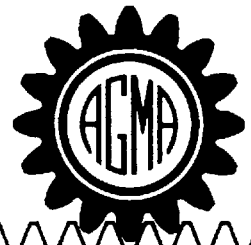


97FTM10

UltraSafe Gear Systems for Critical Applications—Initial Development

by: Raymond J. Drago, Ashok D. Sane, Frederick W. Brown,
Boeing Defense & Space Group, Helicopters Division



TECHNICAL PAPER

UltraSafe Gear Systems for Critical Applications–Initial Development

**Raymond J. Drago, Ashok D. Sane, Frederick W. Brown,
Boeing Defense & Space Group, Helicopters Division**

[The statements and opinions contained herein are those of the author and should not be construed as an official action or opinion of the American Gear Manufacturers Association.]

Abstract

Current design methodologies as applied to rotorcraft transmissions yield reliable, safe, and weight effective systems. In general, weight and performance (i.e., torque transmission per pound of transmission weight) are the primary active variables while safety is addressed by monitoring the design so that conventional criteria, such as calculated stresses, are kept within established allowable levels. This procedure is almost universally applied and has been quite successful both at Boeing Defense & Space Group, Rotorcraft and at other major rotary wing firms.

It is possible, however, to raise these already good safety characteristics to ultra safe levels through the use of certain design approaches. UltraSafe gear system design will require a basic change in the overall philosophy of design as it is practiced today. This change must be revolutionary rather than evolutionary.

In order to develop an UltraSafe transmission a basic paradigm shift will be required. In addition to addressing the design process from the typical “prevent a failure from occurring” point of view, it will also be necessary to view the system from the point of view of “what happens when a failure occurs”.

The current State-of-Technology is such that the concept of safety is addressed during the design process in an indirect rather than direct manner. Major design goals are achieved by designing components so that they meet the applicable allowable stress levels in order to insure that the possibility of a failure occurring is reduced to a finite but acceptable level.

Since it is not possible to design, manufacture, build or use any machine with absolute 100% certainty that a failure, at some point, for some cause, in some manner, will not occur, the design process must address the problem of failure in a more direct manner. The current State-of-Technology does not adequately address this facet of the design process.

The objective of this program is the development of a set of design guidelines and specific design techniques which will make a dramatic improvement in the failsafe operation of helicopter and tiltrotor aircraft transmissions while not significantly affecting either performance or weight characteristics. This paper presents the initial results of an ongoing program aimed at achieving this objective.

Copyright © 1997

American Gear Manufacturers Association
1500 King Street, Suite 201
Alexandria, Virginia, 22314

November, 1997

ISBN: 1-55589-704-5

UltraSafe Gear Systems For Critical Applications Initial Development

Raymond J. Drago, PE
Technical Fellow

Ashok D. Sane
Staff Engineer

Frederick W. Brown, PE
Staff Engineer

Boeing Defense & Space Group Helicopters Division

1.0 INTRODUCTION

The objective of the UltraSafe gear design development program is to develop a set of design guidelines and specific design techniques which will make a dramatic improvement in the failsafe operation of critical gear systems particularly those used in helicopter and tiltrotor aircraft transmissions while not significantly affecting either performance or weight characteristics. The activities required towards accomplishing this objective will be conducted in several logical sequential phases. As a first phase of this program, finite element models were developed and analyzed for several conceptual gear design configurations to evaluate stress distribution under applicable gear loads. Preliminary candidates (gear geometries) selected on the basis of predicted stress distributions will be further evaluated by crack propagation analysis to identify design configurations in which the direction of predicted crack growth is such that a failsafe operation of helicopter is feasible.

It is extremely important to recognize that the basic premise of this development is **NOT** to design more reliable gears which will not fail. This aspect of the design process, while important in its own right, is adequately addressed in the normal design process. UltraSafe gearing addresses the problem of what happens when, not if, a failure occurs.

The next phase of the program is to build and test prototypes of selected design configurations to validate analytical predictions to be followed by required evaluations leading to development of design techniques and guidelines for developing UltraSafe gears. This effort will be reported in subsequent publications.

