all across. Questionable.	Many single rust stain lines - no real depth. OK.
Vert Aft Hi (Reject)	961B981	MRC R230E	04	90° arc of rust stains nearly across.	Medium rust stains. Reject.	Minor rust pitting and lining all around. OK.
Vert Fwd Lo	961B982	SKF 456961A	SGA1L	Double rust lines scat-tered all around. All across. Questionable.	Most look good. Few with single rust line.	Single, wide rust line at each roller location. Depth questionable. Pos-sible reject.
Vert Fwd Hi (Reject)	961B981	MRC R230E	010	90° arc of rust stain 1/3 across.	Medium rust stains, some with depth.	Medium rust pits and lines - some with depth. Reject.

film between the gear teeth under excessive misalignment conditions. Zurn couplings were installed.

6. Each assembled gearbox was subjected to a spin test at 25 percent torque loadings.

GEAR ASSEMBLY AND TESTS

During reassembly of the four bevel gearboxes, a spin test of each overhauled gearbox was made with the assembly loaded to 25 percent torque. Satisfactory tooth contact patterns were found on three units in the assembled condition. From a report by D&L,[42] Figures 20 through 24 show tooth patterns finally produced. In addition, backlash measurements normal to the tooth surface at the heel were made on each gearbox; Table 9 summarizes these measurements and the factory stamped backlash before PLAINVIEW operation. Since these tests were run at 25 percent loading, 100 percent torque will obviously change the pattern of teeth contact. Experience has indicated that increasing torque to 100 percent tends to broaden the load flank pattern toward both toe and hell, tending more toward the heel than toe. Therefore, a centered impression or one contacting slightly toward the toe is considered ideal at 25 percent torque.

On assembly of gearbox components, each gearbox was subjected to the following test:

1. The test drive motor was started and the gearbox brought up to 100 percent speed in 25 percent speed increments. Each speed was held for 10 min in order to monitor temperatures, noise, and vibrations. The 100 percent speed was held for 30 min (100 percent speed = 1,570 rpm shaft speed).

2. At the conclusion of the 100 percent speed run, speed was increased to 110 percent and held for 5 min.

3. The speed was slowly reduced to shut down.

4. At the conclusion of the test, the gearbox was rotated by hand to insure there were no tight bearings.

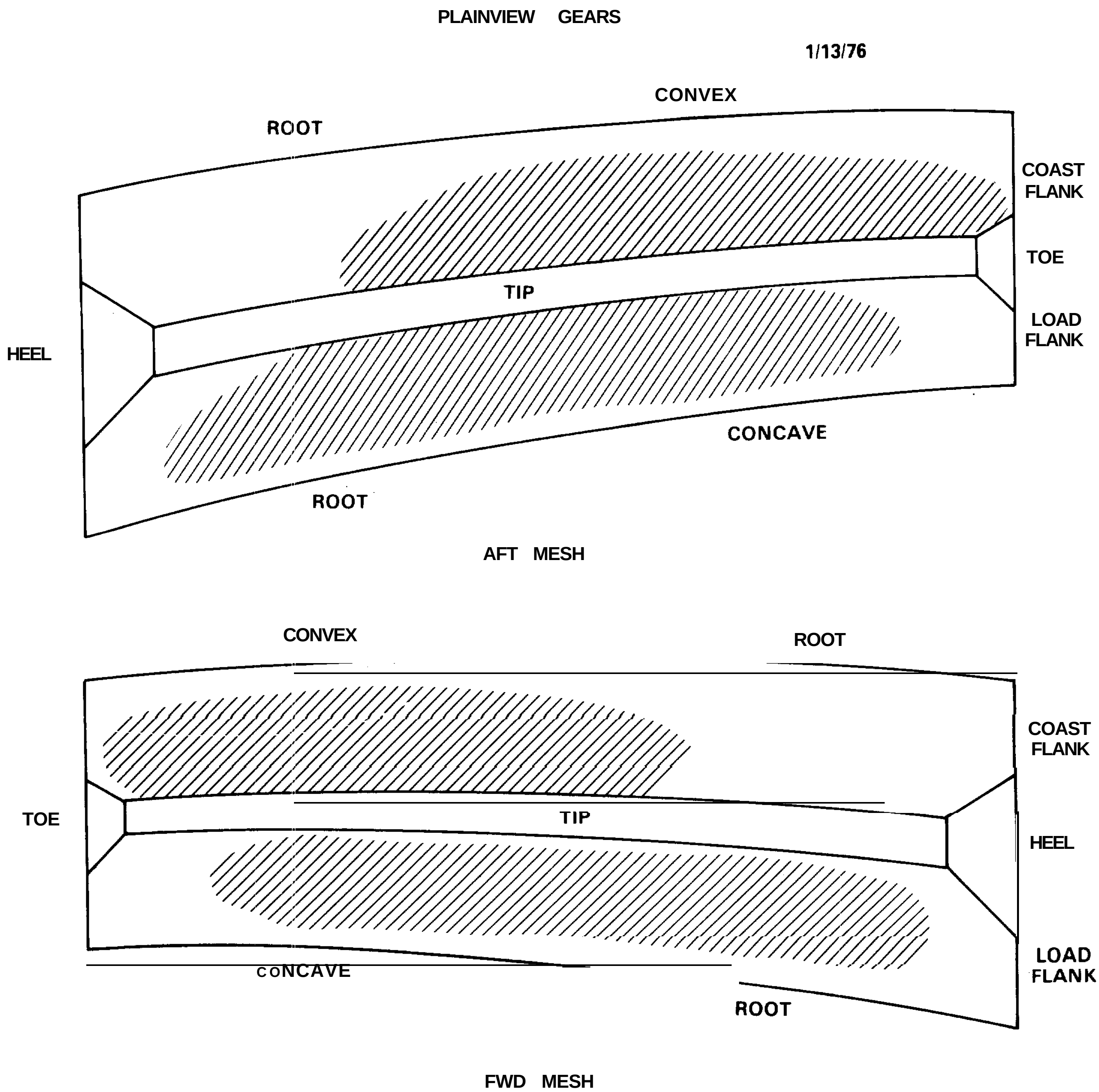
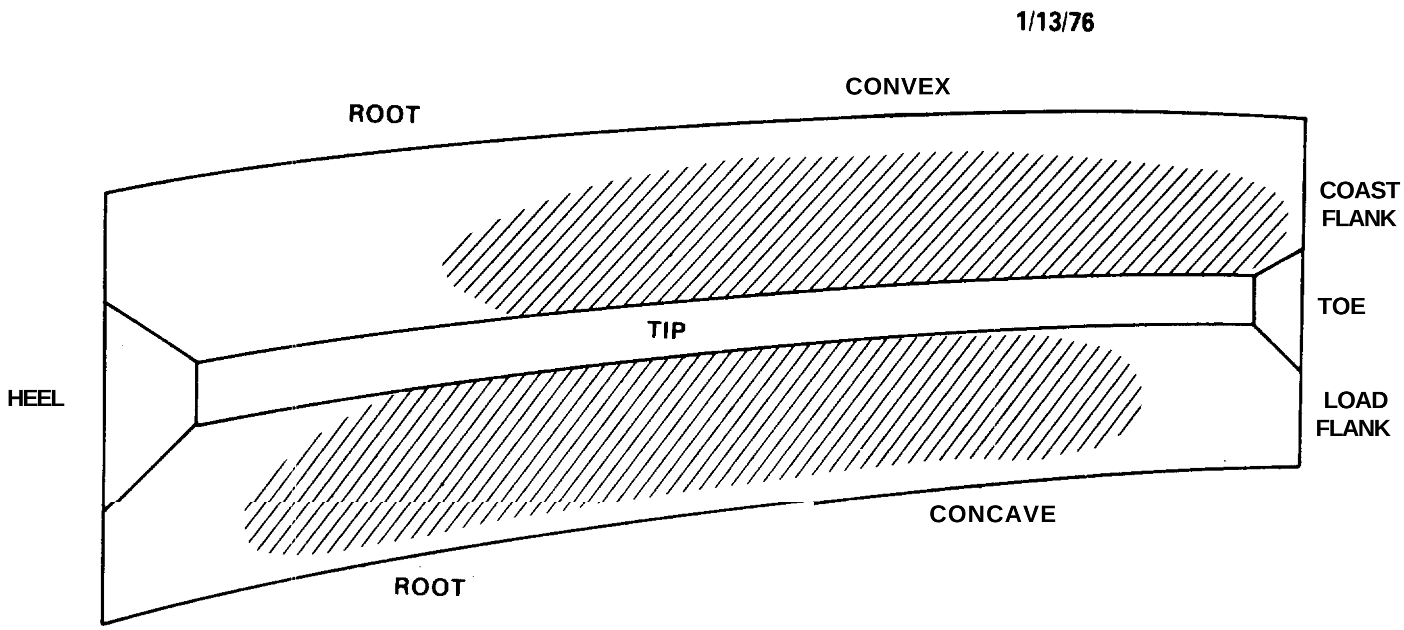
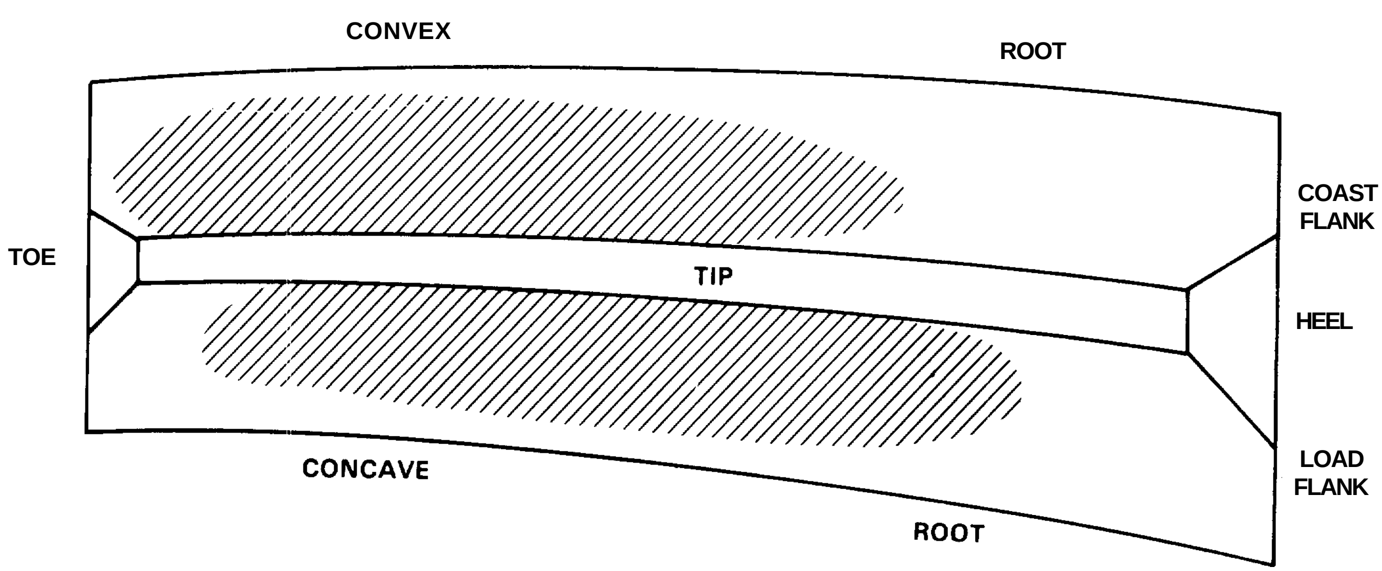


Figure 20 - Tooth Contact Pattern: Port Upper Gear, at 192 Hours of Operation

PLAINVIEW GEARS



AFT MESH



FWD MESH

Figure 21 - Tooth Contact Pattern: Starboard Lower Gear, at 192 Hours of Operation

