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Common Problems and Pitfalls in Gear Design

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COMMON PROBLEMS AND PITFALLS IN GEAR DESIGN

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SUMMARY

There are several pitfalls and problems associated with the successful design of a new gear transmission. A new gear design will require the knowledge and experience of several technical areas of engineering. Most of the pitfalls and problems associated with a new design are related to an inadequate evaluation of several areas such as, the lubrication and cooling requirements, complete static and dynamic load analysis, evaluation of materials and heat treatment and the latest manufacturing technology. Some of the common problems of the gear design process are discussed with some recommendations made for avoiding these conditions.

INTRODUCTION

Gear problems are a common occurrence in the gear industry and are often the results of improper design, a wrong selection of material for a given application or a lubrication system that was not adequate for the conditions encountered. Many times a noise or gear vibration condition will appear as an unexpected problem and may require considerable time and expense to correct. There are quite a number of gear consultants who are kept busy with gear problems. Not all problems are design related but may be the result of some unavoidable or unexpected system condition. It is always good practice with a new design to have experienced engineers or consultants conduct a design review of the system. Years of experience tends to alert one to some of the more common problems associated with the gear systems. The objective of this paper is to point out a few of the more common problems encountered with gear systems and offer ways to avoid at least some of them.

Design Considerations

Sometimes problems are caused by failure to select the best type of gear for a given application. All the design considerations associated with a particular gear type may not have been applied during gear selection. Several important parameters should be considered when selecting a gear type for a given system. For a parallel axis system, there is the possibility of using spur, helical, double helical, or other special types such as Wildhaber Novikov gears. Different contact ratios may also be considered for different applications.

Generally spur gear designs are limited to pitch line velocities of 50 m/sec (10 000 ft/min) or lower. However there are application where spur gears have been operated successfully at over 100 m/sec (20 000 ft/min) which shows that it can be done with the right choice of design parameters. When spur gears are operated over 50 m/sec (10 000 ft/min) problems such as noise, high dynamic loads, and air or oil trapping, can cause excessive wear or reduced life.

Helical gears are preferred for use at the higher speeds because there is a high total contact ratio which reduces the noise and dynamic loads. The air and oil can be pumped off the ends of helical gear teeth without producing high loads or excessive noise. Helical gears have thrust loads and overturning moments that must be accounted for. In some lightweight applications it is difficult to reduce the deflections caused by the overturning moment, to an acceptable level. Many helical gear designs use thrust rings as shown in figure 1. These thrust rings when properly designed can take the thrust load and moment loads so that there are no external forces that require thrust bearings and heavy moment supporting structure. Space should be provided between the end of the teeth and thrust ring to allow the oil or air from the mesh to escape easily. The thrust ring should have a radius to increase the formation of an EHD oil film by wedging action between the thrust ring and the ends of the gear teeth.

Double helical gears are often used to eliminate thrust and moment loads of single helical gears. A double helical can cause noise and high dynamic loads if not properly manufactured. A double helical gear will shift axially to adjust for spacing error and helix error between the two sides of the gear. If these errors are very large the result will be noise and high dynamic loads. Because of this condition it is very difficult to manufacture a planetary gear system using double helical gears. As each planet gear tries to adjust for the sun gear and ring gear at the same time, severe vibration will be generated.

Accuracy

Selection of the right gear accuracy for an application is very important as it effects the cost of manufacturing, the dynamic load and noise. The affect of accuracy on dynamic loads and noise are increased as the speed is increased. Reference 1 indicates that doubling the speed of a gear set increased the noise level by approximately 6 dB. Therefore a moderate to high speed gear set should have an AGMA tolerance class of 10 to 13 for low noise level and reduced dynamic loads.

The highest gear accuracy is usually obtained by finish grinding on a good gear grinder. Good quality gears that are not too hard (less than RC 45) can be obtained by shaving. Hard gears (greater than RC 50) can also be finish cut to good accuracy by skiving with a carbide skiving cutter.