This paper summarizes the preliminary work performed under the first phase of this program. The scope of this work included development of initial gear design geometries and their finite element stress analysis and evaluation.

2.0 CONCEPTUAL DESIGN CONFIGURATIONS

The present study is limited to spur gears, however, the basic principles developed apply equally well to all types of gears, including spiral bevels. A schematic of the initial configurations developed for this study is shown in Figure 2-1. It shows a conventional gear geometry and two alternatives identified by Modifications 1 and 2. The prime objective of these modified geometries is to control crack propagation in a desired direction

such that at least 50% of the gear face remains operational should a failure occur. Modification 2 is the equivalent of two completely separate gears, each mounted on the same shaft. If the shaft is very conservatively designed, as is usually the case simply due to deflection and geometry considerations, then either "gear" could fail completely and the gear would still function, albeit at double the unit tooth loading. Under double unit tooth loading, the remaining "gear" would not have infinite life but it would run for a very substantial period of time (many, many hours, at least) before a catastrophic failure occurred. During this time, the vibration characteristics of the gear would change markedly and it would generate considerable debris thus existing vibration and debris detection systems could easily detect the failure allowing the gear drive to be shut down in an orderly manner. In the case of a helicopter or tiltrotor, of course, this would allow the flight to continue to a safe landing. It should be obvious, however that modification 2 is not really practical for most gear systems, especially weight critical rotary wing applications. Conversely, modification 2 is only a mild departure from conventional design and could be configured to provide results similar to those of Modification 2 under actual operating conditions. The width and depth of the groove and the fillet radius at the groove bottom can be varied under Modification 1 to evaluate several gear geometries. Optimization of these parameters, as well as the design of the teeth themselves, is the focus of the initial UltraSafe effort.

For a realistic evaluation, the geometry/parameters of the existing gear configuration shown in Figure 2-1 were derived from the first stage sun gear of a main rotor transmission. The primary objective of the forward transmission is to transmit engine shaft power to the forward rotor. Secondary functions include rotor position interlock, single and twin engine override, and accessory drive. The first stage sun gear has 28 teeth with a pitch diameter of 5.89 inches, tooth height of .51 inch and a design speed of 3930 rpm. It accomplishes a gear reduction ratio of 4.786 with the four first stage planetary gears. The dimensions and parameters of the existing gear (Gear 1) analyzed in this study are shown in Figure 2-2 and Table 2-1. While most of the gear geometry is identical to that of the forward transmission sun gear, we replaced the common integral shaft of the sun gear that also includes a spiral bevel gear with a combination of .52 inch thick rim, .25 inch thick web and an integral .40 inch thick shaft as shown in Figure 2-2. We also placed two bearings to support the shaft at 2.0 and 3.3 inches, respectively from the center of the web. These

simplifications were made to simulate a realistic gear for this analysis while eliminating complex geometry not relevant to this study. Testing planned for the coming year will utilize a very similar model but with some additional modifications to fit the single tooth fatigue test rig which will be used for the planned crack propagation test program.