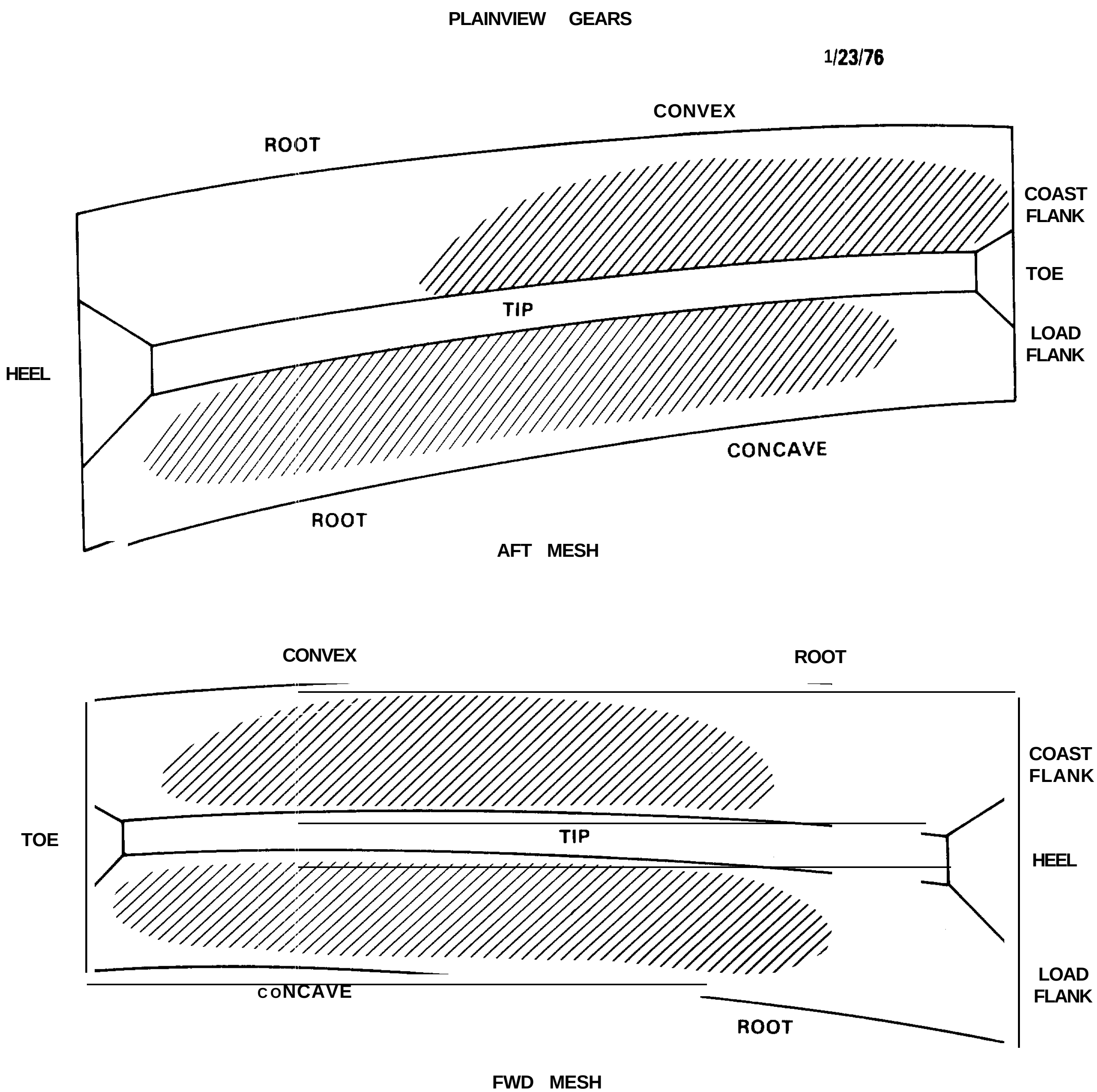


Figure 22 - Tooth Contact Pattern: Port Lower Gear, at 192 Hours of Operation

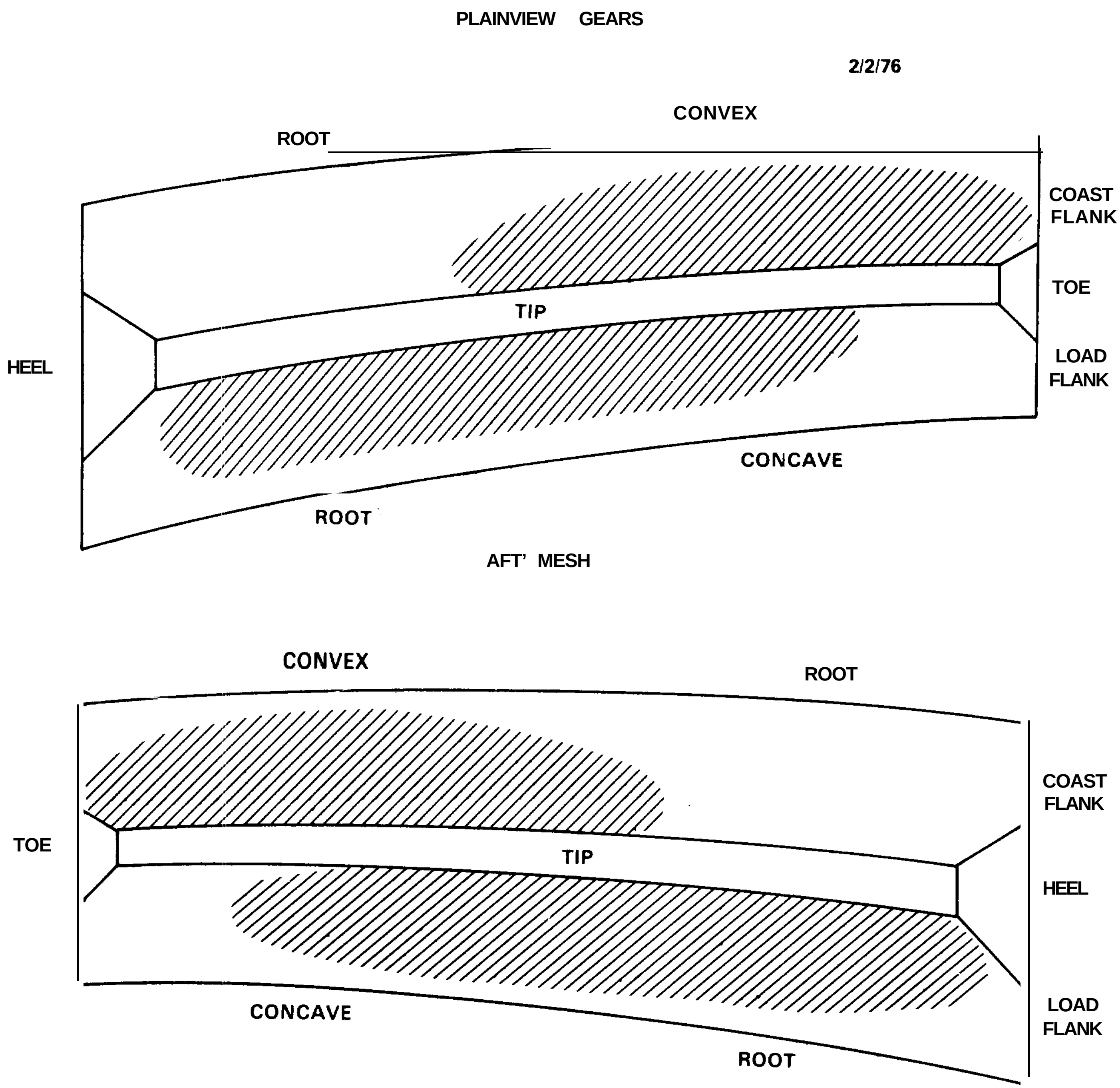
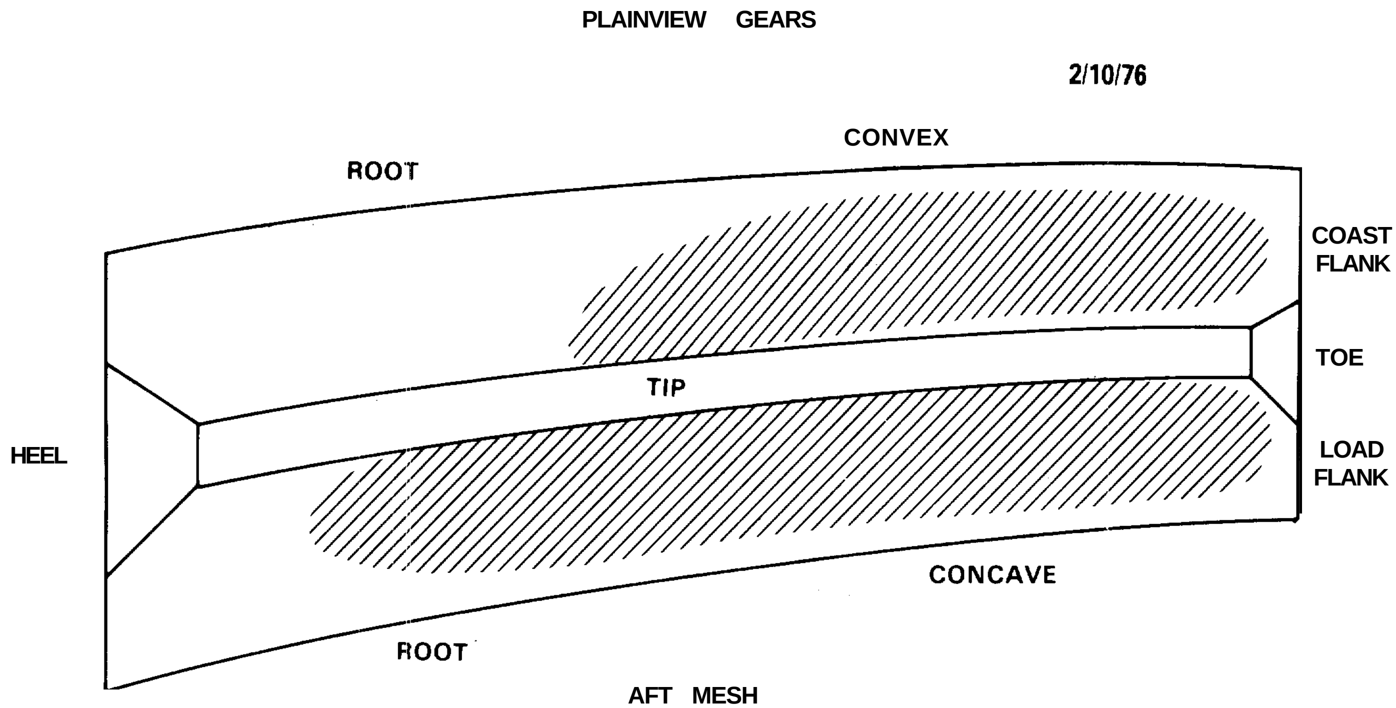


Figure 23 - Tooth Contact Pattern: Starboard Upper Gear After Increasing Bearing Clearance, at 192 Hours of Operation



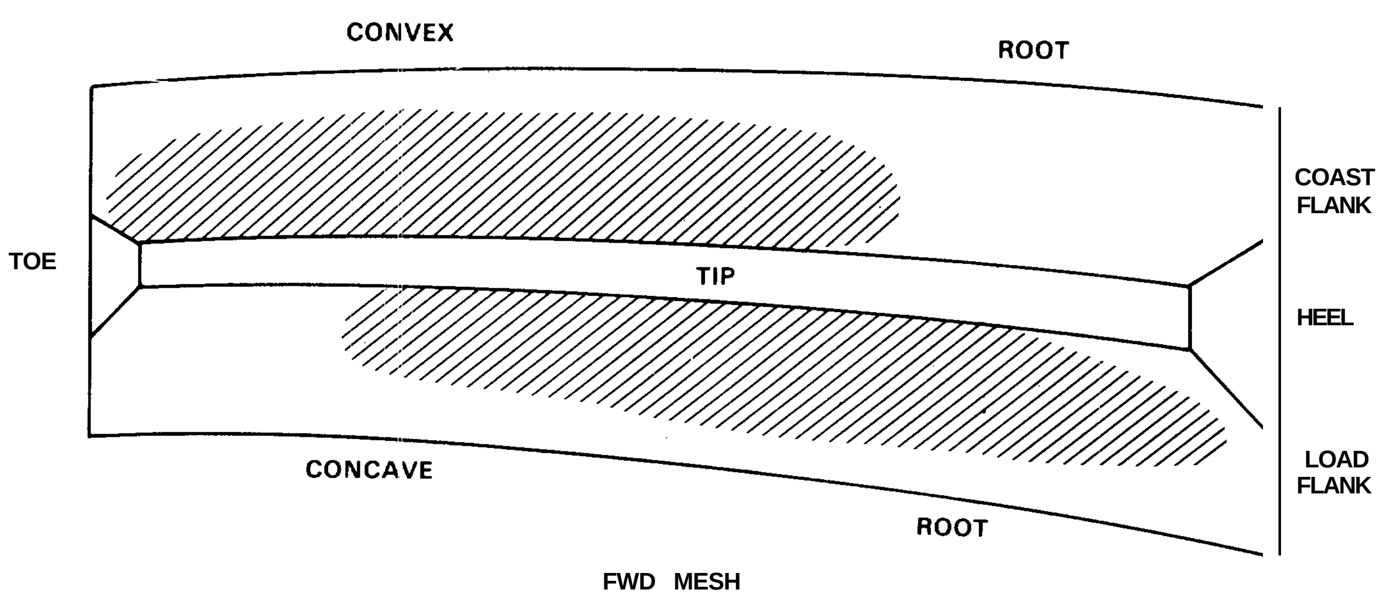


Figure 24 - Tooth Contact Pattern: Starboard Upper After Moving Aft Vertical Gear 0.010 inch into Mesh, at 192 Hours of Operation

TABLE 9 - BACKLASH MEASUREMENTS ON GEARBOXES

Gearbox	Reassembly Date	Backlash Stamped on Gear (in.)		Measured Backlash on Gear (in.)	
		Forward	Aft	Forward	Aft
Port Upper	1/13/76	0.024	0.025	0.027	0.029
Starboard Lower	1/19/76	Not available	0.046	0.041	0.044
Port Lower	1/23/76	0.025	0.038	0.036	0.044
Starboard Upper	2/10/76	0.035	0.038	0.041	0.035

POSTOVERHAUL EXPERIENCES

FOILBORNE OPERATIONS

After the long overhaul period with major modifications to various systems, PLAINVIEW resumed operations and flew for another 74 hr before being taken permanently out of service. During this period an attempt was made to operate the craft at its highest power level, that is, by getting about 14,500 hp from each engine, while holding the craft in the hullborne mode. This operation failed because of bearing trouble in the port-side high speed pedestal bearing assembly. It was shortly after this that all operations with PLAINVIEW were terminated. A total of 268.5 foilborne hours had been accumulated in the 15 yr life of PLAINVIEW, the largest hydrofoil craft in the world.

