NASA
SPACE VEHICLE
DESIGN CRITERIA
(CHEMICAL PROPULSION)

NASA SP-8100

LIQUID ROCKET ENGINE TURBOPUMP GEARS

MARCH 1974

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

FOREWORD

NASA experience has indicated a need for uniform criteria for the design of space vehicles. Accordingly, criteria are being developed in the following areas of technolgy:

Environment
Structures
Guidance and Control
Chemical Propulsion

Individual components of this work will be issued as separate monographs as soon as they are completed. This document, part of the series on Chemical Propulsion, is one such monograph. A list of all monographs issued prior to this one can be found on the final pages of this document.

These monographs are to be regarded as guides to design and not as NASA requirements, except as may be specified in formal project specifications. It is expected, however, that these documents, revised as experience may indicate to be desirable, eventually will provide uniform design practices for NASA space vehicles.

This monograph, "Liquid Rocket Engine Turbopump Gears," was prepared under the direction of Howard W. Douglass, Chief, Design Criteria Office, Lewis Research Center; project management was by Harold Schmidt and Lionel Levinson. This monograph was written by Martin A. Hartman and Myles F. Butner, Rocketdyne Division, Rockwell International Corporation, and was edited by Russell B. Keller, Jr. of Lewis. To assure technical accuracy of this document, scientists and engineers throughout the technical community participated in interviews, consultations, and critical review of the text. In particular, George H. Costomiris of Pratt & Whitney Aircraft Division, United Aircraft Corporation; Wayne H. Glover of The Buehler Corporation; William W. Heath of Aerojet Liquid Rocket Company; and Delmar W. Drier of the Lewis Research Center reviewed the monograph in detail.

Comments concerning the technical content of this monograph will be welcomed by the National Aeronautics and Space Administration, Lewis Research Center (Design Criteria Office), Cleveland, Ohio 44135.

March 1974

For sale by the National Technical Information Service
Springfield, Virginia 22151
Price – $4.50

GUIDE TO THE USE OF THIS MONOGRAPH

The purpose of this monograph is to organize and present, for effective use in design, the significant experience and knowledge accumulated in development and operational programs to date. It reviews and assesses current design practices, and from them establishes firm guidance for achieving greater consistency in design, increased reliability in the end product, and greater efficiency in the design effort. The monograph is organized into two major sections that are preceded by a brief introduction and complemented by a set of references.

The State of the Art, section 2, reviews and discusses the total design problem, and identifies which design elements are involved in successful design. It describes succinctly the current technology pertaining to these elements. When detailed information is required, the best available references are cited. This section serves as a survey of the subject that provides background material and prepares a proper technological base for the *Design Criteria* and Recommended Practices.

The *Design Criteria*, shown in italics in section 3, state clearly and briefly what rule, guide, limitation, or standard must be imposed on each essential design element to assure successful design. The *Design Criteria* can serve effectively as a checklist of rules for the project manager to use in guiding a design or in assessing its adequacy.

The Recommended Practices, also in section 3, state how to satisfy each of the criteria. Whenever possible, the best procedure is described; when this cannot be done concisely, appropriate references are provided. The Recommended Practices, in conjunction with the *Design Criteria,* provide positive guidance to the practicing designer on how to achieve successful design.

Both sections have been organized into decimally numbered subsections so that the subjects within similarly numbered subsections correspond from section to section. The format for the Contents displays this continuity of subject in such a way that a particular aspect of design can be followed through both sections as a discrete subject.

The design criteria monograph is not intended to be a design handbook, a set of specifications, or a design manual. It is a summary and a systematic ordering of the large and loosely organized body of existing successful design techniques and practices. Its value and its merit should be judged on how effectively it makes that material available to and useful to the designer.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

LIQUID ROCKET ENGINE TURBOPUMP GEARS

1. INTRODUCTION

Gear drives for rocket engine turbopumps are used to achieve maximum efficiency in low- to medium-power drives (100 to 4500 hp)[1] when

(1) propellant pumps handling rather dense propellants (specific gravity 0.7 or greater) are to be driven by a small, high-speed turbine; or
(2) a single turbine is to drive turbopumps handling propellants with greatly differing densities that require different pump speeds.

Rocket engines, and therefore their components including gear drives for turbopumps, have a rather unique life requirement: infallible operation, with a rapid start from standstill, for one duty cycle of up to 500 seconds duration. Acceptance testing extends the required life of turbopump gears to approximately one hour. Performance requirements for flight naturally emphasize low weight. The combination of requirements imposed on a turbopump gear can be summed up as follows:

- High reliability
- High load capacity
- High speed capability
- Short life
- Light weight

The unusual operating conditions for turbopump gears and the severe requirements imposed on them required advances in gear technology. Turbopump power gears were brought to acceptable levels of usefulness and reliability through refinements in interdependent areas of design, materials, processing, and quality control combined with extensive development testing that explored problem areas and evaluated potential solutions. This monograph sets forth the technology developed to date, identifies the problem areas, and summarizes proven procedures for successful gear design.

The development of gears for turbopumps required solutions in the following major problem areas:

[1]Factors for converting U.S. customary units to the International System of Units (SI units) are given in Appendix A.

(1) Tooth root fractures – eliminated by a combination of modification of gear dimensions, improvement in material cleanliness, rigid control of manufacturing process, and shot peening.
(2) Compressive contact failures – eliminated by improvement of gear geometry such as addendum proportioning, profile modification, and lead modification; improved material cleanliness; and rigid control of dimensions.
(3) Scoring in lubricated gears – eliminated by control of surface texture, development of better lubricants, and advances in the design of lubricant delivery systems.
(4) Vibrational effects – eliminated by modifying the magnitude or the frequency of exciting forces or by altering the response of the gear elements. Design variations for this purpose include changing the number of teeth, modifying the tooth profile, altering rim and web thicknesses, adding dampers, or reducing the allowable imbalance limits of the rotating members.

These problems and solutions are not entirely independent, and some mutual benefits accrue from each improvement; the overall success of the gear depends on minute attention to details in all areas. Graded materials, rigidly controlled in quality; closely controlled fabrication techniques; increased precision in equipment and techniques for monitoring close tolerances – all are additional factors in successful design and production of turbopump gears.

To obtain an optimum gear design, it is essential that the gear designer take an active part in the turbopump preliminary design by aiding in basic decisions that influence the gearing. He should take the following steps:

(1) Reduce the pitchline velocity to the lowest realistic value.
(2) Determine the lightest weight gear design that can safely be used.
(3) Minimize the number of gears.
(4) Guide the turbopump layout designer in selecting the proper gear type and size.
(5) Provide for balancing as required.
(6) Ensure that provisions are made for the use of the proper lubricant and lubricant delivery system.
(7) Minimize all external forces reacting on the gear casing.
(8) Ensure practical and easy methods of gear, bearing, shaft, and gear-case assembly.
(9) Ensure a rigid mounting for the gears, and avoid design practices that cause gear misalignment.
(10) Ensure by analysis or test that gear system resonances do not coincide with meshing frequencies or operating speed.

After the turbopump layout has been adopted, the gear designer must supervise the individual gear design and detailing, reviewing all dimensions. In addition, he should be satisified with all fits, finishes, and clamping methods used on components that will affect

the gears. The designer should continue to monitor gear design and development through final production release. Operational problems should receive his prompt analysis for possible design change requirements.

This monograph treats gear design and fabrication in the sequence encountered during the design process: selection of overall arrangement, selection of gear type, preliminary sizing, lubrication system design, detail tooth design, selection of gear materials, and finally gear fabrication and testing as it affects the design. There is, of course, a good deal of cross-feed among the phases of design and, accordingly, the monograph frequently cross-references and dovetails related material. The monograph is oriented toward the use of involute spur gears, although reference material for helical gears is cited.