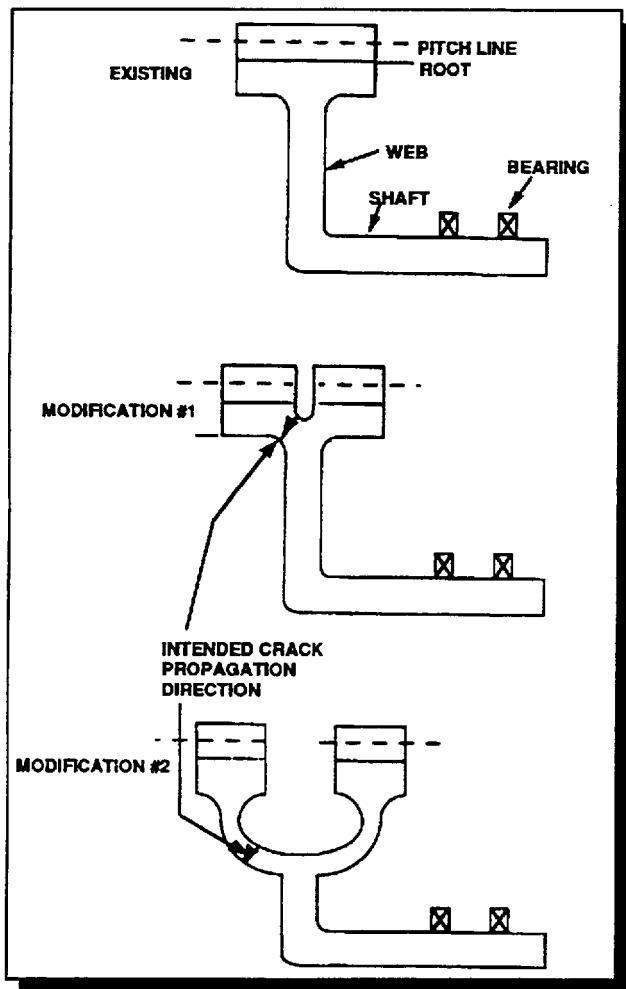


Figure 2-1 Schematic of Initial Gear Configurations

A groove is introduced through the center of the face width of the current/existing gear geometry to arrive at Modification 1 as shown in Figure 2-3; Note that all major gear, shaft, and rim dimensions of the modified configuration are the same as those of the existing gear except for the rim width which is increased to accommodate the groove width. Gear face width and all other gear parameters are however identical. The width and the depth of the groove (below the root of the tooth) can be altered to evaluate several geometries under this modifications. During initial planning, a constant groove width of .25 inch and four groove depths of .07, .14, .21 and .27 inch were considered for analytical evaluation. Accordingly, the existing gear (Gear 1) and the gear with .27 inch depth (Gear 5) were first analyzed. These analyses indicated that a deeper and wider groove may develop a needed stress pattern for cracks to propagate in a desired direction for failsafe operation. Therefore, it was decided to drop cases with smaller groove depths (Gears 2, 3 & 4) and instead analyze and evaluate two more cases (Gear 6 and Gear 7) with increased

groove depth and width. The resulting stress patterns will be considered in deciding additional cases under Modification 1, if any, needed for this study as well as the need/viability of Modification 2 shown in Figure 2-1.. Table 2-1 summarizes geometric dimensions used for the four gear configurations selected for the initial evaluation. Of these four configurations, Finite Element Models's (FEM) are already built and analyzed for Gear 1, 5 & 6. Model development and analysis for Gear 7 with wider gap is in progress and will be reported later.

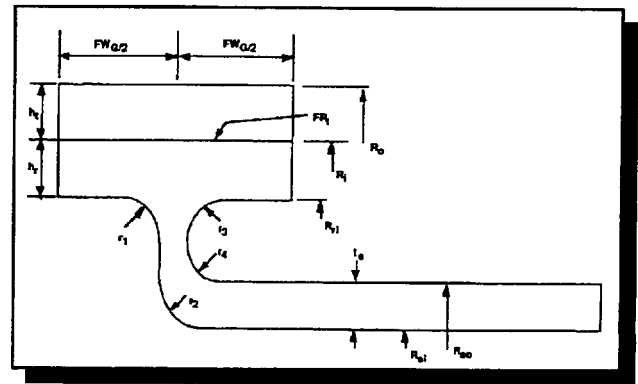


Figure 2-2 Typical Conventional Gear Tooth, Web, Blank Construction

In Table 2-1, the references to Gears 5, 6, and 7 are actually variations of the basic concept defined by the Modification #1 configuration shown in Figure 2-1 while Gear 1 refers to the basic, conventional design shown in Figure 2-1 as "Existing." The variations which are considered are groove depth, width, and fillet radius holding the basic tooth geometry constant.