FINAL GEARBOX INSPECTION

A final gearbox inspection was made on 1 November 1978. Although the upper gearboxes had been removed from the craft, the two lower gearboxes were still mounted in the struts. The upper gears could be rotated through 360 degrees for examination of all the teeth. Since the lower gears were still connected to the long drive shafts in the struts, complete rotation of the gears was impossible. The teeth and bearings of the gears were inspected by removing the inspection covers.[43]

Generally, the surfaces of all visible teeth in all the gears were in excellent condition except for spots of surface rust at the toe of some of the teeth. Contact patterns were observed to be satisfactory. The running tooth contact pattern appeared to be **80-90** percent of the total tooth surface on the loaded side of the teeth. The coast or unloaded side showed a nonuniform pattern. On some of the teeth, the contact pattern occupied about 30 percent of the total tooth surface at the middle of the teeth. Details of the final inspection are given in an NSRDC memo report.[43] Figures **25** through 32 show the tooth contact pattern and give explanatory notes about the general condition of each gear and pinion.

The rollers and races, where visible (bearings were not disassembled), were generally in good condition; however, the ends of some rollers and races of the port upper gear box evidenced some spots of rust. It should be noted that during the previous overhaul in 1976 new bearings had been installed.

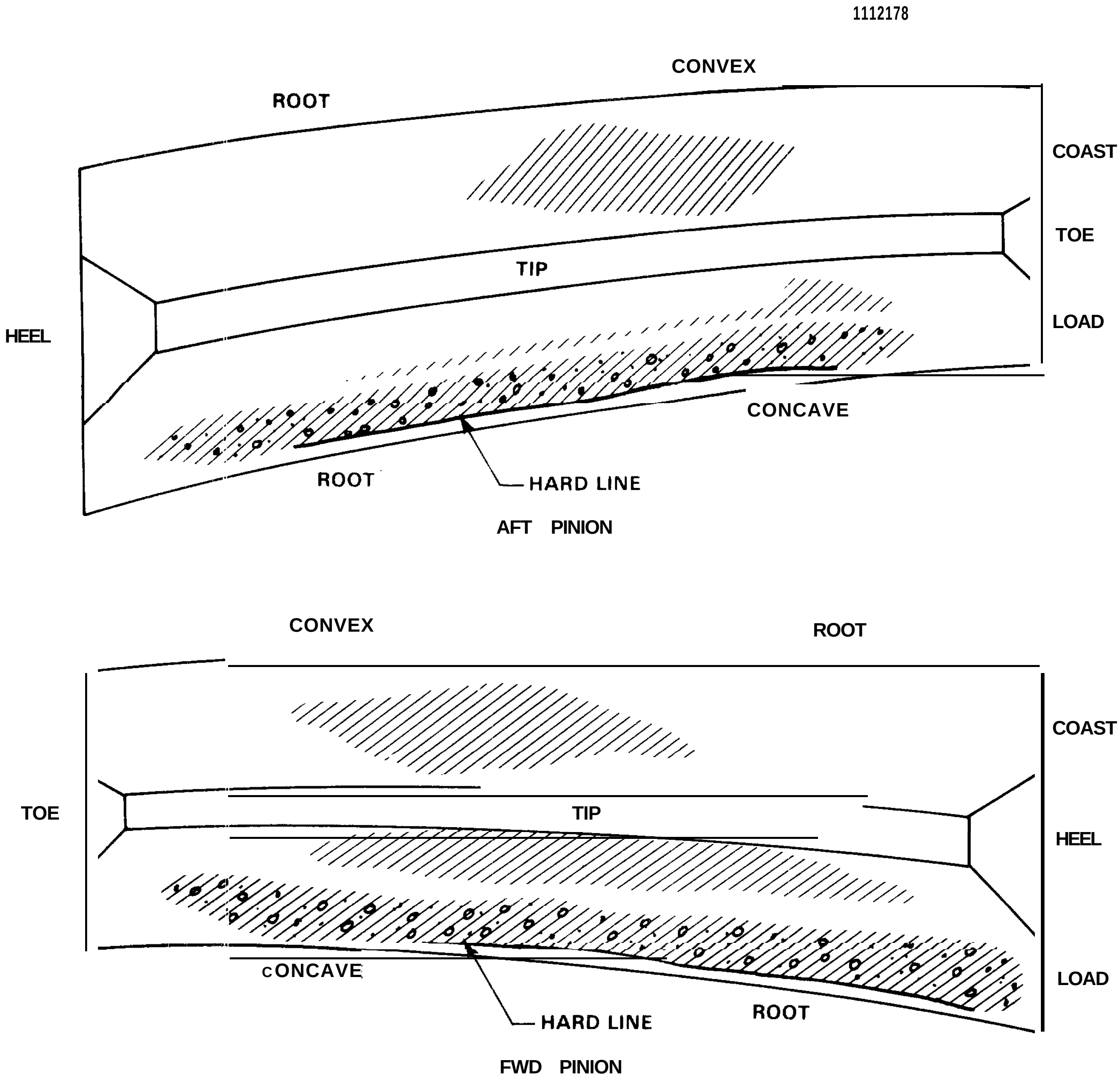
In the gear coupling of the starboard and port upper gear boxes, most of the coupling teeth were covered by a dirty deposit of salt crystals (salt by taste) from apparently contaminated lube oil. Heavy spalling and fretting were observed over 90 percent of the width of the 2 1/2 in. wide tooth. This is very typical of a tooth spline type coupling. Figure 3 shows the starboard lower gear at inspection.

VIBRATION CHARACTERISTICS IN THE POWER SYSTEM

Structureborne vibration data taken in 1970 on the port and starboard SI gear boxes, while the craft was in the foilborne mode. showed about 0.4 in./sec maximum vibration in a frequency range from 200 to 1300 Hz.

Data taken in 1972 on the bevel gear boxes, again under foilborne operations, indicated a maximum vibration of 0.2 in./sec at 68 Hz with other vibration peaks at much lower levels.

In 1976, postoverhaul testing of the port propulsion gas turbine produced excessive vibration levels, 0.010 in. double amplitude and high temperatures at the turbine drive shaft pedestal bearing. Normal vibration values for the LM 1500 gas turbine are 0.001-0.005 in. double amplitude. These symptoms appeared to indicate turbine unbalance and ball bearing skid in the pedestal bearing, the latter due to lack of proper lubrication. In comparison, industrial standards for rotating machinery signals from the port unit at rotational frequency were in the "rough" to "very rough" region. Levels up to 0.002 in. double amplitude were observed at the questionable bearing. Comparable measurements on the starboard unit were less



NOTES:

- Wear patterns on load side are just appearing thru the oxide coating.
- Some scattered pitting seen. Not considered a problem.
- Backlash feels all right – by hand turning.

Figure 25 - Tooth Contact Pattern: Port Upper Input Pinions, at 268 Hours of Operation