Gear Rim Thickness Considerations

Gears are often designed using empirical methods for some parameters such as rim thickness. One design practice that has been used is to make the gear rim thickness equal to the tooth height. This practice has been very successful for gears that are fairly small in diameter compared to the tooth height. When the gear diameter is small the rim is very stiff and will have no low frequency vibration problems. However, when the diameter of the gear is large compared to the tooth height the rim becomes much more flexible and is subject to low frequency vibration modes. The increased flexibility may also cause an increase in bending stress due to the gear tooth moment combined with the rim flexure. A better approach to determine rim thickness would be to use a constant ratio function of r/t so that the rim thickness t increases proportionally to the gear radius r. Webs can be used to increase the effective rim thickness without causing a large increase in gear weight.

Gear Drawing and Specifications

There are many cases where a particular gear with a required tolerance and material condition has been requested from a gear manufacturer but the finished product was not the same as that required. This is primarily the result of inadequate gear drawings and specifications. It is not enough today to send a gear drawing with an AGMA class requested and a note that says carburize and harden.

The drawing should include with the gear tolerance a profile chart similar to figure 2 with specification on profile modification and tolerance. The drawing should also have a note listing inspection required and a list of material and heat treatment specifications which should be a part of the requirements. The material and heat treatment specification should be very specific as to the type of material, material certifications, heat treatment with requirement for certification and sample used for checking the method used. In addition to a good heat treatment specification, a good source should be selected that will do what is required in the specification and provide the required controls and record to verify the procedure. A material sample should be supplied with the finished gears with case depth and hardness verified on the sample and a report delivered with the gears giving a certification of all the required conditions.

Gear Vibration and Dynamic Loads

When a gear system is designed it should be evaluated for dynamic loads and resonant operating condition. There are many times when a gear system has failed because of high dynamic loads from the unexpected vibration of the systems. It is usually not enough to add a dynamic factor per AGMA 218.01 (ref. 2) since these factors are general in nature and do not allow for resonant conditions that may exist. A vibration analysis should be used if available to predict vibration problems. There are several types of vibration conditions that may be encountered in a gear system. There may be shaft vibration, gear rim or web resonant vibration, shaft torsonial vibration coupled with the gear teeth and housing vibrations. The gear transmission should be evaluated for these vibration conditions before and after it is installed in the system. Testing of the gear system should be accomplished with adequate instrumentation to show any vibration problem that might exist in the operating range. If a gear system should happen to have a resonance near the operating speed a rapid failure of the gears could occur. Figure 3 is a plot of a gear system showing one of the types of resonant condition that could occur over a speed range. Gears with fairly thin rims are particularly susceptible to ring vibration modes which can cause high stress leading to failure. Any gear ring should be checked for natural frequencies and these should be avoided in the operating speed range. It is often good practice to use vibration damper rings on gear rings that have vibration problems as shown in figure 4. These rings are designed to rub against the gear rim at a different frequency and thus provide damping. The damping thus provided will considerably reduce the peak stresses at the resonant conditions. Sometimes it may be necessary to change the weight of a gear to move its resonant vibration away from the gear operating speed.

Gear Life

When a gear set is designed is it normally required to have a life that will be equal or better than the other equipment in the system. The two areas of life of concern to the gear designer are the bending fatigue life and the surface fatigue life, assuming there is no significant wear. The bending fatigue life can be designed for essentially an infinite life by keeping the bending stress well below the stress that will give 10^7 cycles before failure, figure 5. For reverse bending application such as epicyclic gears and idler gears the stress should be further reduced by approximately 20 percent.

In surface fatigue life there is no stress condition that will give infinite life. Therefore in applications where a large number of cycles are accumulated in a short period the stress must be reduced to account for this. At higher stress (life to approximately 10^6 cycles), test data reference 3 indicates that the life is inversely proportional to stress to the ninth power (fig. 6). $L_2 = L_1 (S_1/S_2)^9$. There is insufficient data available at lower stress levels (above 10^7 cycles) but there are indications that the life in this region is inversely proportional to stress to the 20th power (ref. 2, fig. 7) that is $L_2 = L_1 (S_1/S_2)^{20}$. This type of data is very expensive to obtain because of the long hours involved in running tests on several specimens.

When long surface fatigue life and moderate to heavy loads are required, a case hardened gears material must be used. Gear teeth that are below a hardness of Rockwell C 58 will have short surface fatigue life at maximum Hertz stresses above approximately 150 Ksi.