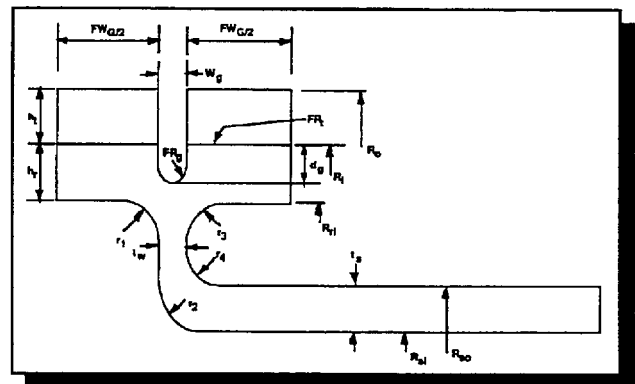


Figure 2-3 Gear Configuration With Groove For FE Analyses

The small loss in face width due to the groove can be made up with a small face width increase, however, as will be shown later in this discussion, changes in the basic tooth geometry design will yield lower overall stress levels such that even holding the same net face width, the gear tooth load capacity is not only not reduced but it can actually be improved. This is quite important in a weight critical design since it implies the ability to incorporate UltraSafe principles in the design while not increasing the weight of the gear. Of even greater interest and importance is the possibility that the weight of the gear may actually be decreased slightly or,

conversely, its load capacity may be increased with constant weight while still embracing the UltraSafe design concept. When evaluating the other aspects of the design of the gear teeth, including high profile contact ratio to provide added redundancy, as well as high face contact ratios to provide lower noise levels as well as higher inherent load capacity, it is clear that the UltraSafe concept involves the combination of many design factors, the most directly visible of which is the dual rim configuration, however, it is important to recognize that a number of other design changes are incorporated simultaneously to obtain the maximum benefit. Some of these additional parameters will be presented in more detail later in this paper.

Since the UltraSafe concept presupposes the fact that the basic gear is very reliable and has long life, the dual rim concept represented in Figure 2-1 provides a way in which localized unanticipated failures can be isolated so that they affect only about half of the gear's inherent capacity. Since the failures addressed in this concept are not the typical fatigue life limiting factors, it is reasonable to assume that the idiosyncratic factor which caused the failure would be randomly dispersed and thus unlikely to affect both load paths in the same gear. In fact, even a problem such as unexpected misalignment, which would manifest itself as pitting fatigue on one end of the tooth, would also be isolated by the dual rim concept. Even if the misalignment remains after one load path fractures, it will simply cause maldistribution on the remaining path and again propagate slowly.

In general, both mating gears in a set would be configured with dual rims as shown in Figure 2-3. This would afford the same measures of safety to both members of the set. In some cases, however, where one member of a particular gear mesh is not critical (for example an accessory gear driven by a main power gear in a rotorcraft transmission) the noncritical member would be made with the conventional single rim construction.

The dual rimmed gears would be otherwise designed as if they were conventional single rimmed gears. Lead and profile modifications would be made to the gear as if it had a full, continuous face width. In normal operation, the UltraSafe dual rimmed gear would operate exactly as if it were a conventional single rimmed gear and thus all other design parameters, especially lead and profile modifications must be addressed in the usual manner. When a failure occurs, of course, the load will be maldistributed on the remaining half face width but this condition is not anticipated as a normal operating condition thus nonuniform loading under emergency conditions is not a concern. In fact, it is desirable because it will increase the vibration level and may well result in some pitting, both of which are conditions which are easily detectable by existing diagnostic systems found on current rotorcraft systems. Increased indication of a problem while continuing to transmit full power is an integral part of the UltraSafe design philosophy.

When using dual rimmed gears in a planetary system, other options, still in development, are envisioned to allow an even greater level of UltraSafe behavior to be obtained. These include methods for keeping debris from a failure on one half of the dual rim from affecting the other half as well as other components within the planetary system (particularly the planet bearings).

These concepts will be presented, as they are more fully developed, in future publications.

Similarly, bevel gear systems may also make use of the same UltraSafe concepts, however, some modifications to the basic configuration of the bevel gear teeth will have to be made to accommodate the rim modifications shown for the parallel axis gears. Typically, small cutter technology bevel gears would be more amenable to the UltraSafe concept because of their tendency to have centrally located, pocketed contact patterns.