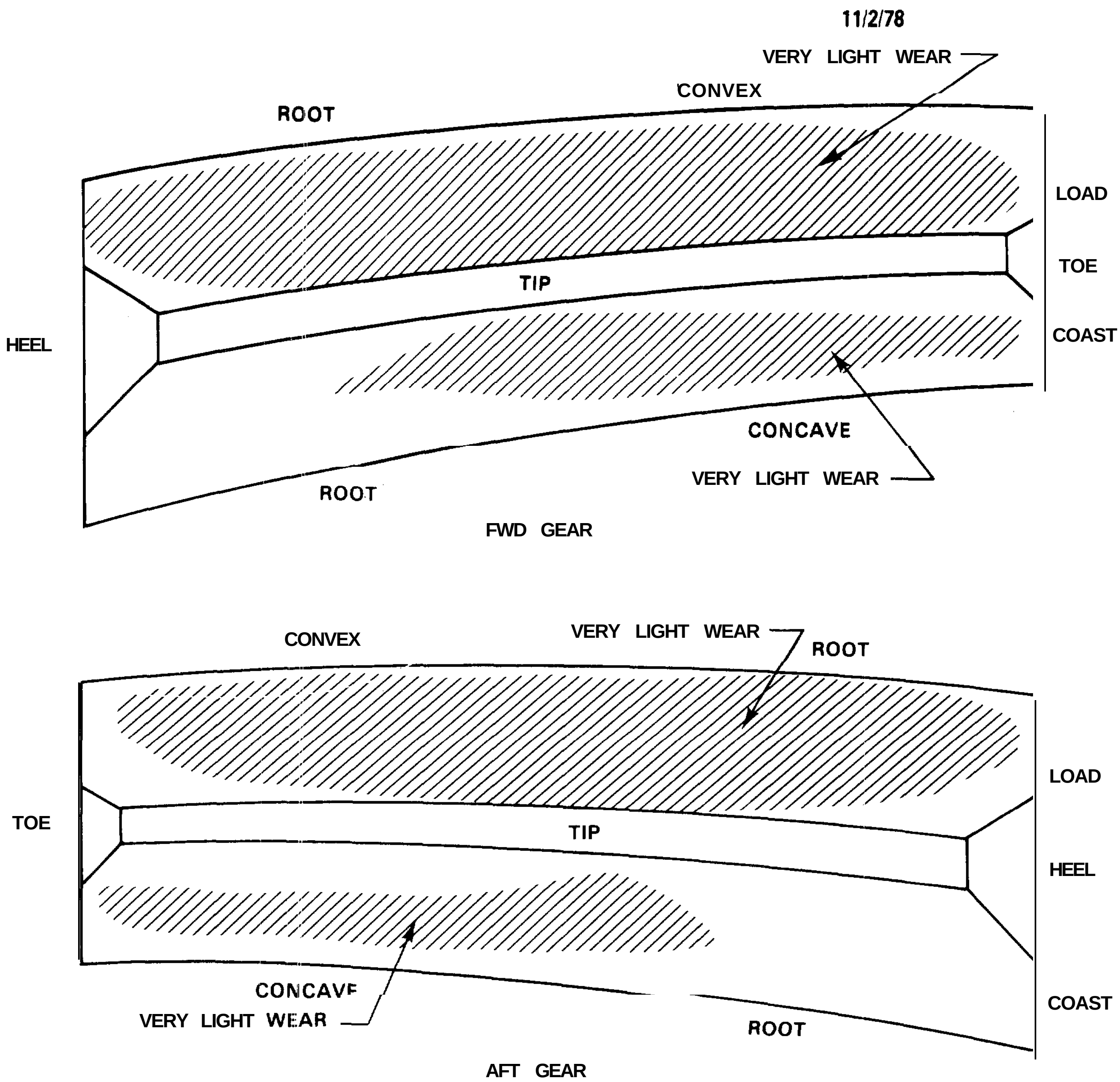
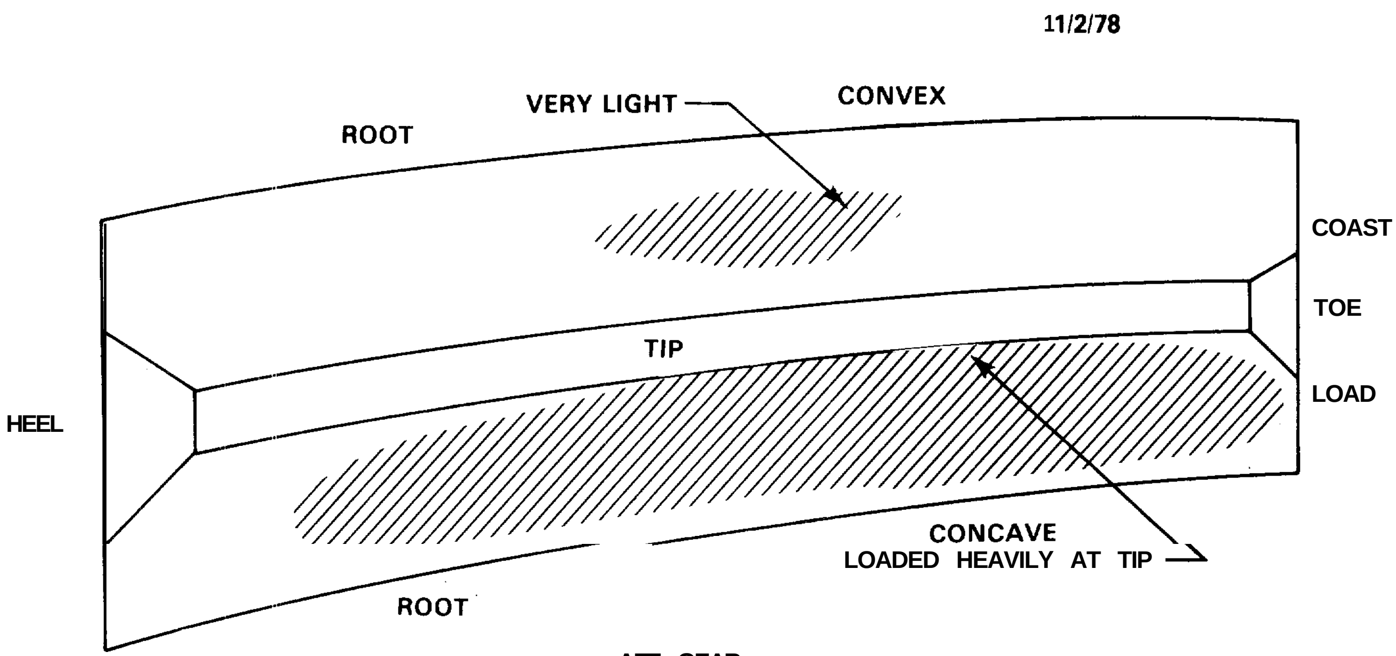
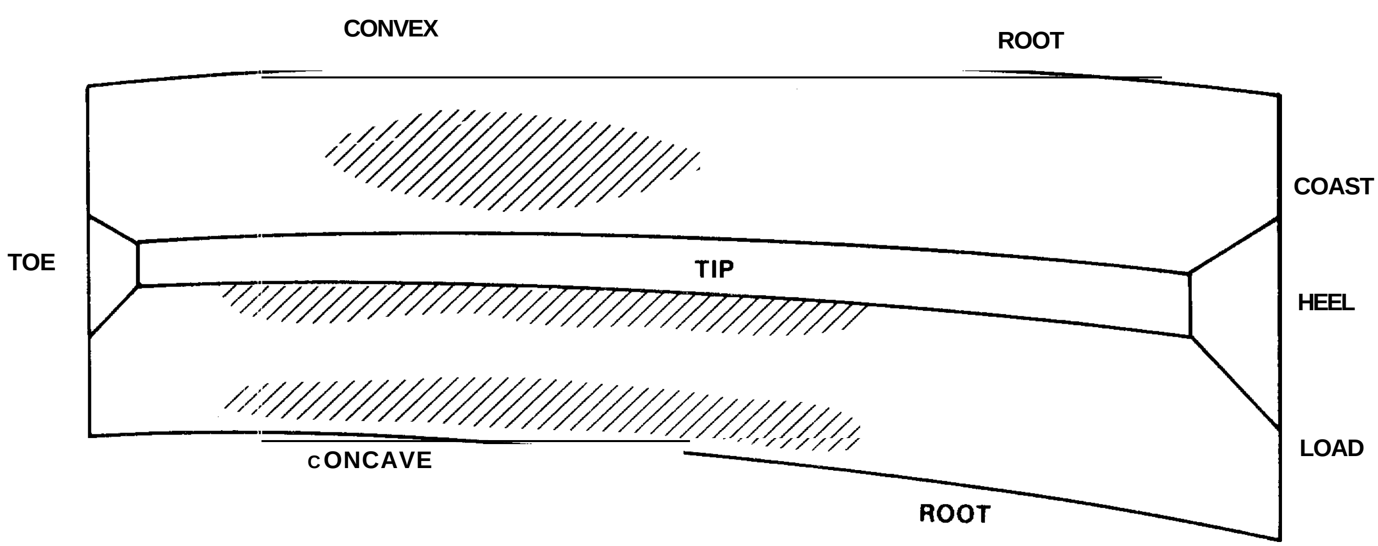


Figure 26 - Tooth Contact Pattern: Port Upper Gears, at 268 Hours of Operation



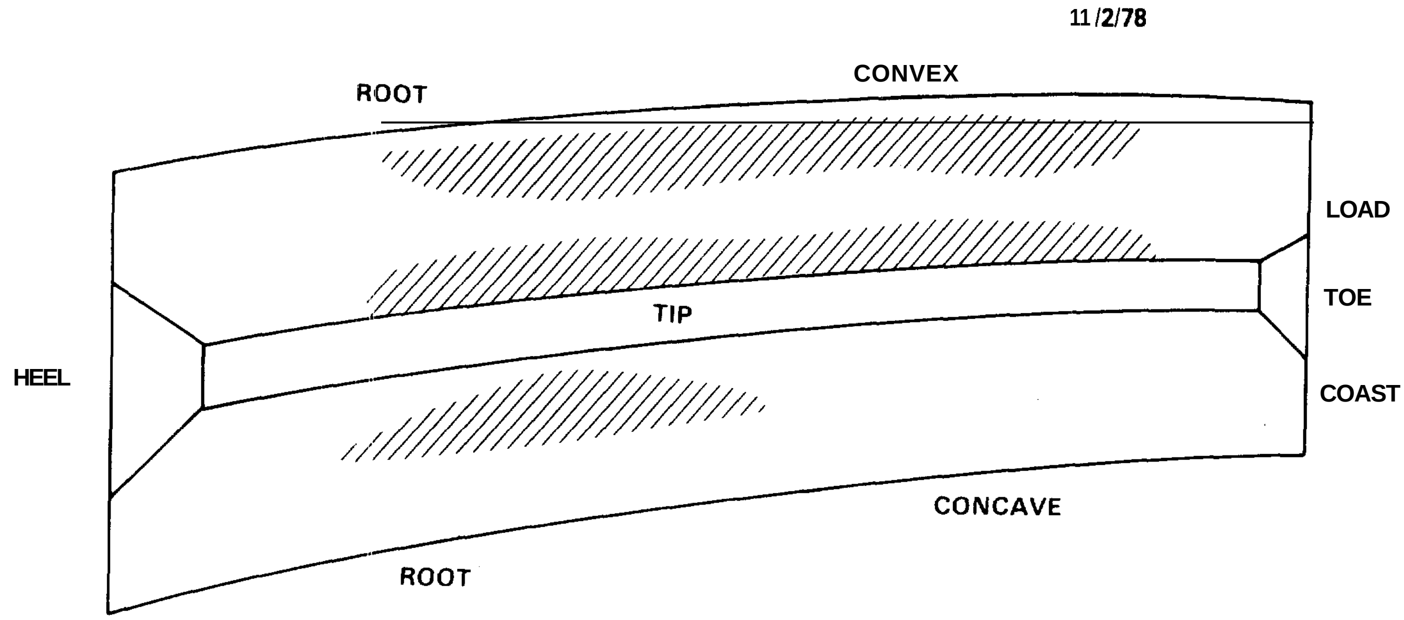
AFT GEAR



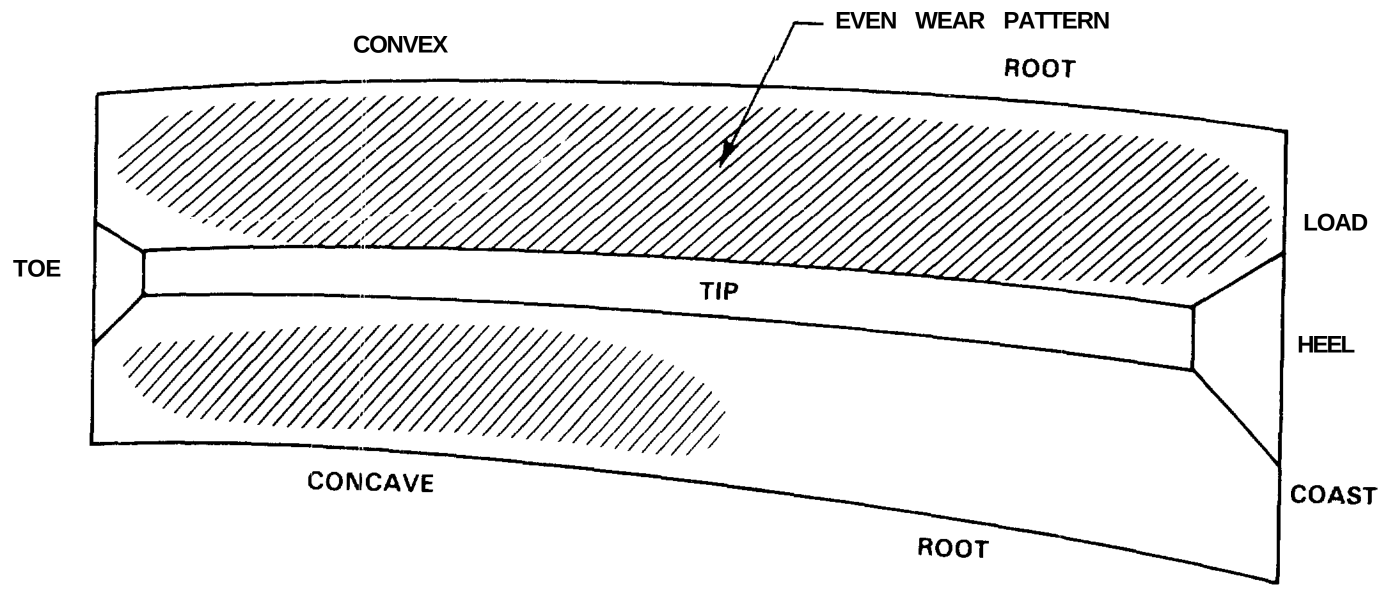
FWD GEAR

NOTES:
- Could not move gear for backlash.
- Wear pattern just appearing.
- Gears in good condition.