2. STATE OF THE ART

Satisfaction of the severe requirements imposed on turbopump gears depends on the following factors:

(1) A design that is technically adequate
(2) Gear materials and manufacturing processes that exploit the potential of the design
(3) Quality-control procedures that ensure consistent conformance to the proper dimensional tolerances, and fabrication processes that can be monitored by nondestructive tests.

Favorable design features include large pressure angle (22.5° to 27.5°), full-depth tooth forms with long addendum pinions, generous root fillets, and profile modifications. Tight tolerances increase the smoothness of operation and reduce peak stresses. Special measuring equipment with increased sensitivity and accuracy plus standardized procedures ensure that specified tolerances actually are obtained.

Procurement of materials with adequate and uniform mechanical properties is ensured by using mill-lot-controlled, vacuum-melted steels and by specifying representative tensile tests and metallurgical analyses.

Specification of gear manufacturing method frequently is necessary for successful design. Gears are machined from close-tolerance forgings with specified grain flow. Gear-tooth contacting surfaces are ground, but root fillets specifically are not ground. Shot peening of gear-tooth roots and gear webs often dramatically increases gear life.

Testing in simulated service with dynamometer or back-to-back loading is an effective design aid. The back-to-back configuration in particular enables economical and rapid accumulation of a large amount of test data. Such tests have been used extensively in evaluating design, processing, and lubrication features prior to incorporation into the final design. Even after the success of a design has been established by test, process uniformity must be maintained to avoid seemingly unexplainable failures. For example, a sudden drop in the fatigue life of one supplier's gears was traced to variations in a nonuniform shot-peening process.

Lubricants and lubricant delivery systems have been the subject of development testing that led to improved gear load capacity with conventional and unconventional lubricants. Testing also is used to monitor lubricant quality and effectiveness.

A typical sequence of actions in the design and manufacture of aerospace gears is as follows:

After engine performance considerations lead to the selection of a geared drive transmission, the general features of the gear system are determined: basic gear type, the size and number of gears, and the speeds and rotational directions of the shafts. The gear design then progresses through preliminary determination of operating stresses to detail selection of pitch, number of teeth, profiles, required modifications, material selection, and creation of manufacturing drawings and specifications of materials and processes. Acceptance tests and performance tests used as design aids conclude the gear development.

2.1 GEAR SYSTEM

Trial layouts of the turbopump system including pumps, turbines, and ducts guide the attempt to achieve the most compact physical arrangement that provides adequate spacing of these components. The number of gears and the diameters thereof then are selected to achieve the desired speed ratio and rotation direction within constraints imposed by

- Center distance and volume available
- Gear-tooth load capacity
- Gear speed capacity
- Practical maximum gear size
- Minimum number of pinion teeth to avoid undercutting.

Lubrication of the gears is a major factor in achieving success in high-power high-speed gear systems, and therefore considerable effort is expended in the selection of lubricants and in the design of the lubricant delivery system.

Particular care is exercised during design to ensure that the gear arrangement obtains proper rotational direction of pumps, turbines, and accessories. Direction of rotation often is dictated by engine requirements and is noted specifically on assembly drawings.

The features of main-power-transmission gear systems for existing major turbopumps are summarized in table I. Table II lists the major gear detail design values of power, speed, lubrication, and stresses. Schematic views of the gear trains used in major turbopump systems appear in figure 1.

Nomenclature used in the tables and figures and throughout the text is based on American Gear Manufacturers Association (AGMA) standards (ref. 1) except where noted.[1]

[1]Symbols; materials; specifications; vehicles, pumps, and engines; and abbreviations used herein are identified in Appendix B.

Table 1. – Gear System Data for Major Operational Geared Turbopumps

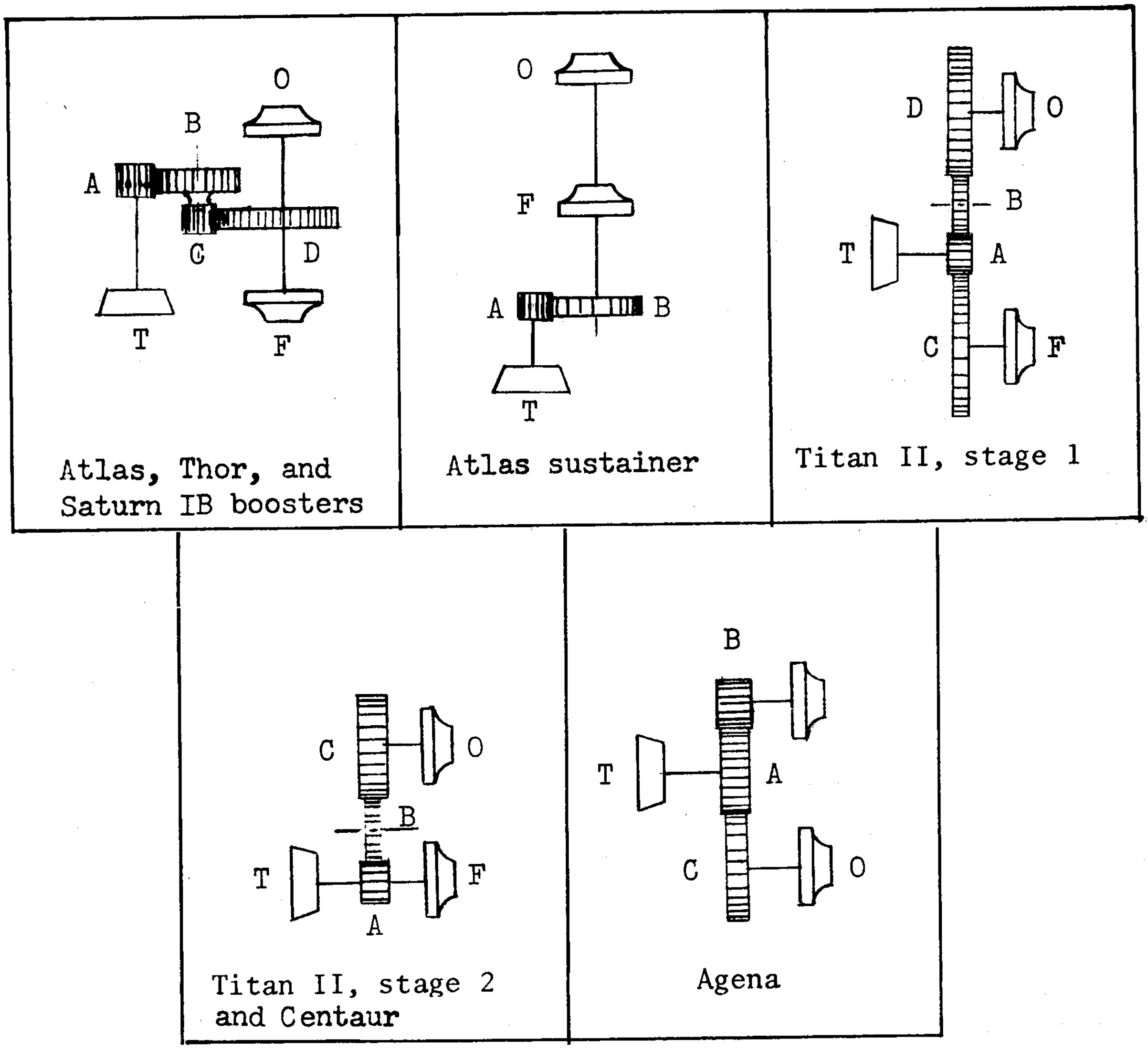
Vehicle (system)	Input speed, rpm	Speed reduction ratio, Input/pump-shaft		Power transmitted, hp	Lubrication system				Gear material	Gearbox weight, lbm	Life, sec	
		Oxidizer	Fuel		Lubricant	Lubricant flow, gpm	Lubricant supply pressure, psig	Gearcase internal pressure, psig			Specified	Attained
Atlas, Thor, Saturn IB boosters (Mark 3)	32 730	4.885	4.885	4920	RP-1 and extreme-pressure additive[a]	5.5	620	4.0	AMS 6265	150	3750	> 65 000
Atlas sustainer (Mark 4)	38 085	3.750	3.750	1650	MIL-L-6086 or MIL-L-25336	1.2	625	1.4	AMS 6265	approx. 40 (gearbox not unitized)	6000	> 20 000
Titan II, 1st stage (LR-87-AJ-5)	23 000	2.87	2.62	4466	MIL-L-7808	3.4	30	15.0	AISI 4620 or 9310	228	2500	> 6 000
Titan II, 2nd stage (LR-91-AJ-5)	22 800	2.82	NA	924	MIL-L-7808	1.5	30	15.0	AISI 4620	114	2500	> 6 000
Centaur (RL10)	30 000	2.50	1.00	80	Dry film[b] and hydrogen	0.4	40	3.5	AISI 9310	Gear case is part of engine case	420	20 000
Agena (LR81-BA-11)	24 800	1.721	0.977	357	MIL-L-7808	1500 cm^3, closed system	Not supplied under pressure	Does not apply	AMS 6250	NA	NA	NA