Materials

Many times a gear material is selected for an application which does not give the required surface durability or bending strength. Some steel materials that are adequate for small gears may not heat treat properly when used in a larger gear. Some lower alloy steels that will harden well in small sizes will remain soft with reduced strength when used for larger gears because it takes longer to quench. There is a wide variety of material available to the gear designer. Proper selection is required to meet the demand of low cost and reliability. Figure 8 from reference 4 gives a general cost level for various gear materials. There are many cases when plastic gears will perform very well because the loads and temperatures are acceptable. Soft steel gears are used for many applications but are generally much larger and weigh more than a comparable case hardened gear. When weight size and reliability are important then a case hardened gear is usually the best choice. Most case hardened gears require finish grinding after heat treatment. However, nitriding steels can be case hardened by nitriding with very little distortion and may not require a grinding operation after heat treatment.

Heat Treatment

Successful heat treating requires two things, a good heat treatment specification and a heat treating source that will follow the specification and do a good job of heat treating the material. All too often specifications are not followed rigorously resulting in poor quality heat treatment and soft or distorted gear. Most carburized and hardened gears require a good quenching die to prevent excessive distortion. Larger, and more complicated parts require more care in designing the quenching die for best results.

At corners and sharp points care must be taken in carburizing and nitriding gear teeth. Sometimes these areas will crack because of the surface compressure stress built up in the material from the carburizing or nitriding. A typical corner crack is shown in figure 9 (ref. 5). This was eliminated by masking the ends and tips of the teeth to limit the amount of carbon near the corner of the tooth. The same type of thing can happen on narrow pointed gear teeth with nitriding. These type problems can be avoided by masking areas that do not required hardening such as the ends and of the teeth thus reducing the corner stresses.

Adequate case depth must be obtained for gears to prevent case crushing as shown in figure 10. Reference 6 gives a good treatment of case depth for carburized gears to prevent case crushing. Generally the case depth should be well below the depth to the maximum subsurface shear stress to prevent case crushing.

Lubrication

Many gear system problems have been generated because of an inadequate appreciation of lubrication methods and requirements for gearing. Gears like rolling element bearing require a very small amount of lubricant for lubrication, however much more lubricant is needed for removing heat from the gears. Several methods are used to lubricate and cool gears. The most common is splash lubrication when one gear dips into a reservoir of oil in the bottom of the gearbox and throws the oil onto the other gear and bearings. With this method care must be taken to assure that an adequate flow of oil is fed to the bearings for good cooling. If this is not done bearing failure will occur because of insufficient cooling and not enough EHD oil film. The most common method used to supply oil to the bearing with the splash lubrication method is to provide a channel to catch the oil splash and feed it to the bearings. When gears operate at higher speeds and loads the splash method of lubrication and cooling will generally not provide the cooling required for the gears. With inadequate cooling, early failure will occur. The failures may be due to scoring, wear, surface fatigue, or tooth breakage. The main cause for these failures sometimes is not recognized as over temperature of the gears from inadequate cooling. The first indication of excessive temperature of the gears may be superficial pitting or grey staining caused by a reduced EHD film thickness as the oil viscosity is reduced. In many applications the designer will use oil jet lubrication with the oil jet directed at the into mesh position or at the out of mesh position. Neither of these methods provide the most effective cooling of the gears and could in many cases allow overheating of the gears. The into mesh method of oil jet lubrication will considerably reduce the efficiency of the gears since the oil going into mesh increases oil churning losses. The into mesh and out of mesh jet lubrication method gives very shallow impingement depth on the gear teeth (refs. 7 and 8) and therefore neither method provide good cooling. Figures 11 and 12 from reference 9 show oil jet impingement for into mesh and out of mesh lubrication.

Much better gear tooth cooling is obtained when the oil jet is directed radially onto the gears with adequate jet velocity. Figure 13 from reference 10 shows a radially directed oil jet impinging on a gear tooth. Here good cooling is obtained because the oil jet is cooling the gear tooth where the heat is generated.

Noise

Industry standards today are requiring that gear systems must meet a noise specification. These requirements usually mean that the designer has to know how to design gears that will operate at a noise level of 85 dB or less. Unless special care is taken in the design and manufacture of the gears, the noise level may be excessive.