Figure 27 - Tooth Contact Pattern: Port Lower Gears, at 268 Hours of Operation



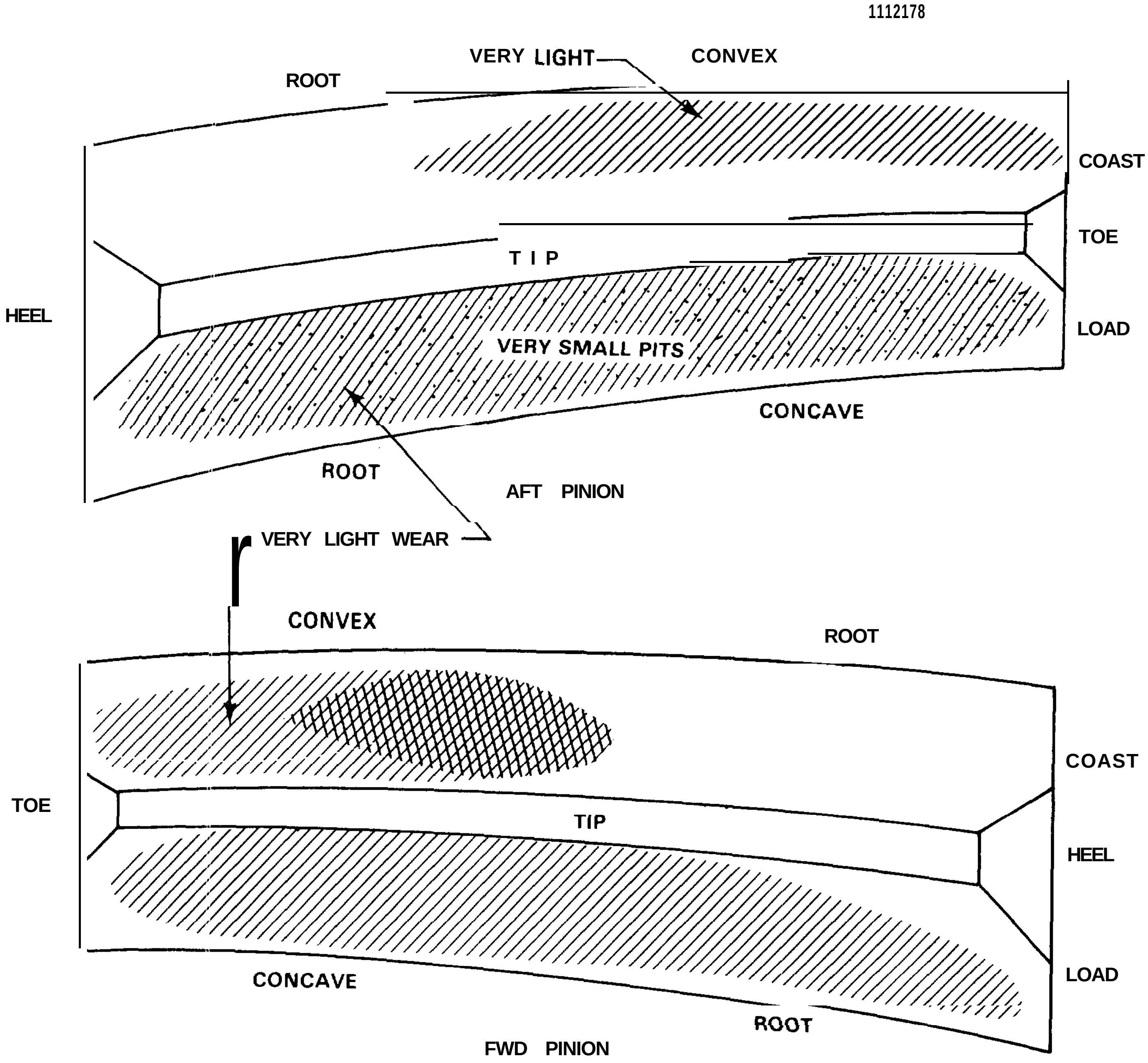
FWD PINION



AFT PINION

NOTES:
- **The two old scoring patterns are tarnished.**
- **Wear patterns are just appearing *thru* the oxide coating.**
- **Pinion in good condition.**

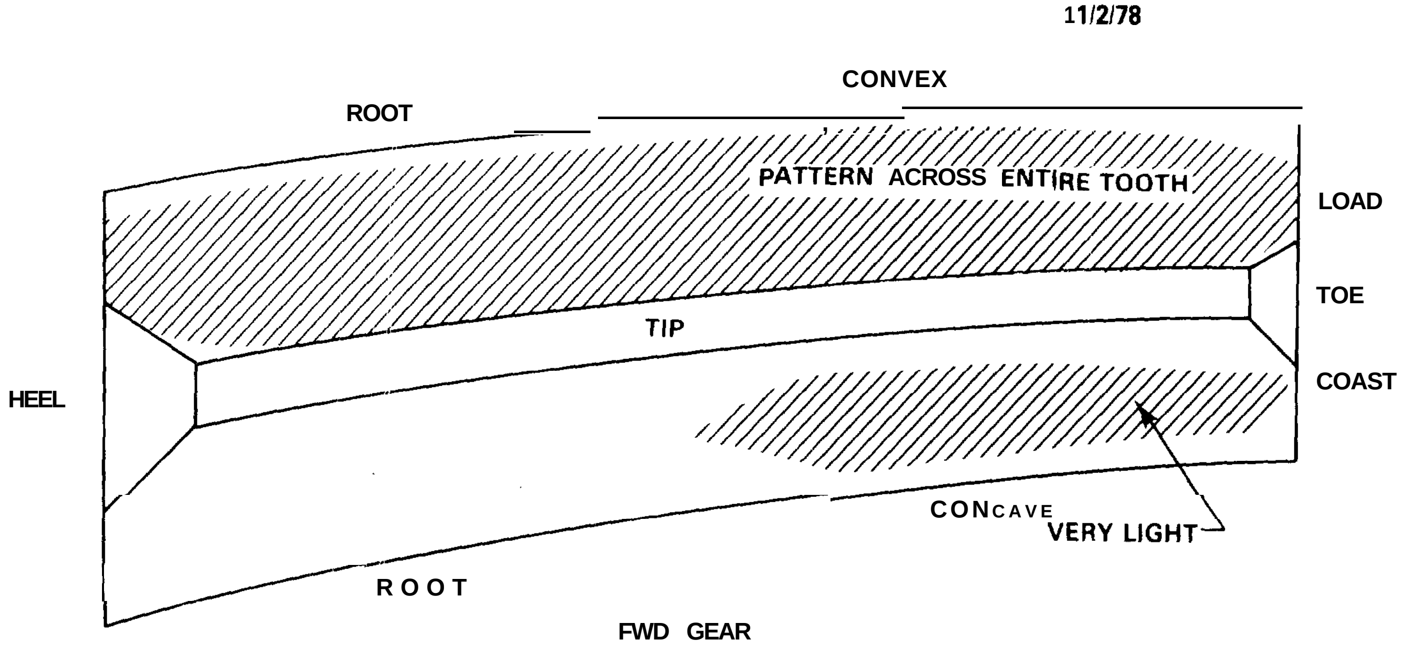
Figure 28 - Tooth Contact Pattern: Port Lower Pinion, at 268 Hours of Operation

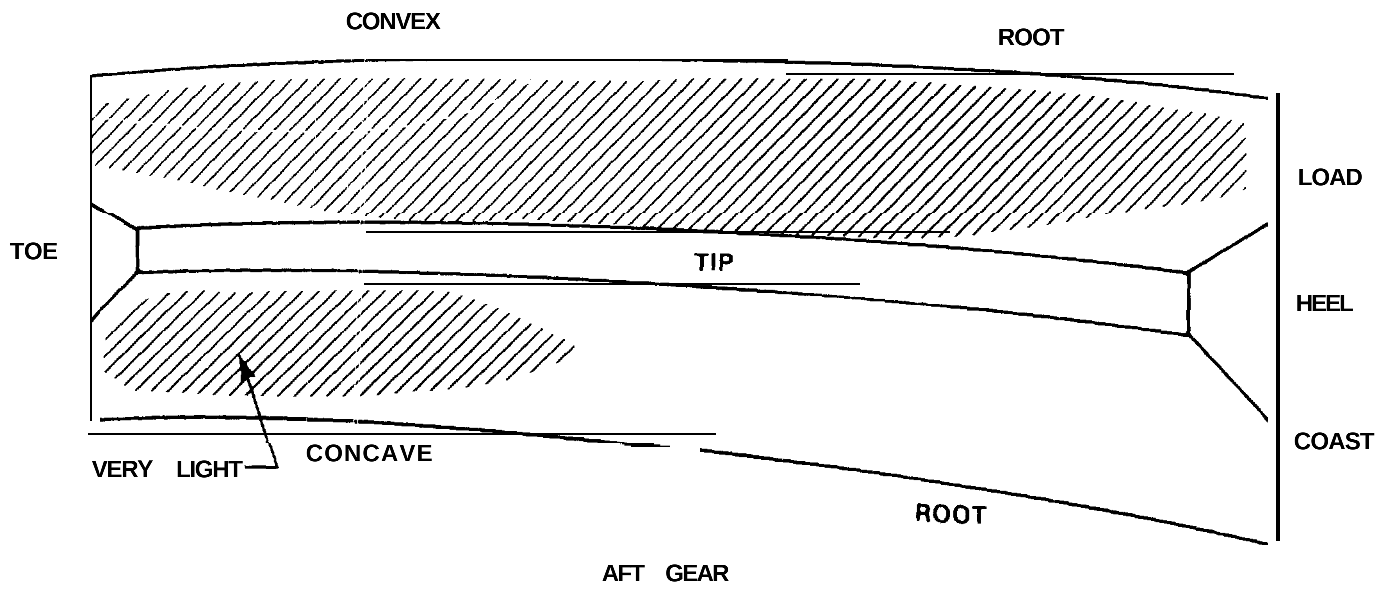


NOTES:

- Wear patterns on load side are just coming thru the oxide coating.
- Backlash feels all right. Seems to be more than port unit.

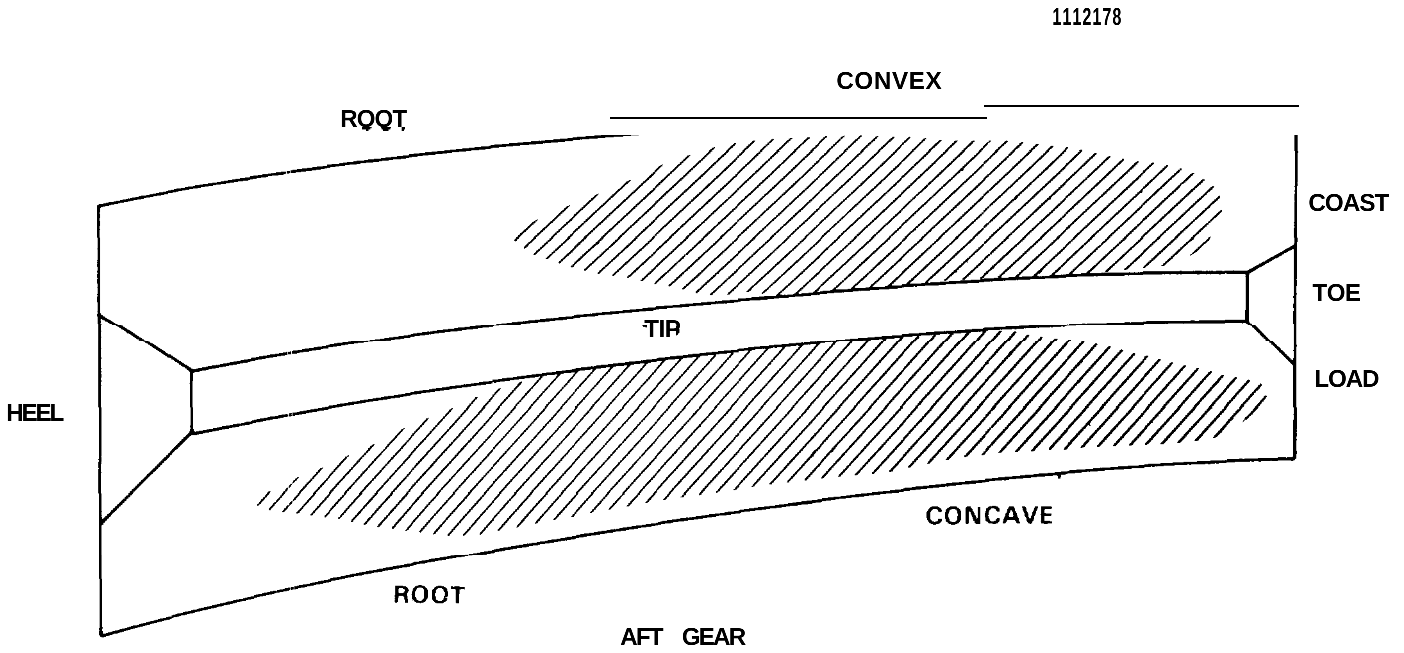
Figure 29 - Tooth Contact Pattern: Starboard Upper Input Pinion, at 268 Hours of Operation

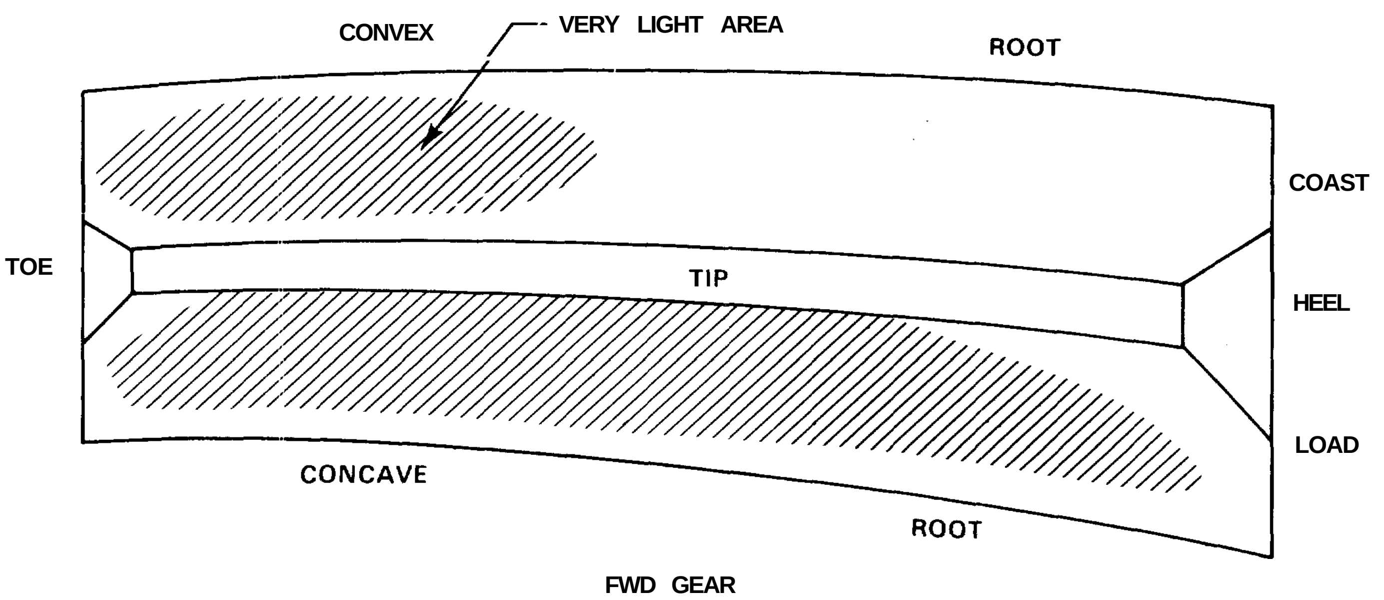




NOTES:
- **Very light wear seen thru the black oxide coating.**
- **Gears look very good.**