[a] 98% RP-1 plus 2% Oronite 262 (zinc dialkyl dithiophosphate).
[b] 25% powder, 75% AMS 3132 varnish, AMS 3170 thinner as required. The powder consists of 10 parts MoS_2 and 1 part graphite by weight.

NA = not available.

Table II. – Gear Design Data for Major Operational Geared Turbopumps

Vehicle (system)	Gear (fig. 1)	Power, hp	Speed, rpm	Torque, in.-lbf	Pitchline velocity, ft/min	Pitch diameter, in.	Diametral pitch, in.$^{-1}$	Number of teeth	Tooth face width, in.	Pressure angle, deg	Unit load, psi	Surface compressive stress, psi
Atlas, Thor, Saturn IB boosters (Mark 3)	A	4 920	32 730	9 474	25 727	3.00	11	33	1.44	25	48 247	263 067
	B	4 846	15 430	19 793	25 727	6.36	11	70	1.32	25	52 633	263 067
	C	4 846	15 430	19 793	16 676	4.125	8	33	1.46	25	52 584	263 067
	D	4 777	6 700	44 935	16 676	9.50	8	76	1.64	25	46 146	263 067
Atlas sustainer (Mark 4)	A	1 650	38 085	2 730	20 000	2.00	12	24	1.12	25	29 200	–
	B	1 650	10 157	10 230	20 000	7.50	12	90	1.00	25	32 800	–
Titan II, 1st stage (LR-87-AJ-5)	A	4 466	23 000	12 200	16 600	2.75	11.25	31	2.125	20	26 100	185 000
	B	2 320	11 000	13 300	16 600	5.78	11.25	65	1.99	20	25 600	191 000
	C	2 146	8 010	16 900	16 600	7.91	11.25	89	2.00	20	25 500	175 000
	D	2 320	8 800	16 600	16 600	7.20	11.25	81	2.125	20	26 108	142 000
Titan II, 2nd stage (LR-87-AJ-5)	A	924	22 800	2 550	16 400	2.75	12	33	1.00	20	22 300	173 000
	B	924	10 900	5 340	16 400	5.75	12	69	0.875	20	25 500	184 000
	C	924	8 100	7 190	16 400	7.75	12	93	1.00	20	22 300	129 000
Centaur (RL10)	A	80	30 000	167	15 720	2.01	13.96	28	1.20	22.5	2 140	56 400
	B	80	12 200	414	15 720	4.94	13.96	69	1.10	22.5	3 340	56 400
	C	80	12 200	420	15 720	5.01	13.96	70	0.70	22.5	3 350	53 600
Agena (LR81-BA-11)	A	357	24 800	907	17 400	2.6875	16	43	0.500	20	16 200	121 000
	B	157	25 300	390	17 400	2.625	16	42	0.375	20	12 700	120 000
	C	200	14 410	874	17 400	4.625	16	74	0.375	20	16 200	126 000



A, B, C, D: gear identity (used in table II)

O: oxidizer pump

F: fuel pump

T: turbine

Figure 1. – Schematics of gear arrangements in major operational turbopumps.

2.1.1 Speed Ratio

Gear trains in rocket engine turbopump main-power drives permit matching normally high optimum turbine speeds with normally much lower optimum pump speeds. Table I lists main-power-train speed ratios for major gear driven turbopumps. Accessories such as lubrication pumps, hydraulic pumps, and electric generators are driven at the desired speed through smaller accessory-drive gear trains.

The following limitations are imposed on speed ratio:

(1) There must be a whole number of teeth on each gear; hunting-tooth action usually is desirable and is obtained by choosing numbers of teeth with no common factors in a mating set of gears.
(2) Multiple reductions are required to obtain large overall ratios. Losses in tooth strength and nonoptimum tooth action will result if too large a reduction is attempted in a single mesh. In general, a maximum ratio of 5 per mesh is permitted.
(3) The number of teeth in the pinion must be maintained above the minimum number required to avoid weakening the teeth. Minimums for varying gear sizes are given in reference 2 (ch. 5).

Although the fixed speed ratio of a gear train often is an asset, the fixed ratio limits flexibility in engine performance if a change in relative output is required for a performance change or for throttling. For most rocket engines that use fixed mixture ratios, fine adjustments in flow balance are made by trimming pump impellers or by changing pump discharge orifices.

2.1.2 Speed Capability

Gears have been designed and operated at pitchline velocities up to 50 000 ft/min; operation at higher speeds is so difficult as to be impractical. Table III lists broadly classified speed ranges with corresponding risks, requirements, and potential problem areas. Pitchline velocities up to 25 727 ft/min have been utilized in operational turbopump gears (table II); R&D tests indicate that these gears can operate satisfactorily at speeds up to 27 200 ft/min.

Table III. – Characteristics of Speed Ranges for Gears*

Pitchline velocity, ft/min	General classification of speed range	Characteristics and hazards
4 000 to 15 000	Normal	These speeds are representative of those attained with most high-speed gears. Centrifugal stresses cause no problems. Many manufacturers are capable of building the units. Dynamic balancing is not critical. Moderate gear tooth accuracy is required.
15 000 to 25 000	High	First reductions of turbine-driven pumps are in this range. Centrifugal stresses can cause problems. Gear tooth accuracy is critical. Dynamic balance is important. Relatively few manufacturers are qualified and experienced.
25 000 to 30 000	Very high	These speeds generally are found only in rocket engine and aircraft gas turbine test rigs or in large industrial gas turbine drives. Centrifugal stress problems are critical. Failures are potentially dangerous to human life because of probable casing rupture. Solid rotor designs usually are required for the gear. Gear tooth accuracy is critical. Lubrication is critical because of windage problems and possible excessive temperature rise on working tooth surfaces. Rotors must be balanced to fine limits. Few gear manufacturers are qualified to fabricate gears for this speed range.
30 000 to 45 000	Ultra high	This is the "frontier" area of extremely-high-speed gears. Failures are highly dangerous. Even the best solid rotor designs may rupture because of small metallurgical defects. High-speed balancing techniques are required. The best gear manufacturers in the United States have had some successes and some real problems in the few special aerospace gears that have been fabricated. No turbopump applications in this speed range are in use.

*Adapted from a report presented by D. W. Dudley at ASME Annual Meeting, 1966.

2.1.3 Gear Type

Turbopump main-power trains incorporate coplanar involute spur gears exclusive of other types. For turbopump gearing, which involves high loads and wide temperature ranges, one of the most advantageous features of involute gearing is its tolerance of small variations in center distance. Spur gears were initially applied to turbopump power trains because of their high efficiency and simplicity; helical gears with their higher load capacity and smoother tooth action might have been used if the present high power and speed levels shown in table II had been the initial design requirements. However, most turbopump gears were originally designed for more modest loads and speeds; as uprating took place in moderate increments of power, it was economically more desirable to adapt and improve existing spur gears rather than to initiate new helical designs. Further uprating beyond the level shown for the Mark 3 gear train (table II) will make the use of helical gears attractive despite the cost of a new design and higher cost of manufacture.