3.0 INITIAL EVALUATION GEAR CONFIGURATION

The scope of this investigation is presently limited to spur gears. As stated earlier, two design configurations were considered for the initial FEM stress evaluation for this phase of the study. The first includes a single geometry and depicts a conventional spur gear used in many current applications while remaining three geometries are included under the second configuration with a groove cut through the center of gear face width. The four design geometries developed for the present evaluation are:

Gear 1: Conventional existing spur gear (Figure 2-2 & Table 2-1)

Gear 5: Spur gear with .25" wide & .27" depth groove (Figure 2-3 & Table 2-1)

Gear 6: Spur gear with .25" wide & .37" depth groove (Figure 2-3 & Table 2-1)

Gear 7: Spur gear with .375" wide & .25" depth groove (Figure 2-3 & Table 2-1)

Of the above, finite element models and analyses have been completed for Gears 1, 5 and 6 and are presented in this report. Gear 7 work is still under progress and will be included either in a future publication.

The goal of the finite element analysis of these gears is to develop the optimized configuration for the groove depth, width, fillet radius, placement, etc. which will result in a strong gear rim to web to hub system with no substantial change in overall gear deflection while maintaining a stress condition in which, should a crack occur, it will progress through one half of the rim rather than down into the web and hub. This failure progression mode is the key to UltraSafe performance. In order to accomplish this, the stress levels in the tooth roots and the groove must be determined and their relative magnitudes must be controlled so any crack progression follows the rim and not the web path.

In addition to stress analyses, crack propagation models will also be created to evaluate the actual expected path which tooth root cracks would take under the applied loading. The results of both the stress and crack propagation analyses will be compared to experimental data obtained by single tooth bending fatigue testing of optimized UltraSafe test gears. These tests will be conducted in the next (experimental) phase of the program which will begin at the end of this year. An update report Sanitizing the initial testing is completed.

Table 2-1 Geometric Dimensions of Initial Gear Configurations

Geometric Dimension (in)	Gear Configurations Identification			
	Gear 1	Gear 5	Gear 6	Gear 7
Whole Tooth Height , ht	0.51	0.51	0.51	0.51
Rim Thickness , tr	0.52	0.52	0.52	0.52
Backup Ratio , mB = tr/ht	1.02	1.02	1.02	1.02
Gear Face Width , FWg	1.92	1.92	1.92	1.92
Width of Groove , wg	-	0.25	0.25	0.375
Depth of Groove , dg	-	0.27	0.37	0.27
Groove Radius , FRg	-	0.125	0.125	0.1875
Tooth Fillet Radius , FRt	0.093	0.093	0.093	0.093
Web Thickness , tw	0.25	0.25	0.25	0.25
Shaft Thickness , ts	0.4	0.4	0.4	0.4
Tooth Pitch Radius, Rp	2.94	2.94	2.94	2.94
Tooth Outer Radius, Ro	3.18	3.18	3.18	3.18
Tooth Root Radius, Ri	2.67	2.67	2.67	2.67
Rim Width, Rw	1.92	2.17	2.17	2.295
Rim Inner Radius, Rri	2.15	2.15	2.15	2.15
Shaft Outer Radius, Rso	1.4	1.4	1.4	1.4
Rim Fillet Radius , r1 to r3	0.38	0.38	0.38	0.38
Rim Fillet Radius , r4	0.25	0.25	0.25	0.25
No. Of Tooth = 28 , Speed = 3930 rpm Torque = 18042 in.lb , Tangential Load = 6121 lb , Radial Load = 2854 lb				

4.0 DEVELOPMENT OF FINITE ELEMENT MODELS

All initial models were created by using PATRAN Version 3 and were subsequently converted to Version 5.1 [1] pre-processor. While the method of building a finite element model (FEM) is a matter of individual preferences, source of available geometry, analysis objective, mesh size desired, model size limitations etc. partially lead to the specific sequential approach adopted for developing these models.