There are several things that contribute to gear noise. Some of these have more effect than others. Profile variations seems to have the greatest effect on gear noise, especially at the gear mesh frequency. Spacing errors, runout, lead error, poor surface finish, the wrong amount of profile modification will increase the gear noise. Reference 11 describes how the noise of a gear set was reduced 5 dB when profile modification, (i.e., tip relief) was added to an unmodified gear set. Increasing load or speed will usually cause an increase in gear noise (ref. 1). Therefore high speed gears will require more accuracy and careful profile modification. It is good practice to require that the gear box meet the noise requirement at operating load and speed. A gear box which is very quiet at low loads may become very noisy at full design load. Figure 14 is a vibration spectrum of a double helical gear box operating at moderate load. This gear box was very quiet at light load but became very noisy at moderate loads.

CONCLUDING REMARKS

Many of the more common problems that occur in gear transmissions can be avoided in the design and manufacturing process. The gear type for the particular application should be selected and the accuracy requirements determined that will give the desired results with the minimum cost. For lightweight designs rim thickness should be carefully evaluated for vibration problems and maximum bending stresses.

The drawing and specifications should be specific and include all the required information to assure delivery of the correct end product. Bending and surface fatigue design life should be adequate for the application.

The gear material and heat treatment are very important to the life and cost for a given application and should be carefully determined. The lubrication system must be designed to provide good cooling and lubrication for the gears to prevent early failure. In recent years noise has become an important design consideration and must be evaluated to assure that the gear system meets the requirements.

Careful attention to the above mentioned design factors will provide a gear system with less problems and longer life.

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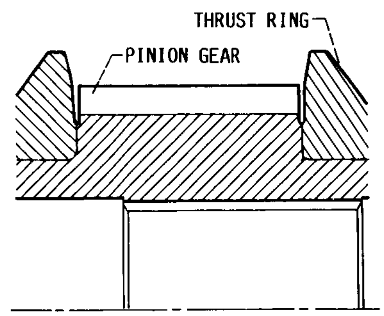


FIGURE 1. - HELICAL GEAR WITH THRUST RINGS.

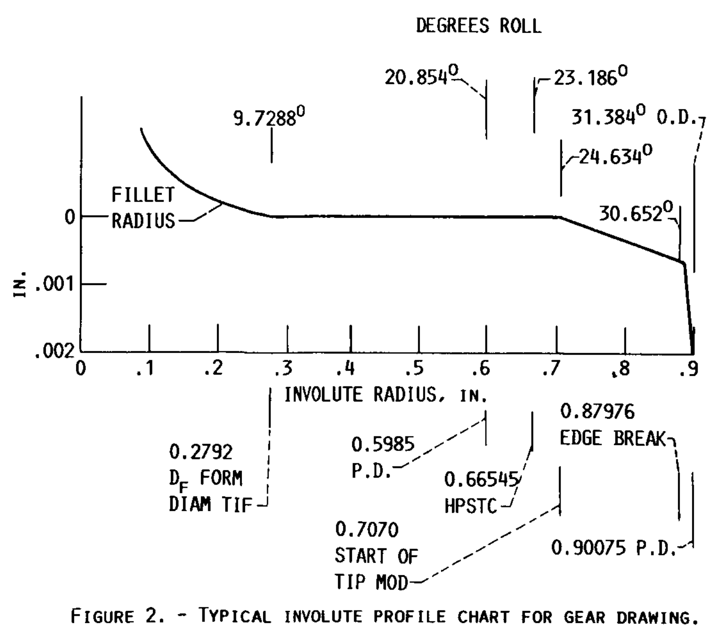


FIGURE 2. - TYPICAL INVOLUTE PROFILE CHART FOR GEAR DRAWING.

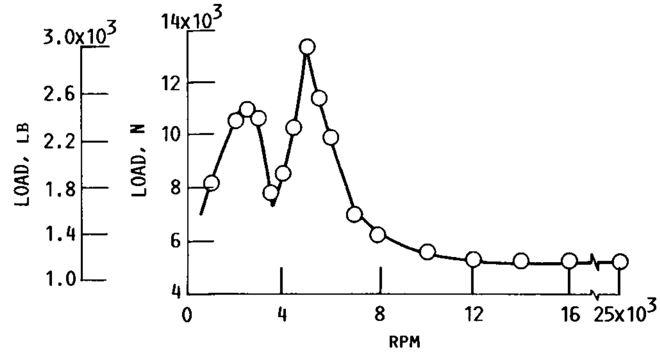


FIGURE 3. - DYNAMIC LOAD VERSUS SPEED FOR GEAR SET.