The choice between helical and spur tooth configurations normally is based on speed, helical gears being used more as pitchline velocity increases. Figure 2 presents the relative usage of

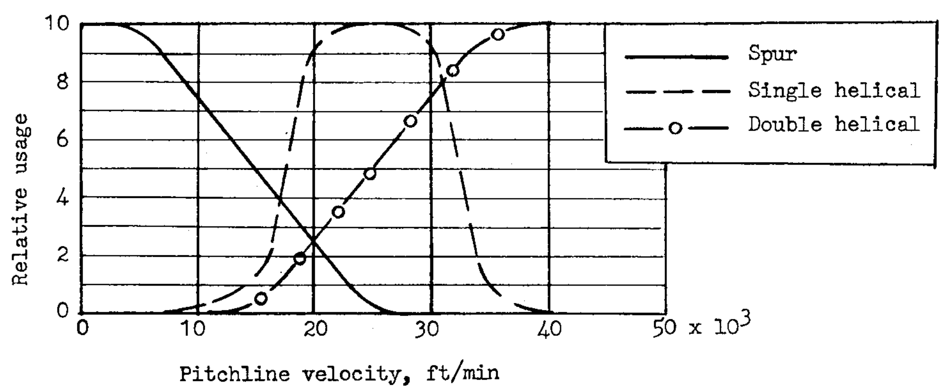


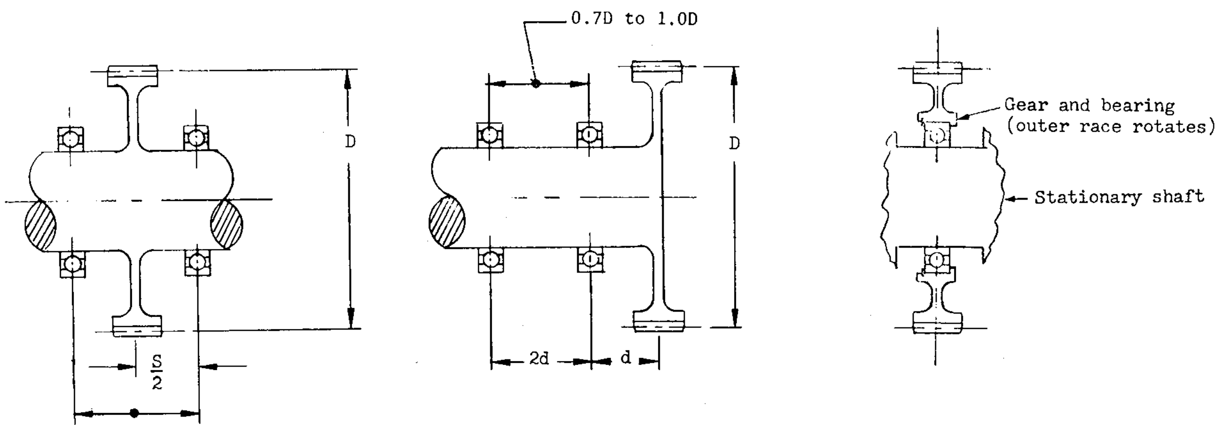
Figure 2. – Relative usage of gear types as a function of pitchline velocity.

the two types in power gearing. Smoothness of operation is a definite advantage of helical gearing; loads can be transferred more gradually to successive teeth because of tooth overlap. For example, in a non-turbopump power-gear application, vibrational amplitude was reduced by a factor of 6 when helical gears were substituted for spur gears.

Non-coplanar gearing such as bevel, worm, and hypoid gears have not been used in turbopump main power trains because of their lower efficiency (increased heat generation). Bevel gears have been used for drive connections to power trains for hand-rotation torque tests and for accessory drive gears in turbopump systems.

2.1.4 Gear Mounting

Figure 3 presents three methods for mounting gears.



(a) Straddle mount

(b) Overhung mount

(c) Mount for idler gear with single bearing

Figure 3. – Gear mounting methods.

Straddle mounting (fig. 3(a)) is used whenever possible in order to minimize deflections under load that cause changes in center distances and misalignment. The proportions shown are general guidelines; the actual dimensions are the result of attempts to maximize mounting rigidity and are selected after detailed calculations of stress and deflection for the individual application.

Overhung mounting (fig. 3(b)) sometimes is used when space saving is a major goal. An example is the turbine shaft of the Mark 4 (Atlas sustainer) turbopump. As with straddle mounting, maximum rigidity is sought. General guidelines for proportions of an overhung mounting are shown in the figure, but actual dimensions result from detailed analyses of the deflections under load.

Mounting of idler gears on a single ball bearing (fig. 3(c)) has been used advantageously on the RL10 gears. This configuration tolerates misalignment by rocking the ball bearing so the idler gear teeth line up with the driving gear teeth. A potential problem area is bearing capacity, since the tangential forces on the idler gear teeth are additive. In addition, the life of the bearing is shortened by rotation of the outer race.

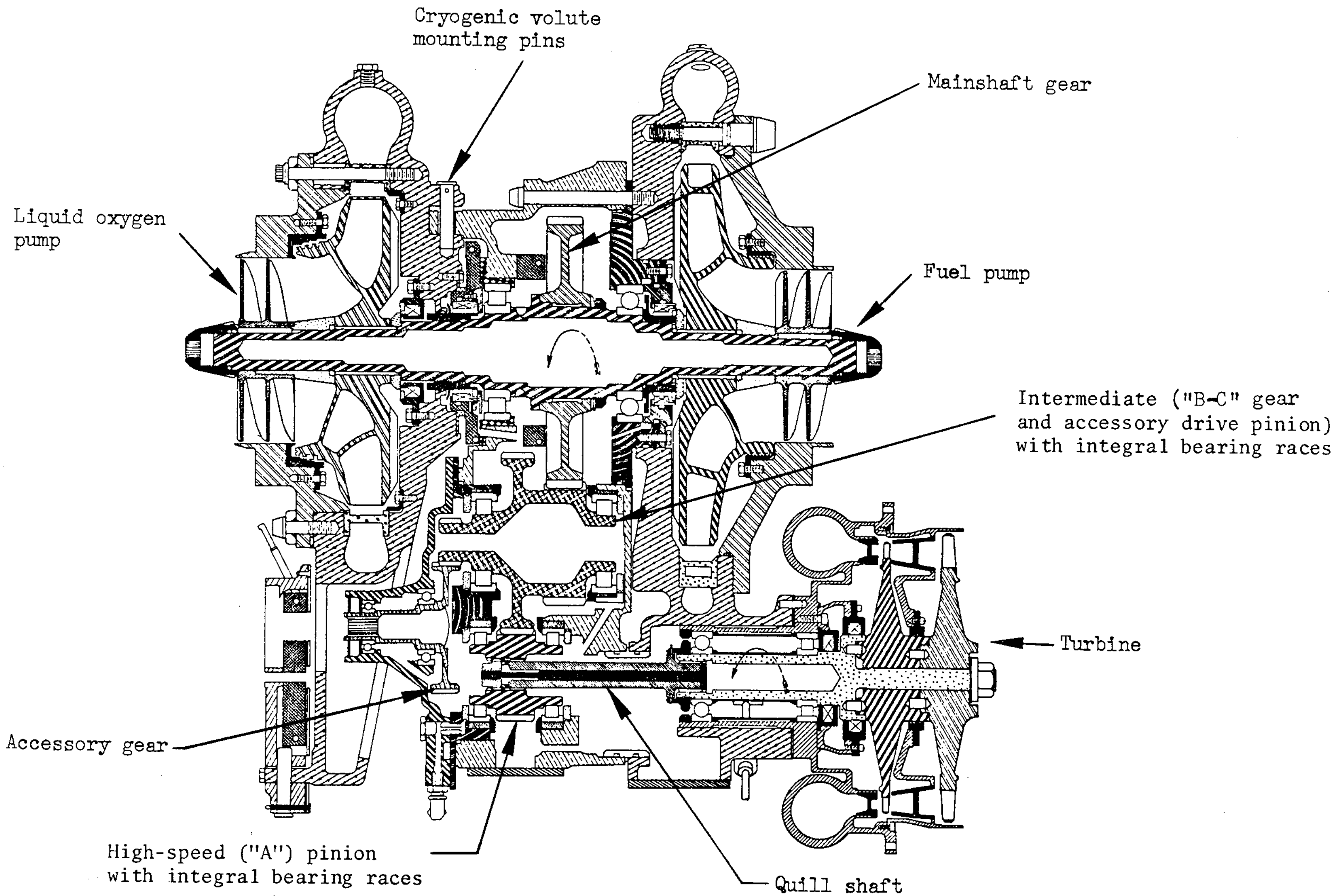


Figure 4. – Schematic of Mark 3 turbopump (Atlas, Thor, and Saturn IB boosters).