Figure 30 - Tooth Contact Pattern: Starboard Upper Gears, at 268 Hours of Operation

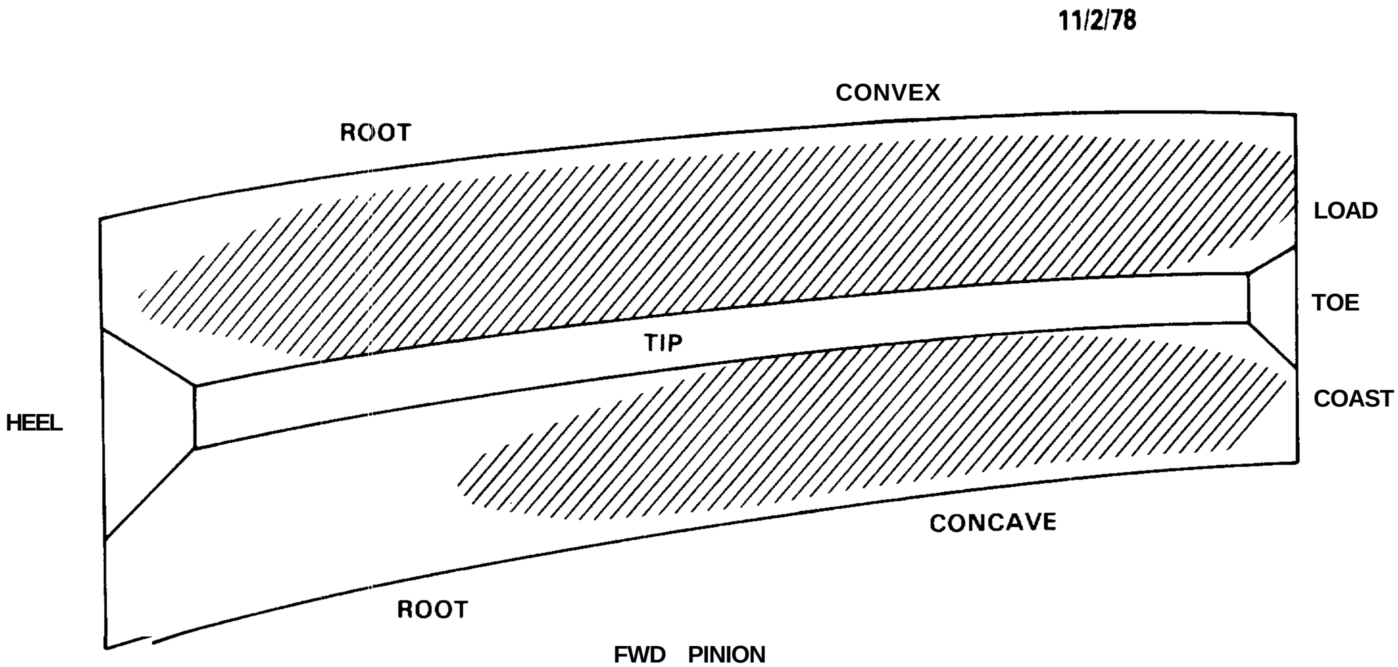


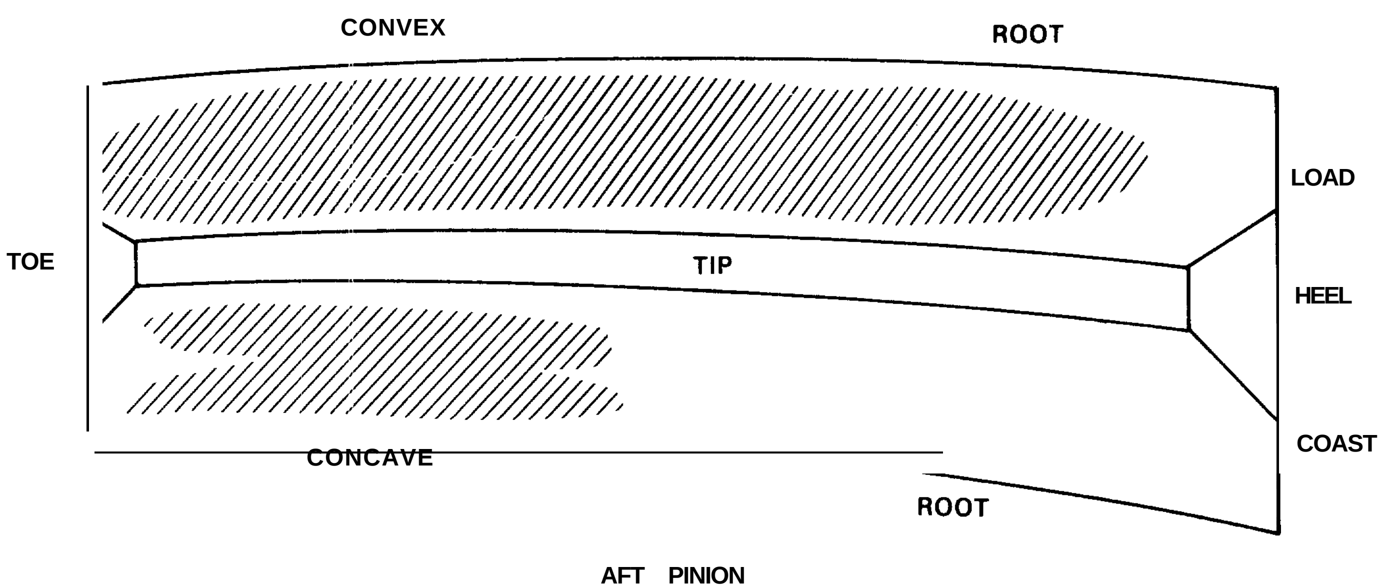


NOTES:

- **Wear patterns on load side are just appearing.**
- **Backlash feels all right.**
- **Gears look in good condition.**

Figure 31 - Tooth Contact Pattern: Starboard Lower Gears, at 268 Hours of Operation





NOTES:
- **Wear patterns just starting to appear thru oxide coating.**
- **Pinion in good conclition.**

Figure 32 - Tooth Contact Pattern: Starboard Lower Pinion, at 268 Hours of Operation

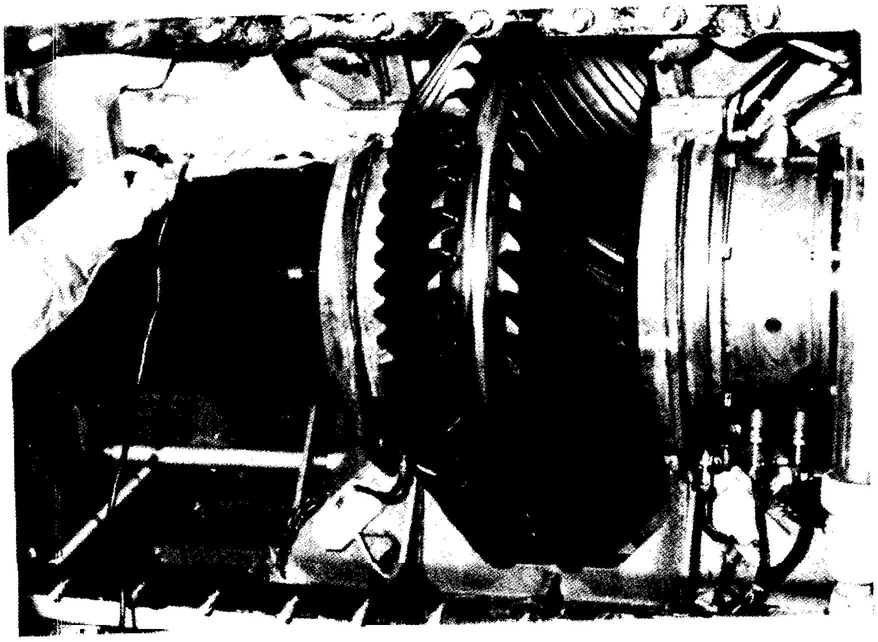


Figure 33 - Starboard Lower Gear Box

than 0.001 in. double amplitude. Subsequent adequate lubrication reduced the bearing vibration level to acceptable limits and recommendations to improve turbine balance were made..

In 1977, a vibration survey was taken to establish a vibration monitoring system for the foilborne transmission system. Again the bevel gears indicated low vibration levels, less than 0.4 in./sec.

Figure 34 illustrates vibration levels of different components and systems of advanced craft, as described by Harbage.[44] Most of the vibration data is in the "slightly rough" and "rough" regions of this rotating machinery vibration chart; some vehicles show even higher vibration levels. It should not be assumed that because of the high levels indicated, these gears or components are headed for imminent failure; rather, they exhibit different characteristics from the conventional relatively "heavy and slow speed" systems for which this guide was established.

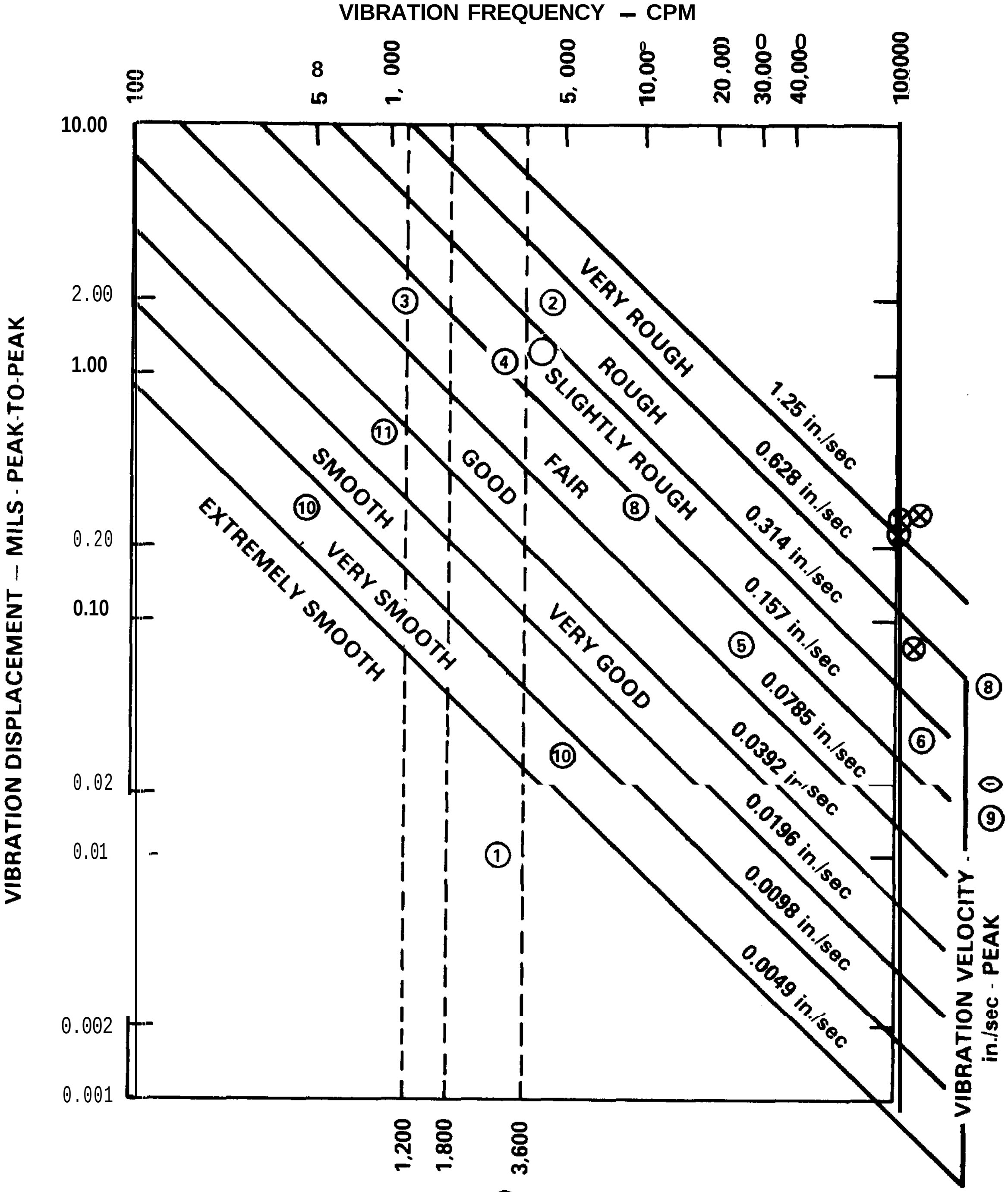
The gears listed, for instance, are usually small and lightweight (0.2 to 0.5 lb/hp) for the large amount of power they transmit and, thus, have little or no internal damping qualities. They are run at high speeds, so balance and alignment of the power systems are critical. Alignment is difficult to maintain because of the flexibility of the aluminum hull structure and foundations. These gears are designed to be more closely related to helicopter gears than marine type gears, and it is known that the former routinely run at vibration levels exceeding 5 in./sec. Only a greater number and longer operation in the marine environment will help establish a more realistic criteria for vibration levels.