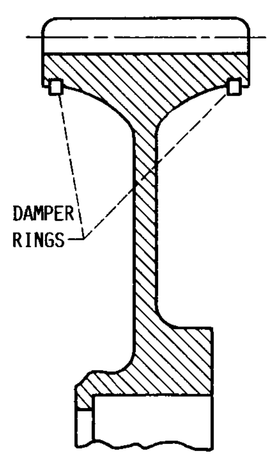


FIGURE 4. - GEAR WITH VIBRATION DAMPER RINGS.

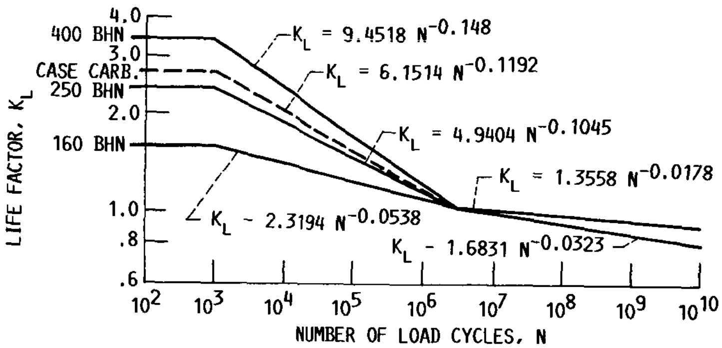


FIGURE 5. - BENDING STRENGTH LIFE FACTOR, K . (NOTE: THIS CURVE DOES NOT APPLY WHERE A SERVICE FACTOR, K_{SF}, IS USED.)

NOTE: THE CHOICE OF K_L ABOVE $3x10^6$CYCLES IS INFLUENDED BY:

PITCHLINE VELOCITY

GEAR MATERIAL CLEANLINESS

RESIDUAL STRESS

GEAR MATERIAL DUCTILITY AND FRACTURE TOUGHNESS

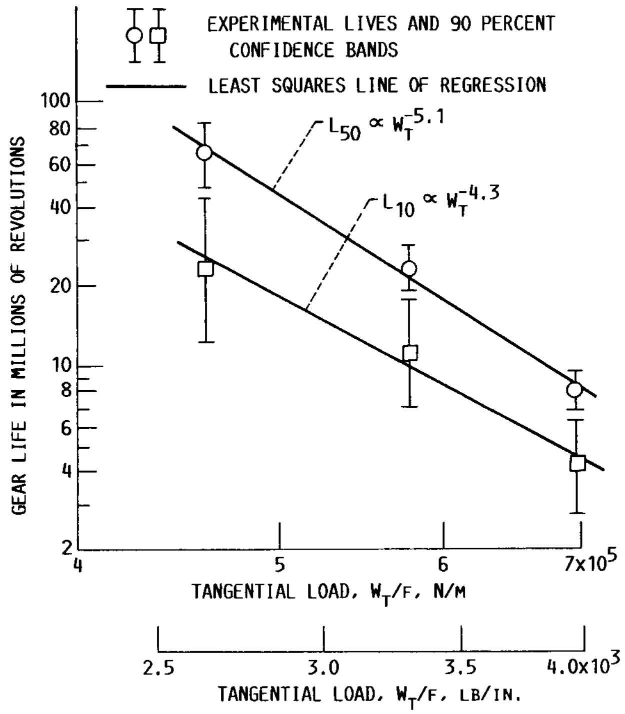


FIGURE 6. - LOAD LIFE RELATIONSHIP FOR (VAR) AISI 9310 STEEL SPUR GEARS SPEED 10 000 RPM, LUBRICANT SUPER-REFINED NAPHTHENIC MINERAL OIL WITH ADDITIVE PACKAGE.