2.1.5 Gear Attachment

Secure attachment of the shaft is necessary for gears under high loads, because any looseness will create additional deflections. Relative motion will result in fretting and damage to the mounting surfaces. In extremely highly loaded gears, a satisfactory solution to fretting has been to make the gear integral with its shaft. For example, figure 4 shows the one-piece gears in the Mark 3 turbopump gear trains, the "A" pinion and the "B-C" accessory-pinion cluster gears.

If the gear must be removable from the shaft, the mounting method shown in figure 5 is

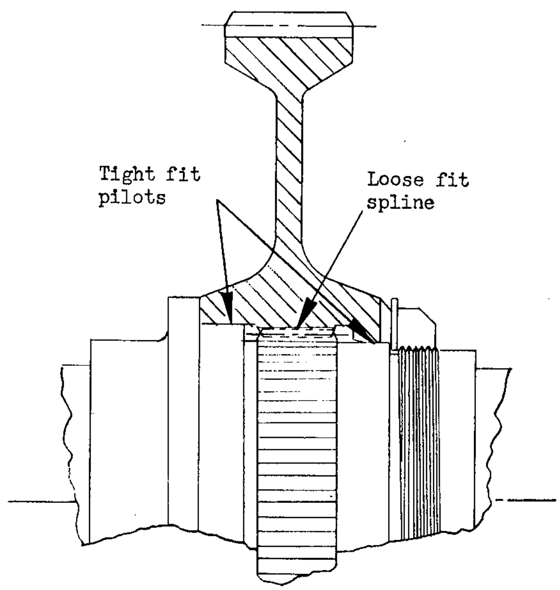


Figure 5. – Method for mounting a detachable gear.

used. Radial position is maintained by the tight-fitting pilots; driving torque is transmitted through the spline. The clamping-nut torque is made high enough to prevent relative motion between stacked components due to thrust loads, bending, or thermal expansion. Severe fretting of the faces of the Mark 4 (Atlas sustainer) turbopump pinion was eliminated by increasing the clamping-nut torque.

2.1.6 Backlash

Backlash is the clearance between the space width of a given gear and the tooth thickness of meshing gear. Minimum but sufficient backlash (ref. 1) is provided in the design to avoid contact on the nondriving side of the gear tooth under any combination of tolerance stackup, differential thermal growth, and lubricant film buildup. Test experience may indicate the need for a change in backlash. Absence of backlash will lead to actual gear interference, high forces, and heat generation leading to lockup and gear failure.

2.1.7 Load Capacity

Gear tooth mechanical strength forms the basis of gear load capacity. However, achieving maximum capacity also requires optimizing lubrication (sec. 2.1.8), alignment (sec. 2.1.9), tooth details (sec. 2.2), materials (sec. 2.3), and fabrication (sec. 2.4). Further, in determining the required gear load capacity, the designer recognizes that nearly every turbopump has been uprated; therefore, provisions for uprating through future design changes are considered in the original design. An example of evolutionary development is the Mark 3 gear train used in the engines for the Atlas, Thor, and Saturn IB boosters. Originally designed for 1800 horsepower, the gear train was developed over a period of approximately 10 years to handle nearly 5000 horsepower (table II). The incremental nature of the power increase required small improvements in materials, gear detail design, lubrication, and fabrication techniques; these changes were then consolidated into design requirements after experimental verification.

Tooth mechanical strength is composed of root bending strength, compressive contact strength, and the tooth's resistance to chipping of the edges. The allowable bending and compressive stress levels depend on the required life (number of stress cycles), the desired reliability, the gear material quality level, and manufacturing tolerances. Chipping resistance depends on the material, its heat treatment, and avoidance of stress concentrations at the edges.

2.1.7.1 TOOTH ROOT BENDING STRENGTH

Since a gear tooth is similar to a cantilever beam, physical size is an index of strength; a larger tooth, designated by a numerically smaller diametral pitch, is stronger. This relationship is reflected in the gear tooth breakage index, "unit load," which is a gauge of gear tooth strength used in preliminary design calculations. Unit load for spur gears is defined as

$$U_L = \frac{W_t P_d}{F} \qquad (1)$$

where

U_L = tooth breakage index, or unit load, psi

W_t = total tangential tooth load, lbf

F = effective face width, in.

P_d = diametral pitch of gear tooth = $\frac{\text{number of teeth}}{\text{pitch diameter, in.}}$

Turbopump gears often are designed for unit loads higher than those used for aircraft gears. The resulting use of smaller gears and the consequent weight saving is made possible by the short required life, special design features, and more stringent quality control imposed on materials and manufacturing for turbopump gears. Unit loads for existing turbopump power gears are listed in table II. Geometrically similar gears made of AGMA material quality grade 2 (aircraft quality) generally are limited to a unit load of 25 000. Accessory gears made of AGMA material quality grade 1 are limited to a maximum unit load of 12 500. Reference 3 discusses aircraft material quality grades. Dimensional tolerances are listed as AGMA quality classes in references 2 (pp. 9-16 through 9-19) and 4.

Refinement of unit load to a more accurate estimate of root bending stresses is obtained by multiplying the unit load by factors compensating for tooth geometry and by derating factors dependent on the gear quality, severity of service, and alignment.

The AGMA methods outlined in reference 3 assume the gear tooth to be a shaped cantilevered plate. Tests have shown this approach to be the most accurate available (refs. 5 [spur gears], 6 [helical gears], and 7 [spiral bevel gears]).

Values for root bending stress are calculated by methods presented in reference 8. A large-scale layout of the gear tooth provides the basis for selecting the stress-modifying factors that account for tooth form (Y factor) and geometry (J factor). Further modifying factors are used to account for size (K_s), dynamic loads (K_v), overloads (K_o), misalignment (K_m), and temperature (K_t). When possible, the values for the modifying factors are derived from experience. In lieu of empirical values, suggested values given in reference 8 are used. Digital computer programs sometimes are utilized to determine form and geometry factors for gears.

Allowable values for gear-tooth-root bending stress for given cycle life have been established by test for carburized and nitrided steels (refs. 3 and 9). The special-quality materials, processes (e.g., shot peening), and tolerances developed and refined for turbopump gears allow use of higher stress levels.