While a number of finite element codes can be used for model analysis, all of the work reported herein was conducted through the use of NASTRAN as the analysis tool. Similar results could be obtained with other packages but our gear work is largely standardized on NASTRAN.

Four out of 28 teeth on each gear were modeled with a fine mesh,

22 used a coarse mesh and the remaining two were used to accomplish transition from fine to coarse mesh tooth. This variation of mesh was required to ensure that the model size was within the acceptable limits imposed by the available computer hardware. The sequential steps involved in generating these models are:

- 1) Generate spur gear tooth coordinates by spline coordinate program (a proprietary Boeing program which calculates tooth coordinate data for the entire profile and fillet surface area).
- 2) Create a typical tooth geometry in PATRAN using these coordinates to generate the required curves, surfaces and solids as shown in Figures 4-1 and 4-2 to develop desired finite element mesh.



Documents

Document Sharing Hub

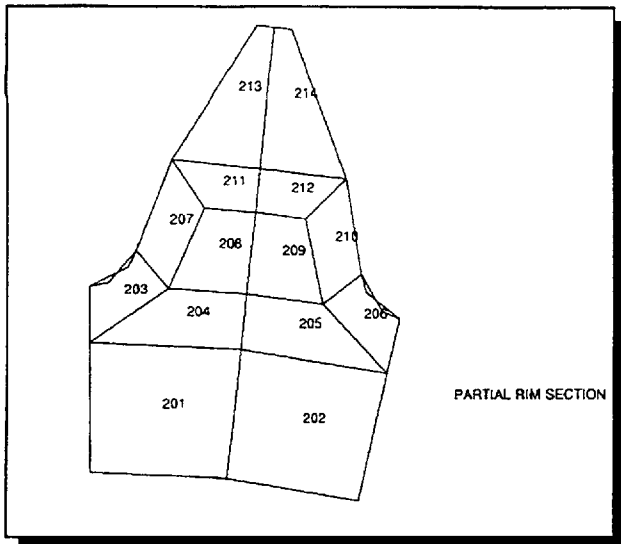


Figure 4-1 Gear 1, Tooth Profile 2-D Axial View

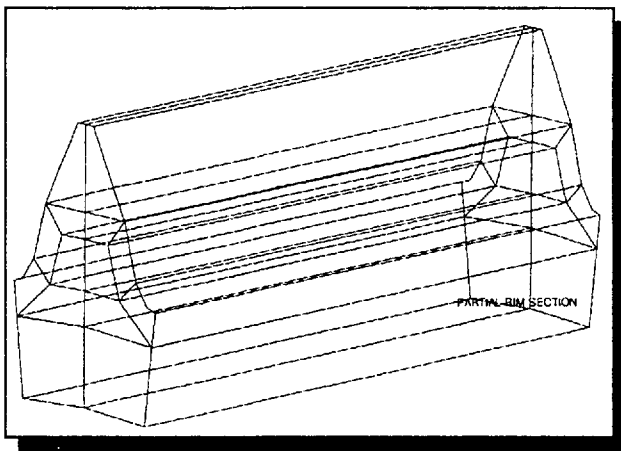


Figure 4-2 Gear 1 Tooth and Partial Rim Solid Geometry

- 3) Create backup structure geometry (Figure 4-3) consisting of gear rim, web and integrated shaft in PATRAN based on the dimensions of gear under consideration.

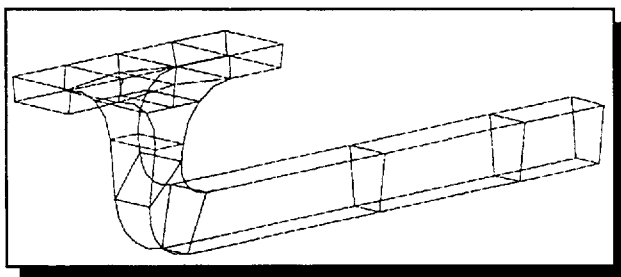


Figure 4-3 Gear 1 Backup 3-D Geometry for Tooth 1

- 4) Merge the tooth and backup structure geometry to create a combined geometry of one tooth segment for the first tooth (Tooth 1 or T1) as shown in Figure 4-4. Tooth 1 to

4 were used to define fine mesh size in all gears to monitor critical region stresses.