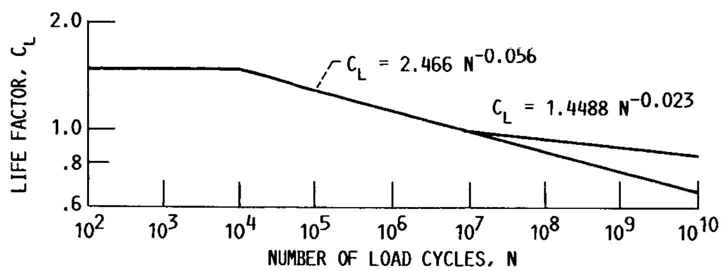


FIGURE 7. - PITTING RESISTANCE LIFE FACTOR, C_L. (NOTE: THIS CURVE DOES NOT APPLY WHERE A SERVICE FACTOR, C_{SF}, IS USED.)

NOTE: THE CHOICE OF C_L ABOVE 10^7 CYCLES IS INFLUENCED BY:

LUBRICATION REGIME
FAILURE CRITERIA
SMOOTHNESS OF OPERATION REQUIRED
PITCHLINE VELOCITY
GEAR MATERIAL CLEANLINESS
MATERIAL DUCTILITY AND FRACTURE TOUGHNESS
RESIDUAL STRESS

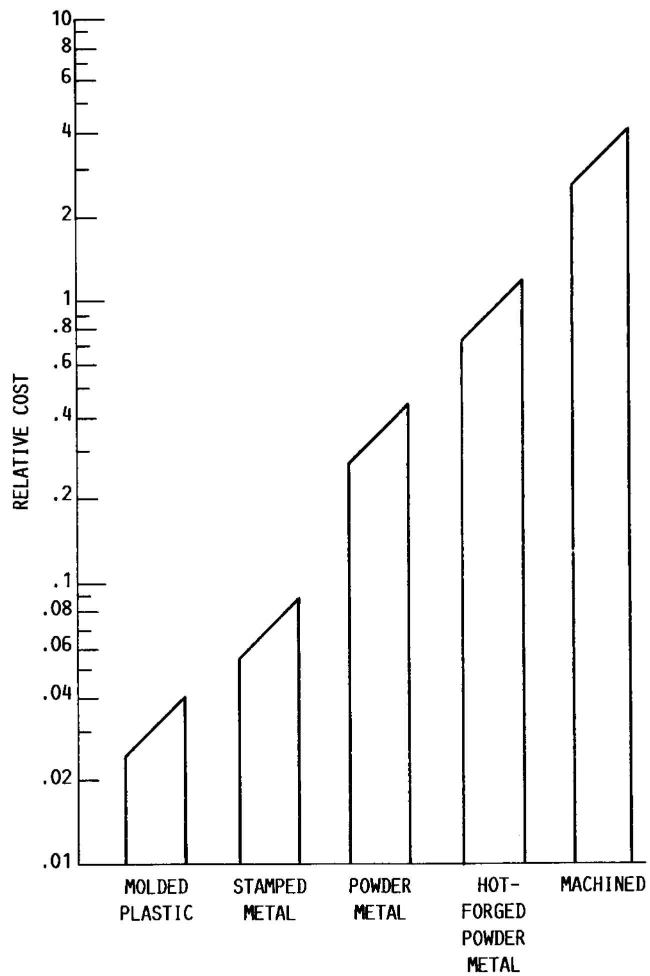


FIGURE 8. - RELATIVE COSTS OF GEAR MATERIALS.