Figure 6 presents the allowable bending stress as a function of the cycle life for carburized

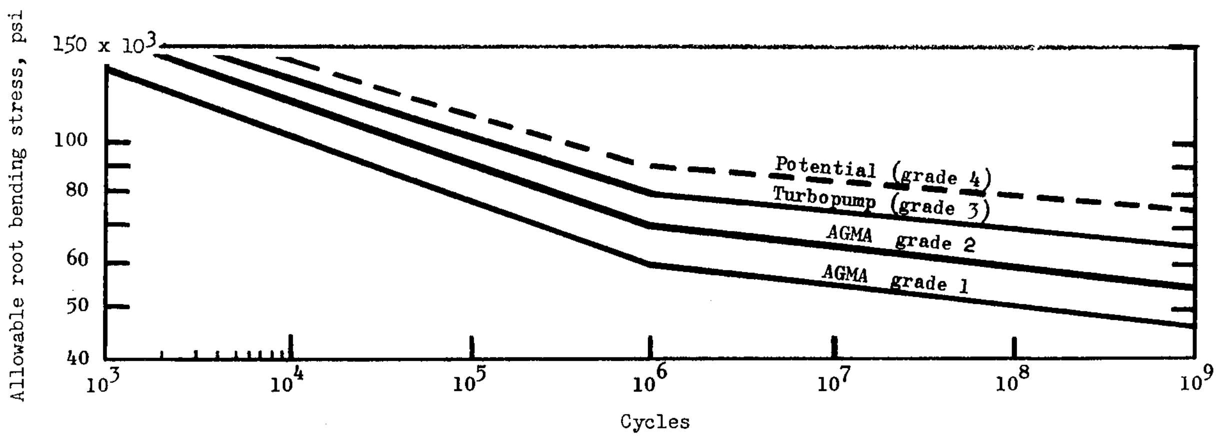


Figure 6. – Allowable root bending stress vs cycle life for gears of four material quality grades.

steels used in aerospace gears of various material quality grades. The stress values for AGMA quality grades 1 and 2 are taken from reference 3 and represent aircraft practice. Material grade 3 represents the stress levels attained for turbopump gears under present practice, and grade 4 represents the levels attainable with the finest available design, manufacturing, and lubrication technology. Figure 7 relates the probability of tooth breakage to bending stress and gear quality. Note that the scale on the abscissa is not truly logarithmic but has been altered to obtain a linear plot; the relationship presented is nonlinear.

2.1.7.2 TOOTH FACE COMPRESSIVE STRENGTH

Gear tooth capacity to withstand compressive stress is reflected in the tooth pitting index K that is used in preliminary designs:

$$K = \frac{W_t}{Fd}\left(\frac{m_G + 1}{m_G}\right) \quad \text{(external gears)} \tag{2a}$$

$$K = \frac{W_t}{Fd}\left(\frac{m_G - 1}{m_G}\right) \quad \text{(internal gears)} \tag{2b}$$

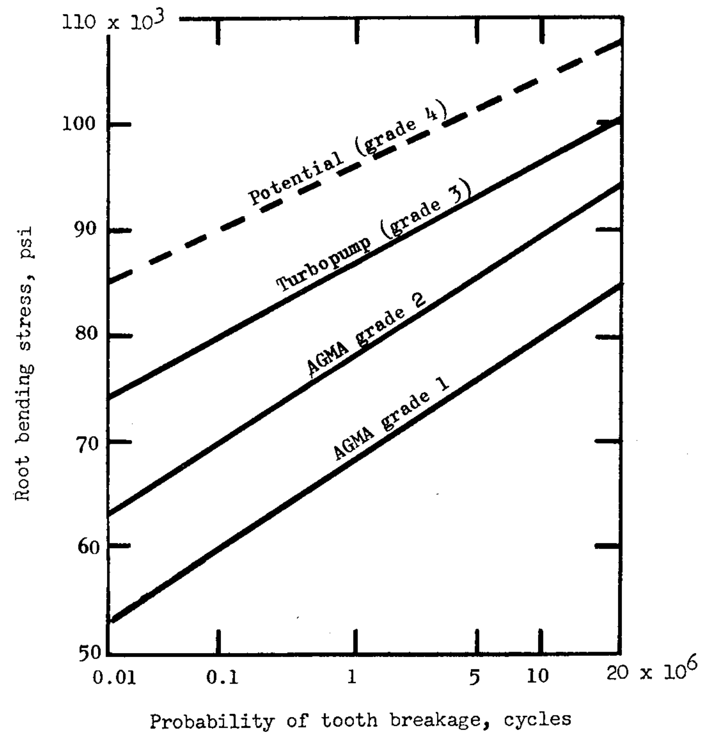


Figure 7. – Probability of tooth breakage vs root bending stress for gears of four material quality grades.

where

K = tooth pitting index, dimensionless

W_t = total tangential tooth load, lbf

F = effective face width, in.

d = pitch diameter of pinion, in.

$$m_G = \text{gear ratio} = \frac{\text{number of gear teeth}}{\text{number of pinion teeth}}$$

Table IV presents the K values used in current turbopump gears.

Table IV. – Tooth Pitting Index K for Current Turbopump Gears

Type of gear	PLV, ft/min	m_G	d, in.	K
Main power	18 000	2.1	2.75	1 200
	21 000	3.75	2.00	1 800
	27 000	2.1	3.00	2 050
	13 000	2.3	4.12	2 500
Accessory	2 000 to 7 000	1 to 5	1 to 6	500 to 1 000
Actuator	0 to 100	∞ (rack	2.5	2 000

For preliminary design, the tooth contact compressive stress may be estimated as 6500 $K^{1/2}$ for 25° PA gears, and as 7100 $K^{1/2}$ for 20° PA gears; a closer determination is made by applying derating factors depending on load application, misalignment, surface conditions and quality (refs. 10 and 11). For final design, values for compressive stress are calculated by the methods given in reference 11.

Figure 8 presents the allowable compressive stress level as a function of cycle life for

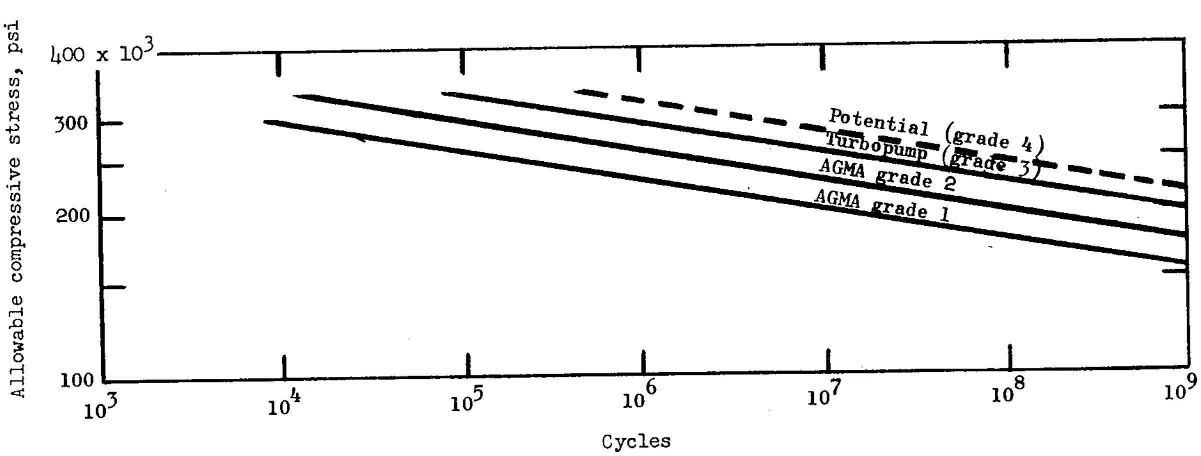


Figure 8. – Allowable compressive stress vs cycle life for gears of four material quality grades.

carburized steels used in aerospace gears of various material quality grades. The stress values therein for AGMA quality grades 1 and 2 are taken from reference 3 and represent aircraft practice; the stress values shown for the material quality grade 3 in figure 8 are allowable compressive stresses for turbopump gears designed and fabricated with present practices; the curve labeled grade 4 represents the stress levels attainable with maximum use of presently available technology in design, manufacturing, and lubrication. Figure 9 presents the probability of tooth pitting as a function of compressive stress (as before, the scale has been modified to obtain a linear plot).