Recent data obtained from bevel gear systems in both Navy 160-ton air cushion vehicles show operating vibration levels range from 0.5 to 2.0 in./sec. These gears operate in a power range of about 2600 to 7600 hp and have a weight which ranges from 0.08 to 0.17 lb/hp.

SUMMARY

This report traces the operational life of the PLAINVIEW foilborne transmission system from its development in 1962 to the retirement of PLAINVIEW in 1978.

The transmission system bevel gears were designed to have a life of approximately 30,000 hours when operated for 20 percent of the time at take-off power and 50 percent of the time at maximum continuous power. They are considered to be the largest gears ever manufactured for the power output -- 26 in. diameter and



①	AGE H HULL MOTIONS
②	AGE H PORT PIER BEARING
③④	PCH HULL MOTIONS
⑤	TUCUMCARI HULL MOTIONS
⑥	TUCUMCARI TURBINE GEARBOX
⑦	SES-IOOA BEVEL GEAR
⑧	SES-100B TRIM RATE ON BEVEL GEARS AND BLADE RATE ON PLANETARY
⑨	SES-100A PLANETARY AHEAD OF PUMP
⑩	GOOD NAVAL MACHINERY IN LAB
⑪	IN VEHICLE
O	PLAINVIEW BEVEL GEARS
⊗	AALC GEARS

Figure 34 – General Machinery Vibration Severity Chart

17,500 hp/mesh. The history of the gear systems shows that most of its operational life was at a reduced percentage of rated design power. For example, in over 350 total hours of operation (through February 1978) more than 268 hours were at approximately 60 percent of rated power. About 20 hours were at 90 percent of rated power. This sum was accumulated from totaling all the take-off power conditions; records show that more than 500 take-offs were made.

Three prinicpal problems were predominant in the power transmission system: (1) gear fretting corrosion, (2) bearing corrosion and damaged bearings, and (3) spline couplings failures. The first was due primarily to vibratory forces separating the bolted ring gears from their shafting; the second was caused by water penetrating into the lube oil system; the third was the result of misalignment of the driven system. All these problems can be resolved by more attentive design, manufacturing, and assembly details.

It is believed that in their present condition, the gear sets are capable of many hundreds more hours of operation.

FUTURE DESIGN CONSIDERATIONS

1. A gear drive system cannot tolerate water in its lubrication system. Throughout the life of PLAINVIEW, the transmission system was exposed many times to considerable seawater contamination. Although the gears and companion bearings were enclosed in a sealed gear box, water entered the box through many improperly sealed pipes, cable penetrations, and instrumentation connections. While oil pressure seals around the propeller shaft are successful in keeping water out, complete sealing of the gearbox may not be possible and other precautions must be incorporated in redesign. Serious consideration should be given to using lubricants which will tolerate some water and yet still form a protective film on metal surfaces. An oil-water separator has to be an important part of the lube oil system. Careful monitoring of lube oil for contamination must be part of the operational procedures. Each gear box should have its own independent lubrication system to avoid contamination spreading by way of a common sump or supply system.

2. To prevent fretting in the gears themselves, the gear and its shafting should be an integral unit. If this is not possible, because of the consequent size and weight increases, extreme care must be taken to ensure a tight fit between the gear and its shafting through body-bound bolts and dowels. (Appendix B gives calculated effects of fretting corrosion on reduced capacity of bevel gears.)

3. Although proposed by some experts, it is impractical to consider the use of integral inner roller races in the large size bearings because of the very high cost of replacing shaft and bearing if any failures occurred.

4. Gear boxes should be designed to eliminate inside structural members where water or dirt can be trapped. For structural strength, ribs should be added exterior to the box, if possible. "O" ring sealing should be used on removable inspection plates.

5. Some experts say that the present lightweight transmission system is not structurally sturdy enough to operate satisfactorily over the long term at high power requirements. This claim cannot be resolved at the present time. More test and operating data at full design conditions are needed, and it is recommended that the present system undergo a vigorous shore based test to resolve this issue.

APPENDIX A[10,11]

QUESTIONS PUT TO GENERAL ELECTRIC (GE) AND LOCKHEED SHIPBUILDING AND CONSTRUCTION COMPANY (LSCC) ON AGEH 1 GEARSETS
12 SEPTEMBER 1968

1. (Q) Who designed the gears? Gleason or GE?

   (A) Basic design by GE in cooperation with Gleason (tooth data and cutting data by Gleason). After grinding by Gleason, GE set up and torque tested gears, and if contact pattern was doubtful, sent gears back to Gleason for correction to achieve good contact under torque test in actual gearbox. (Note: This may account for stampings of 0.038 in., 0.033 in., and 0.046 in. on lower gears.)

2. (Q) What is "circular thickness"? Where and how measured?

   (A) Answer not known.

3. (Q) To what extent has the actual tooth profile (geometry) of gear (which had wear pattern) been established?

   (A) No information on this.

4. (Q) What are GE's (LSCC's) intended procedures and objectives during the reopening inspection of the gearboxes?

   (A) The lower port unit showed slight scuffing (or polishing, depending on interpretation) was scheduled by LSCC to be opened at two to ten hours foilborne operation depending on test schedule. Inspection will be visual check for water intrusion and general condition of parts, particularly bearings and gear tooth scuff pattern. Vibration data (from two pickups on gear cases) will be available for study (to date, pickups have been installed but no data recorded). It was pointed out that more gear units should be inspected due to high incidence of bearing failure. As an additional minimum, the two top units should also be opened. LSCC has not formally agreed to do this but McKernan (LSCC) said, "we will open them." It was emphasized that ten hours might be too long for visual inspection since it was desired to see the tooth contact developed under load on the blued teeth and observe for any other potential trouble at an early stage of operation. It was agreed that the inspection would be made after two to four hours of foilborne operation. LSCC agreed that hull operation with the foils down would be limited to ship operation preparatory to "flying." NAVSEC was particularly concerned about possible gear overloads resulting from thrust for displacement operation going up to hump exceeding that after hump and thus exceeding gear design loads.

5. (Q) What are principal stress levels and behavior of the bearing which caused bearing failure, and how has this been alleviated by the bearing fix?

(A) No specific information on actual stress levels available.

6. (Q) Have the plug roll-pins been installed per drawing (GE) 662E747?

(A) SUPSHIPS wanted confirmation that roll-pins are all in place. It could be verified that plugs are in place but pins are not visible. No verification is possible without disassembly. We must assume GE at Lynn followed drawing.

7. (Q) What is the stress concentration factor on the revised keyway notch?

(A) Not known.

8. (Q) What is the reason for the necessity of different sized shims to obtain assembly of port lower gearbox?

(A) Not discussed again as was covered in general background information and general coverage of one, two, three and four and may be cleared up by information to be requested from Lynn by local GE representative.

9. (Q) What is the history and conclusions of the requirements on backlash as a result of GE's gearbox failures?

(A) Local GE representatives' (who worked in Gear Department at GE plant) opinion is that insufficient backlash is the critical factor; if there is sufficient backlash (which usually begins at 0.020 in.), he doesn't believe it is important beyond that if there is good contact pattern. Question remains as to what stamped backlash on gears means and GE reported concern for it in prototype development. Further information from GE may clarify.

10. (Q) Has the axial alignment of the gear shafting (i.e., in gearboxes) been checked? When and how?

(A) Local GE representative assumes this was done in original production. It has not been verified locally (no facilities). He feels that the ability to get good uniform no-load contact tapes now in his shop shows there is no permanent deformation of gear boxes.

11. (Q) What procedure for reinstallation; inspection and alignment in reassembly and obtaining proper wear patterns will be followed?

(A) Technical Manual on gearboxes is now incomplete and too brief on this. At Navy request, LSCC will ask GE to supplement present information to include how to adjust gears and what correct tooth contact should look like from tape contact test.

Note: The manual for the hullborne transmission is more complete as regards assembly requirements and procedures, and it is essential that the foilborne gearboxes receive at least equal treatment so standard approved procedures are available for future guidance of maintenance personnel.

APPENDIX B

HORSEPOWER CAPACITY OF THE BEVEL GEARBOX USING ALTERNATE DESIGN LIMITS

1. Calculations of horsepower capacity are presented to determine the actual capacity estimated on the basis of present guidelines, and to determine the reduced gear capacity due to the observed fretting corrosion. The latter appeared on the pilot surfaces of the shaft during ship operation.

2. Horsepower capacity of AGEH-1 gearboxes based on the fatigue bending stress at the root of teeth, which may be assumed equal to 35,000 psi for 10^{10} cycles, is given by the equation:[45]

$$HP_b = \frac{S_{ab} \cdot \eta \cdot d \cdot F \cdot J \cdot K_v}{126,000 \cdot P_d \cdot K_o \cdot K_s \cdot K_m} \qquad (B.1)$$

where

HP_b = Maximum continuous horsepower based on bevel gear bending strength criterion

S_{ab} = Allowable bending stress for 10^{10} cycles

η_p = Pinion rpm = 1572

d_p = Pinion outer pitch diameter = 25.5 in.

F = Face width = 5.4 in.

J = Geometry factor = 0.32 [45]

K_v = Velocity factor = 1

P_d = Diametrical pitch = 2

K_o = Overload factor = 1

K_s = Size factor = 0.85 [45]

K_m = Load distribution factor = 1.1 for straddle mounted gears.[45]

Substituting the above values into Equation (B.1)

$$HP_b = 10,289/\text{mesh}$$

The torque capacity of bevel gears/mesh

$$T_{max} = \frac{10,289 \cdot 63,000}{1,572} = 412,945 \text{ lb-in.}$$

During the testing, the maximum torque imposed was 395,000 lb-in./mesh.

3. Horsepower capacity, based on allowable compressive stress near the pitch line of the tooth, which may be assumed to be equal to 200,000 psi for 10^{10} cycles, is given by Equation (B.2):[45]

$$H_c = C \cdot d_p^2 \cdot \eta_p \cdot F \cdot I \qquad \text{(B.2)}$$

where

H_c = Maximum continuous horsepower based on allowable compressive stress

C = Constant = 0.0368

C_o = Overload factor = 1

C_s = Size factor = 1

C_v = Dynamic factor = 1

C_m = Load distribution factor = 1.1 for straddle mounting

dp = Pitch diameter = 25.5 in.

η_p = Pinion rpm = 1572

F = Face width = 5.4 in.

I = Geometry factor = 0.092.

Substituting the above terms into Equation (B.2), we have

$$Hc = 18,688/\text{mesh}$$

Since the horsepower capacity based on the fatigue bending stress at the root is much lower than that based on compressive stress, the actual horsepower capacity of bevel gearing should be considered to be governed by the fatigue stress.

4. Horsepower Capacity of the Fretted Surface Shaft. During the testing of the gearboxes by GE, the most serious problem was the fretting corrosion which appeared on gear flanges. During the ship operation, more serious fretting corrosion appeared on pilot surfaces of the shaft. Consequently, the horsepower capacity must be governed by the strength of the shaft, not by gear capacity.

Figure B.l shows the principal dimensions of the shaft and of the bevel gear assembly required to calculate torque capacity and the stresses in the shaft.