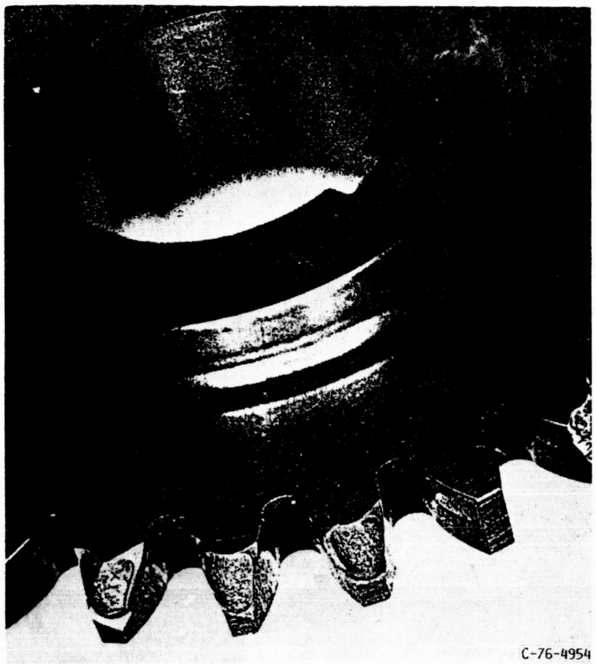


FIGURE 9. - TYPICAL FRACTURE OF MODIFIED VASCP X-2 GEAR TEETH WITH NASA SPECIFIED HEAT TREATMENT AND CARBURIZED ON ALL SIDES.

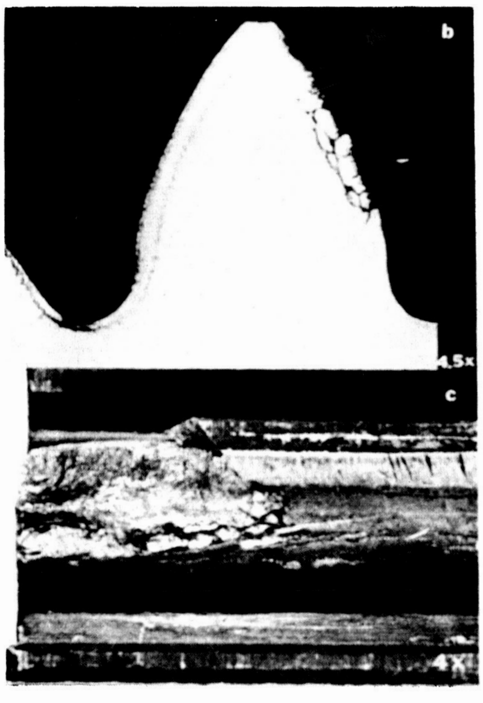
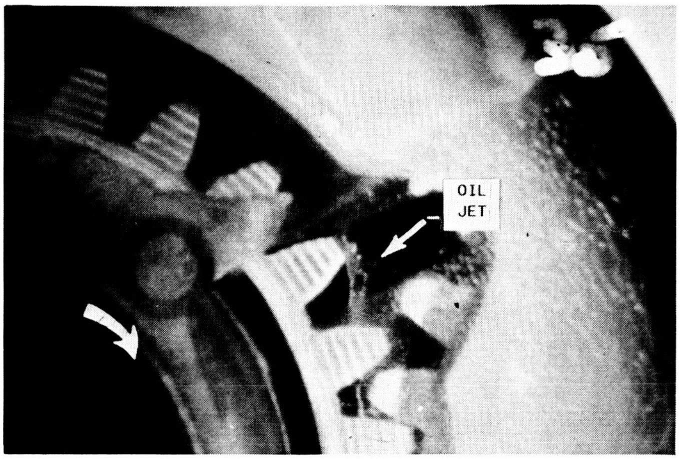
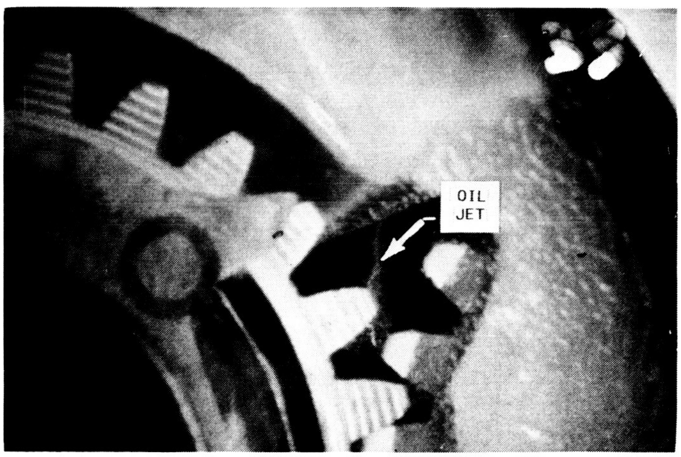


FIGURE 10. - CASE CRUSHING OF CASE HARDENED GEAR TEETH.