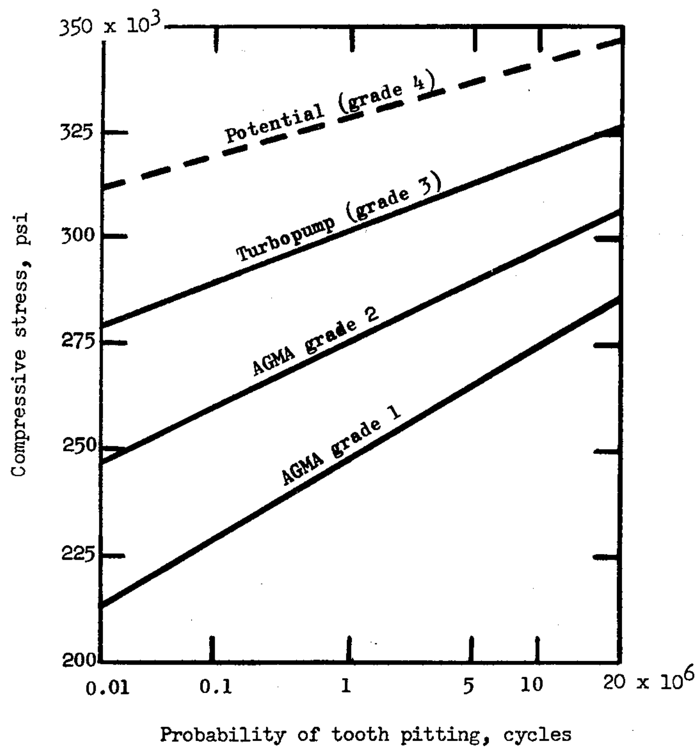


Figure 9. – Probability of tooth pitting vs compressive stress for gears of four material quality grades.

2.1.7.3 CHIPPING RESISTANCE

Chipping of tooth edges may result in progressive degradation of the gears as a result of (1) loss of load capacity because of reduced load-carrying surface and (2) contamination of the gear system.

The potential for chipping exists in nitrided gears because of the extreme brittleness of the nitrided case; the condition is aggravated by the corner buildup that occurs in the nitriding process. Chipping tendencies are reduced by

- Limiting case depth in thin sections
- Establishing a minimum tooth tip width
- Providing adequate radii and blends at corners of tooth tips, ends, and edges.

Tip, end, and edge radii on the gear tooth are controlled carefully to prevent excessive loss of active surface resulting from unacceptably large radii or stress concentrations arising from insufficient radii. Because these radii often are hand ground, careful control is exercised to prevent wide variation. Chamfers are avoided because they can lead to stress concentrations. On critical gears, radii are established before carburizing and are refined after heat treatment as a final finishing step.

2.1.8 Lubrication and Cooling

Gear lubricants perform a complex function of reducing friction, preventing destructive scoring at the sliding contact, and removing the heat generated by the tooth action. In order to minimize the weight charged against the lubrication system, the designs for large turbopump gear trains incorporate sophisticated flow systems that meter the most effective lubricant at the minimum quantity required to maintain gear system temperatures within material capabilities.

2.1.8.1 HEAT REMOVAL

The lubrication system must remove the heat generated in the transmission as a result of (1) tooth friction losses, (2) windage and oil churning, and (3) bearing losses. The total of these losses for spur gear trains is roughly 0.5 to 0.7 percent of the power transmitted per mesh. The largest loss occurs as a result of tooth friction except in very-high-speed gear trains where oil churning may absorb substantial power. Losses for rolling-contact bearings are generally much less than the associated gear losses. Methods for calculating gear loss based on theoretical considerations or on empirical results are given in reference 2 (ch. 14) and in reference 12.

Low gear losses can be achieved by any or all of the following actions:

- Lowering the lubricant viscosity
- Carefully designing internal contours of the gear case to prevent oil trapping
- Using helical gears; a theoretical advantage is cited in reference 2 (ch. 14, p. 6). (Gear designers are not unanimous on this point, however.)
- Ensuring that more tooth load transfer occurs during the arc of recess than during the arc of approach (ref. 2, ch. 5, p. 18)
- Reducing the gearcase internal pressure to reduce windage.

2.1.8.2 SCORING PREVENTION

The scoring resistance of a gear design is affected by many factors and is not represented by any known single value. The following factors are known to affect gear scoring:

(1) Gear design
- Contact stress
- Sliding velocity
- Material
- Surface roughness
- Accuracy of tooth surfaces

(2) Lubricant properties
- Flash temperature
- Viscosity
- Chemical activity (extreme-pressure compounds react chemically with tooth surfaces)

(3) Lubricant variables
- Temperature
- Flowrate
- Method of application
- Cleanliness

Table V summarizes the various scoring indexes that predict scoring resistance based on a limited number of variables; also listed in the table are factors that probably affect scoring resistance but have not been incorporated into any scoring index formula. The scoring indexes in descending order of apparent accuracy are (1) Bodensieck specific film thickness (ref. 13), (2) AGMA flash temperature (ref. 14), and (3) PVT (ref. 10, p. 53). PVT is utilized only as a rough estimate of scoring risk and is not considered a valid basis for design. Reference 16 presents an evaluation of various scoring indexes.

2.1.8.3 LUBRICANT PROPERTIES

Heavily loaded turbopump gear trains have been lubricated with petroleum-base oils, synthetic-base oils, and fuel-with-additive mixtures. Gaseous hydrogen has been used as a coolant in conjunction with dry-film lubricants applied to the gear teeth, and with a mist additive. Various propellants have been tested for load-carrying ability, but are not used as lubricants in operational turbomachinery.

A summary of the properties of turbopump gear lubricants is given in table VI. The -30° F minimum operating temperature requirement of some vehicles (Thor, Atlas, Titan) led to a search for low-temperature oil-type lubricants. MIL-L-7808 oil[1], used in the Titan engines, possesses insufficient scoring resistance for the Thor and Atlas gear trains. MIL-L-25336 oil was developed to fulfill both the Thor and Atlas high-tooth-load and low-temperature requirements; this lubricant tends to deteriorate in storage and must be checked periodically for scoring resistance.

---

[1]Appendix B presents complete titles for material specifications.

Table V. – Factors Involved in Various Gear Scoring Indexes

Scoring factors[a]	Scoring index				
	Specific film thickness (ref. 13)	Flash temp. (ref. 13)	PVT[b] (ref. 10)	PV[b]	Contact time (ref. 15)
Constant load	X	X			
Instantaneous load	X				
Unit load	X	X	X	X	X
Rolling velocity	X	X	X	X	X
Sliding velocity	X	X	X	X	X
Entraining velocity	X	X			X
Slide/roll ratio (specific sliding)					
Radii of curvature of tooth	X	X	X		X
Surface roughness	X	X			
Gear accuracy	X	X			
Initial temperature	X	X			
Material constant	X	X			X
Conductivity					
Tooth load sharing	X	X			
Profile modification	X	X			
Oil viscosity	X				
Tooth surface topography	X				
Waviness					
Lay					
Surface hardness					
Extreme-pressure property of lubricant					
Density of lubricant					
Specific heat of lubricant					
Coefficient of friction	X	X			
Overloads (nature of application)					
Lubricant jet velocity					

[a]Factors marked with X enter into the calculation of the listed scoring index.
[b]P = Hertz contact pressure, psi; V = sliding velocity, ft/sec; T = distance from pitch point to tip of tooth, in.