The forces transmitted by the gear induce a maximum bending moment in the shaft in the cross section 5.5 in. from the bearing centerline. Due to the shaft rotation, the bending moment should be considered as an alternating stress. In addition, the shaft is subjected to torque inducing a shearing stress.

Assuming that alternating torque is equal to 20 percent of the maximum torque, we must distinguish between a steady shearing stress induced by the steady torque and an alternating shearing stress induced by the alternating component of the torque. In consequence, the shaft will be subjected to a steady stress, S_s, and an alternating combined bending and shearing stress, S_a, having components, S_{ba} and S_{sa}. All steps required to calculate the above mentioned stresses, in order to determine the factor of safety of the unfretted shaft by means of Goodman Diagram, will follow. Finally, assuming the line of failure of the fretted shaft the torque capacity will be calculated.

Assuming the maximum continuous horsepower of the gas turbine = 14,000 and propeller rpm = 1572:

$$\text{The propeller torque } T = \frac{63,000 \times 14,000}{1,572} = 561,069 \text{ lb-in.}$$

The tangential force at the mean diameter/gear (Figure B.l):

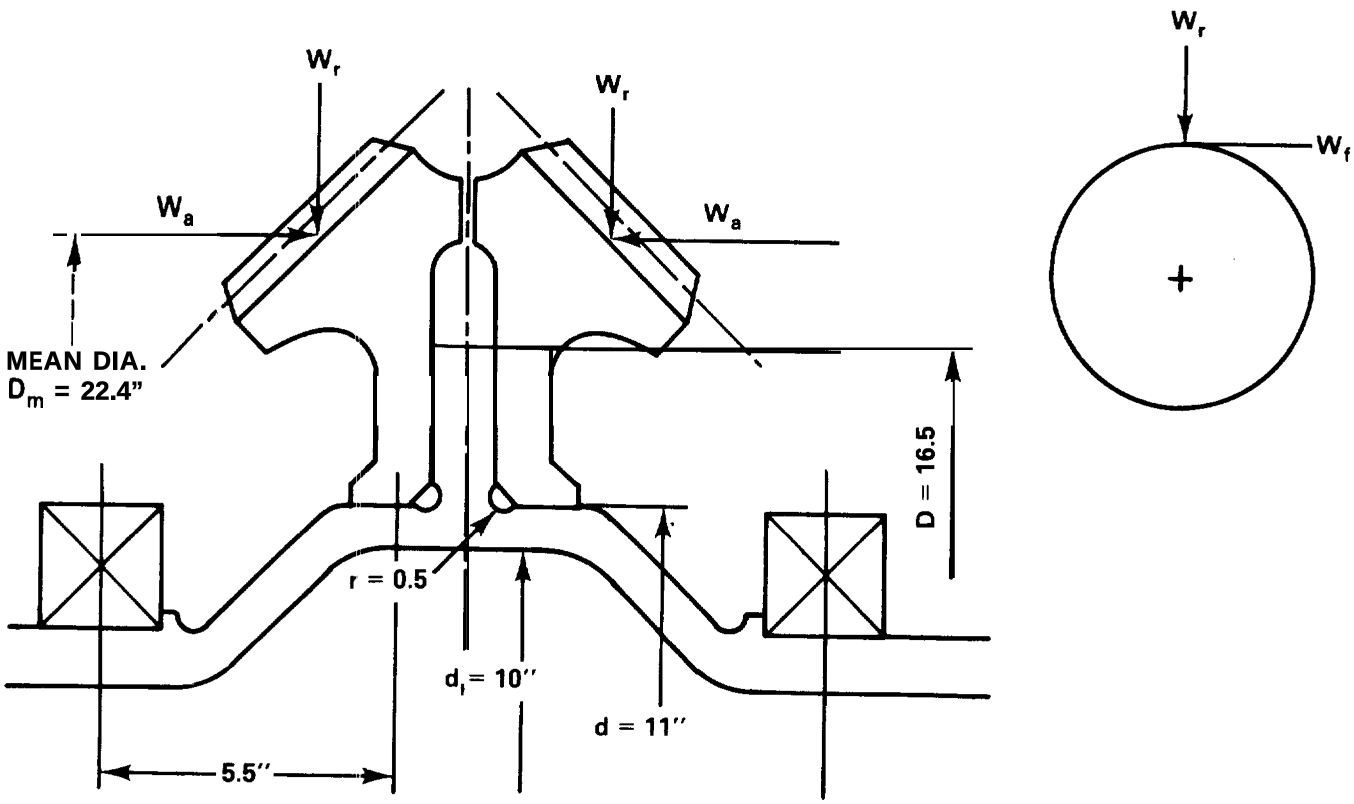


Figure B.1 - Bevel Gears Assembly.

$$W_t = \frac{561{,}069 \times 2}{22.4 \times 2} = 25{,}048 \text{ lb}$$

The axial force at mean diameter

$$W_a = 0.7 \times W_t = 0.7 \times 25{,}048 = 17{,}533 \text{ lb}$$

The radial force at mean diameter

$$W_r = 0.11\ W_t = 0.11 \times 25{,}048 = 2{,}755 \text{ lb}$$

The bearing load

$$W = \sqrt{25{,}048^2 \times 2{,}755^2} = 25{,}200 \text{ lb}$$

Bending moment at the fretted zone

$$M_b = 5.5 \times 25{,}200 = 138{,}600 \text{ lb-in.}$$

The alternating bending stress is given by:

$$S_{ba} = \frac{M_b \cdot r}{I} \tag{B.3}$$

where

r = Shaft: radius, in. = 5.5 in.

I = Moment of inertia of the shaft cross section = $0.049\ (11^4 - 10^4)$ = 227 in.^{4}

$$S_{ba} = \frac{138{,}600 \times 5.5}{227} = 3{,}358 \text{ psi}$$

The steady shearing stress:

$$S_s = \frac{T \cdot r}{J} \tag{B.4}$$

where

J = Polar moment of inertia of the shaft cross section = 2 I

$$S_s = 561{,}069 \quad 2 \times 227 \times 5.5 = 6{,}797 \text{ psi}$$

This stress is plotted on the horizontal axis of the Goodman Diagram, Figure B.2. The shearing stress induced by the alternating torque

$$S_{sa} = 0.2 \times S_s = 0.2 \times 6797 = 1359 \text{ psi}$$

The combined bending and shearing alternating stress

$$S_a = \sqrt{(K_t \times S_{ba})^2 + 3(K_s \times S_{sa})^2} \qquad (B.4)$$

where

K_t = Stress concentration factor at bending = 1.87 for r/d = 0.05 and D/d = 1.5

K_s = Stress concentration factor at torsion = 1.7[46]

$$S_a = \sqrt{(1.87 \times 3358)^2 + 3(1.7 \times 1359.4)^2} = 7445 \text{ psi}$$

This stress is plotted on the vertical axis of Figure B.2.

Endurance limit S_e of the AISI-4340 steel for combined alternate bending and shearing has been assumed = 40,000 psi for unfretted steel and 10,000 psi for the fretted zone,[47] yield point for the steady stress = 100,000 psi. Point A on Figure B.2 shows the condition of the stresses in the shaft. For the unfretted surface, the factor of safety is about 4, while for the fretted zone only 1.2.

Assuming the factor of safety in case of fretting equal to 4, the allowable shearing stress S_{sf} is as follows:

$$S_{sf} = \frac{1.2}{4} S_s = \frac{1.2}{4} \times 6797 = 2039 \text{ psi}$$

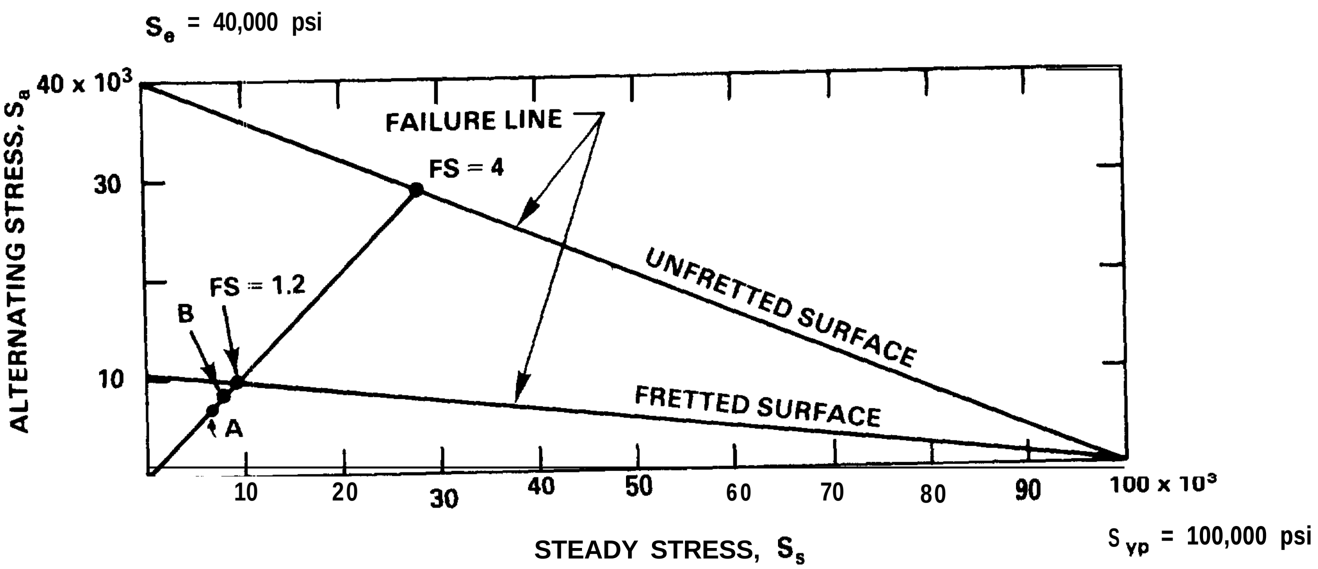


Figure B.2 - Goodman Diagram for the Shaft

The allowable torque that may be transmitted by the shaft in case of fretting

$$T_f = \frac{S_{sf} \times J}{r} = 2{,}039 \times \frac{459}{5.5} = 168{,}310 \text{ lb-in.}$$

and the maximum horsepower capacity of one bevel gearbox is

$$HP_{max} = \frac{168{,}310 \times 1{,}572}{63{,}000} = 4{,}200 \text{ hp}$$

As the last alternate design limit, consider that the fretted zone has been eliminated by removing 1/16 in. on the radius and the alignment of the gears with respect to the shaft has been provided by body-bound bolting.

The new shaft diameter of the shaft d = 10.875 in.

I = The moment of inertia of the shaft cross section

$$I = 0.049\ (10.875^4 - 10^4) = 195 \text{ in.}^4$$

The bending alternate stress

$$S_{ba} = \frac{138{,}600 \times 5.438}{195} = 3{,}865 \text{ psi}$$

The polar moment of inertia

$$J = 2\ I = 2 \times 195 = 390 \text{ in.}^4$$

The steady shearing stress

$$S_s = \frac{561{,}069 \times 5.438}{390} = 7{,}823 \text{ psi}$$

The shearing stress induced by the alternating torque

$$S_{sa} = 0.2 \times 7{,}823 = 1{,}565 \text{ psi}$$

The combined bending and shearing alternate stress on the basis of Equation (B.3)

$$S_a = \sqrt{(1.87 \times 3865)^2 + 3(1.7 \times 1565)^2} = 8570 \text{ psi}$$

The point representing the condition of stresses in the shaft is now Point B, Figure B.2. The factor of safety with respect to stresses at Point B is about 3.55.

Assuming the factor of safety equals 4, as in the case of the fretted shaft material, the steady shearing stress

$$S_s = 8570 \times \frac{3.55}{4} = 7605 \text{ psi}$$

The shaft torque capacity based on the above stress:

$$T = \frac{7{,}605 \times 390}{5.438} = 545{,}475 \text{ lb-in.}$$

and the horsepower capacity of the gearbox

$$HP = \frac{545{,}475 \times 1{,}572}{63{,}000} = 13{,}610$$

If the AGEH-1 gearboxes (or similar designs) are to be used in the future, certain steps should be taken to eliminate or alleviate the fretting corrosion. The fretting corrosion was believed to be caused by the membrane mode vibration of the gear flanges and torsional vibration of the gear assembly. This hypothesis was supported by test results of similar bevel gears built by Gleason in 1964 for a 500-ton hydrofoil concept. The recommendation to introduce the body-bound bolting, proper distribution of the compressive stress over the contact area on the flanges and stiffening of the gear assembly by adding clamp straps under the bolting, were proposed by the Navy in 1975.[48] These recommendations were supported in the GE final report.[4]

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