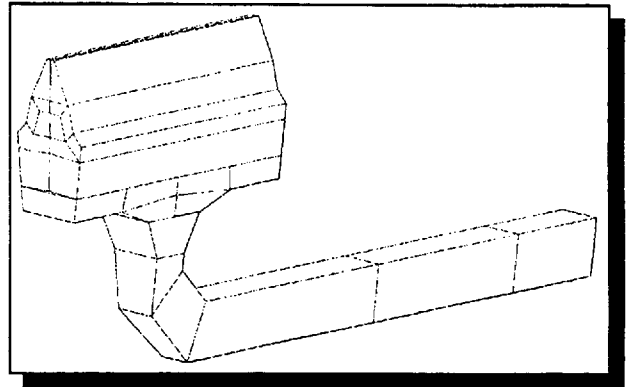


Figure 4-4 Gear 1 Tooth and Backup 3-D Solid Geometry

- 5) Rotate Tooth 1 geometry by 4 tooth angles to create geometry for tooth 5 (Figure 4-5) to develop finite element mesh for a transition tooth.

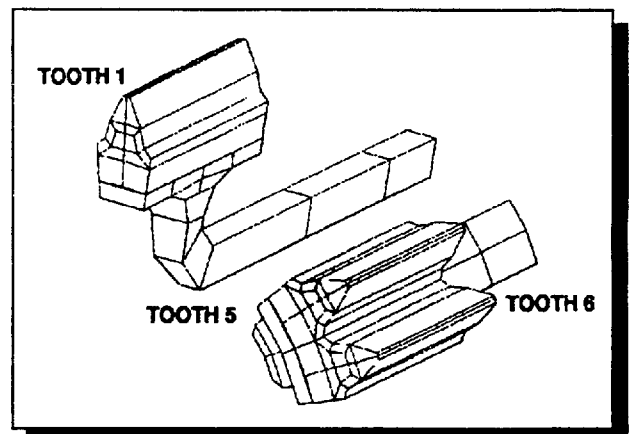


Figure 4-5 Gear 1 Full Geometry for FEM Generation

- 6) Rotate Tooth 1 geometry by 5 tooth angles to create geometry for tooth 6 (Figure 4-5) to be used for generating finite element model of a coarse mesh tooth.
- 7) Specify desired mesh seeds (number of elements) on all curves of tooth 1, 5 and 6 geometry to generate desired size elements for fine, transition and coarse mesh.
- 8) Create finite element model with fine mesh for Tooth 1 as shown in Figure 4-6.

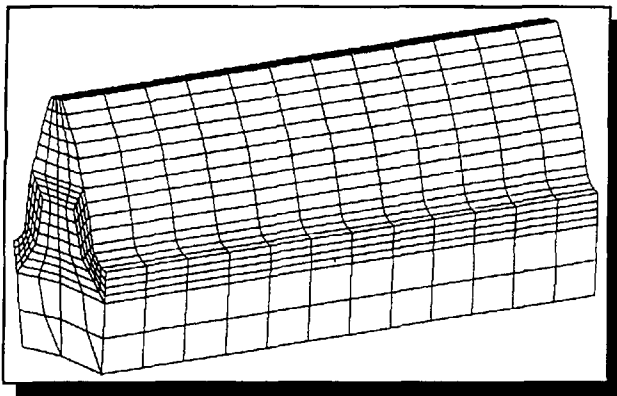


Figure 4-6 Gear 1 Tooth and Partial Rim FEM

- 9) Create finite elements for backup structure of Tooth 1 (Figure 4-7).