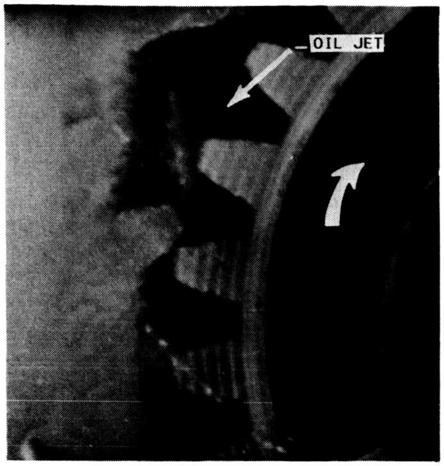


(A) INTO MESH LUBRICATION FOR $V_J > V_G$.

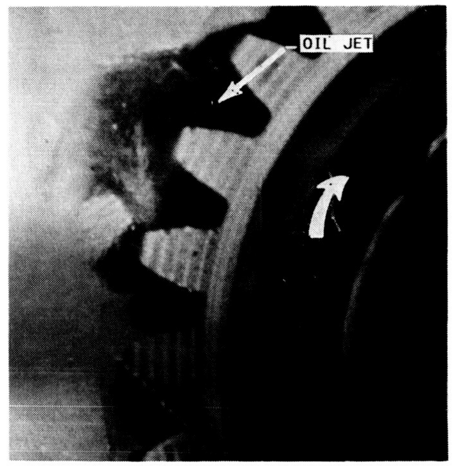


(B) INTO MESH LUBRICATION FOR $V_J < V_G$.

FIGURE 11. - INTO MESH JET LUBRICATION OF GEARS.



(A) OIL JET CLEARING PINION TEETH.



(B) OIL JET CLEARING GEAR TEETH.

FIGURE 12. - OIL JET IMPINGEMENT DEPTH, OUT OF MESH. SPEED, 3600 rpm ; JET PRESSURE, 8.3×10^4 N/M^2 (12 psi).

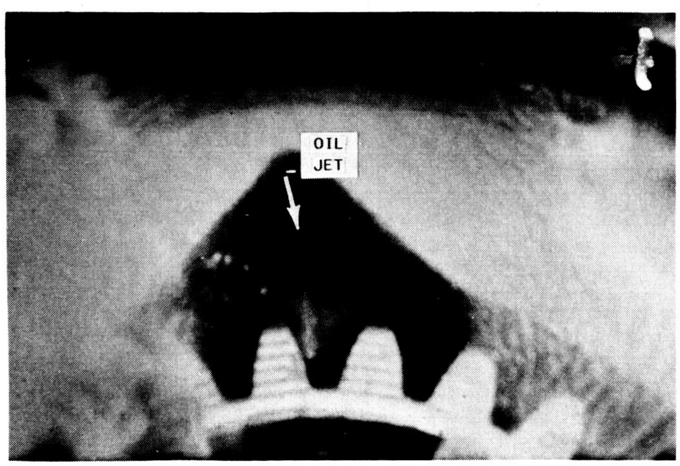


FIGURE 13. - RADIAL JET LUBRICATION 5000 rpm OIL JET PRESSURE
21 N/CM^2 (30 psi).

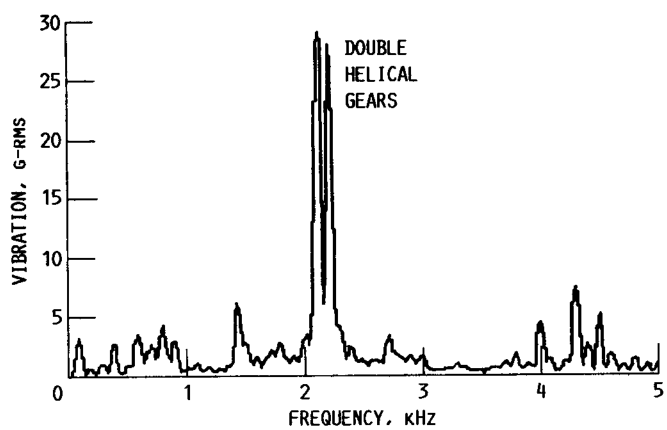


FIGURE 14. - SPECTRUM ANALYSIS OF DOUBLE HELICAL GEAR SET.

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