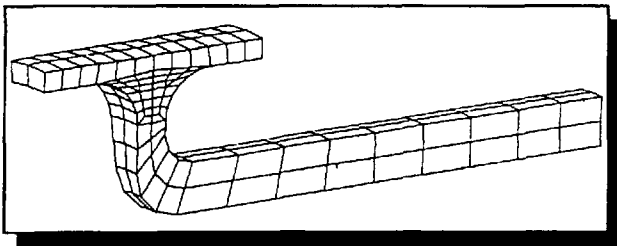


Figure 4-7 Gear 1 FEM for Tooth 1 Backup

- 10) Merge finite elements generated in steps 8 & 9 to create a complete one tooth segment for fine mesh Tooth 1 as shown in Figure 4-8.

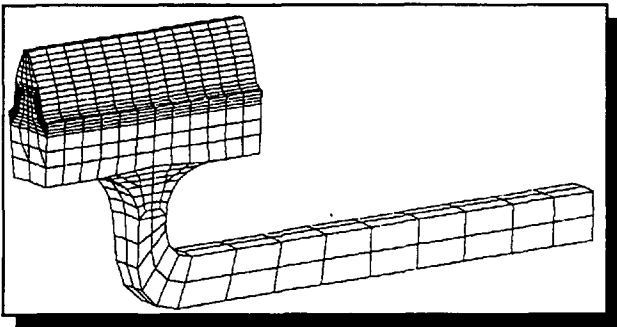


Figure 4-8 Gear 1 Fine Mesh FEM of First Tooth (T1)

- 11) Rotate fine mesh tooth segment model of step 10 to generate finite element mesh for Tooth 2, 3 & 4.
12) Create finite element mesh for Tooth 5 from its geometry to generate one tooth segment model for transition tooth displayed in Figure 4-9.

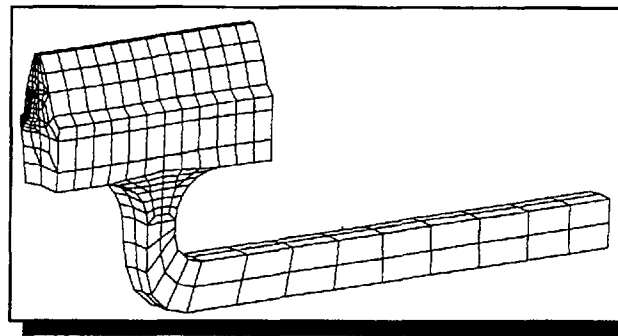


Figure 4-9 Gear 1 Transition Mesh FEM of Tooth 5 (T5)

- 13) Create one tooth segment finite element model with coarse mesh for Tooth 6 as shown in Figure 4-10 from its geometry.

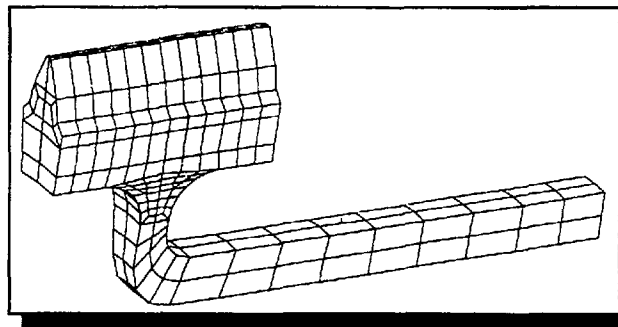


Figure 4-10 Gear 1 Coarse Mesh FEM of Tooth 6 (T6)

- 14) Rotate Tooth 6 model by one tooth angle 21 times to create finite element mesh for Tooth 7 through 27.
15) Generate finite element model for transition Tooth 28 from Tooth 5 model by using mirror option about appropriate plane to complete full 360 degree model of the gear under consideration. Note that equivalencing and node renumbering operations are required to delete duplicate nodes and compact node numbers. These operations can be performed either after each step in which finite elements are generated or after the entire model is created.
16) Create rigid elements extending from the shaft axis of rotation to its inside surface to simulate the two bearings at their appropriate axial locations (Figure 4-11).
17) Add applicable boundary conditions and loads and write out the analysis deck to complete the finite element model development. An isometric view illustrating the transition from fine to coarse mesh for Gear 1 is shown in Figure 4-12 while the full model is depicted in Figure 4-13.