Table VI. – Properties of Turbopump Gear Lubricants

Property	Lubricant							
	MIL-L-6086		MIL-L-7808		MIL-L-25336		Fuel-additive mixture[a]	
	Petroleum base		Diester base		Diester base		Hydrocarbon compound	
	Required value	Typical value	Required value	Typical value	Required value	Typical value	Required value	Typical value
Viscosity, centistokes								
210° F	–	4	3 min.	5	3 min.	4	–	–
100° F	25 to 34	30	11 min.	17	11 min.	12	–	2
–40° F	–	30 500	–	1300	–	1500	–	11
Flash point, °F	280	Pass	400	430	400 min.	450	110	130
Pour point, °F	–40	Pass	–75	Pass	–75 max.	Pass	–36 (freezing point)	Pass
Load-carrying capacity, Ryder gear test:								
Load, ppi	None	3 450[b]	1700 min.	1900 to 3100[b]	2800 min.	2500 to 5000[b]	Not defined	4000 to 6300[c]
$\frac{\text{Test material}}{\text{MIL-L-6081}} \times 100$	–	140 to 160[b]	76 (2 tests) 72 (4 tests) 70 (6 tests) 68 (8 tests)	76 to 130[b]	116 (2 tests) 111 (4 tests) 109 (6 tests) 107 (8 tests)	116 to 100[b]	–	140 to 200[b]
Load-carrying capacity, Shell four-ball tests	40	50[d]	–	40[d] 80[e]	–	22 to 25[d]	–	29 to 33[d]

[a]RP-1 with 2 to 3 percent of Oronite 262 additive.
[b]Rocketdyne data; test: Federal Standard Test Method 791, method 6508.1 (see Appendix B for complete designation of this and other referenced test methods).
[c]Shell Research Corporation data.
[d]Rocketdyne data; test: Federal Standard Test Method 791, method 6503.2.
[e]Pratt & Whitney data; test: ASTM D-2596.

Maximum scoring resistance has been obtained with a mixture of RP-1 and Oronite 262, a zinc dialkyldithiophosphate additive (ref. 17). Turbopump proof-test runs are conducted with 10 percent by volume of the additive; in subsequent operation, the lubricant is RP-1 fuel mixed with 3-percent additive. The 10-percent-additive concentration used to "run in" the gears gives added scoring resistance in subsequent operation, apparently because of a residual extreme-pressure film. A heater blanket is used to maintain a relatively constant viscosity of the additive over the ambient temperature range expected and thus prevents excessive variations in additive concentration.

The RL10 turbopump gears (AMS 6260) are cooled by hydrogen. These gears operate successfully at approximately 250 ppi (pounds per inch of face width) face loading at 15 800 ft/min PLV. Except for the gear bore, which is chrome plated, a dry-film lubricant[1] is applied to the entire gear for lubrication of the active tooth contacting surfaces and for corrosion protection of the rest of the gear. The hydrogen is injected into the gear case as a liquid but probably performs its cooling function during vaporization and as a gas.

Some rig testing has been conducted in which gears untreated with solid lubricants were run while submerged in various propellants (refs. 17 and 18). These tests showed that rocket engine propellants, although they may be good coolants, are poor gear lubricants and that the materials compatible with propellants are unsatisfactory for gears. High wear rates and extensive scoring occurred at face loads of 500 to 1000 ppi at a PLV of 10 000 ft/min with the fuels RP-1, liquid hydrogen, ethylene diamine, UDMH (unsymmetrical

[1]The dry-film lubricant consists of 1 part by weight of a powder mixed with 3 parts by weight of AMS 3132 varnish with AMS 3170 thinner as required. The powder consists of 10 parts by weight of molybdenum disulfide and 1 part of graphite powder. The coating thickness is specified to be 0.5 to 2.0 mils.

dimethylhydrazine), and N_2H_4; oxidizers giving similar results were liquid oxygen, IRFNA (inhibited red fuming nitric acid), and N_2O_4 (ref. 18). As a result, cooling with propellant has been confined to low load and speed levels.

2.1.8.4 LUBRICANT DELIVERY SYSTEM

The Titan engine turbopumps use a recirculating oil system with a feed pump, scavenger pump, and heat exchanger. The single-pass systems used in lubrication systems for Thor, Atlas, and Saturn S-IB engines are operated by gas pressurization of the lubricant tank (Thor), positive displacement pumps (Atlas), or a fuel-additive blender unit activated by fuel pump pressure (S-IB).

In the Titan, Thor, Atlas, and S-IB systems, the lubricant spray streams are directed to the disengaging side of the mesh. Some designers prefer lubricant impingement on the engaging side of the mesh. The technical rationale for the choice is summarized in the table below.

Further discussion of lubricant delivery may be found in references 2 (ch. 15), 19, and 20.

Lubricant delivery point	Advantages	Disadvantages
Engaging side	Provides maximum potential for elastohydrodynamic film generation.	Trapping of oil between tips of teeth and roots of meshing teeth may result in surface erosion of teeth.
	Allows use of lower lubricant pressure	Requires tight control of lubricant flow.
Disengaging side (preferred for high-speed gears)	Provides cooling at the point where gear tooth surface is hottest; heat is removed before it is conducted into gear mass.	Requires careful targeting and high lubricant velocities.
	Reduces the possibility of gear coolant trapping.	Most lubricant may be thrown off before next mesh occurs.

Some gearbox bearing failures have been associated with a change in lubricant circulation caused by a progressive drop in gearbox internal pressure during flight. A concurrent increase in foaming of the lubricant also detracted from the cooling effectiveness of the lubricant. Remedial practice has been to (1) redesign the bearings, (2) use lubricants with low foaming tendencies, or (3) pressurize the gear case to improve lubricant circulation by increasing windage.

2.1.9 Gear Case

To achieve rigidity and light weight, gear cases are made of light metal (aluminum or magnesium) castings with integral mounting pads and stiffening rings and ribs. The gear-case loads arise from several sources:

(1) Gear tangential driving and separating loads
(2) External loads reacted through the gear case (e.g., pump and turbine thrust and radial loads)
(3) Loads arising from the use of the gear case as the turbopump mounting
(4) Internal pressure
(5) Thermally induced loads (e.g., from cryogenic pumps and hot turbines)

Cryogenic pump volutes often are pin mounted to the gear case (fig. 4) to minimize heat flow and prevent uneven chilling of the gear case and consequent misalignment. Electric heaters sometimes are used to reduce the cooling influence of cryogenic propellants.

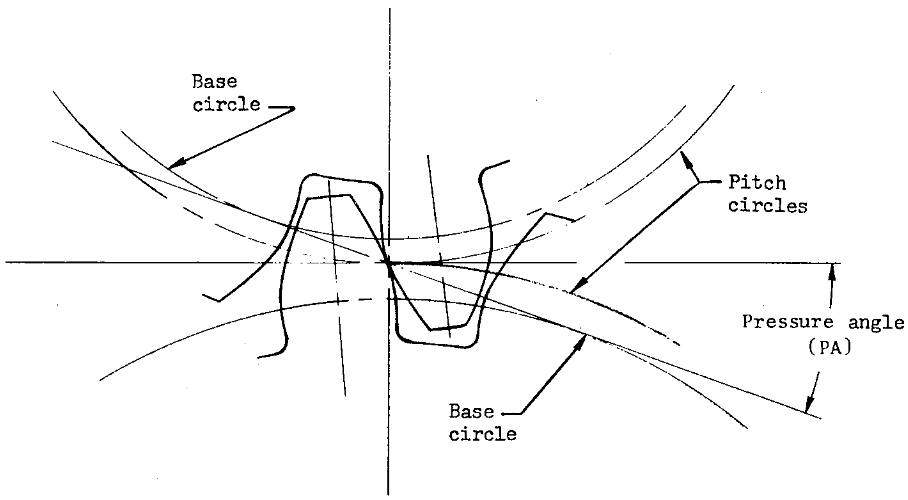
The gear cases for the turbopumps in early Atlas and Thor engines were made in two halves clamped together by bolts. The bearing bores were line bored with the gear case assembled, and relocation of gear-case halves was obtained with dowel pins. Although adequate for the original design loads, the split gear case did not posses sufficient rigidity to maintain gear alignment under the higher loads accompanying subsequent uprating. The solution was to redesign the gear case as one-piece construction.

Gear-case design must include provisions that minimize or eliminate internal fasteners (nuts, bolts, screws, safety wire, and snap rings) that might loosen or back out because of vibration during operation. Joints in the gear case are clamped tightly enough so that friction prevents relative movement of the surfaces. Recessed static seals such as O-rings are used rather than gaskets, because gaskets allow relative motion of the flanges and require more fasteners to prevent bowing and leakage between fastener locations. To confirm design calculations, an instrumented gear case is subjected to full design torque while dial indicators, strain gauges, or brittle lacquer detect deflections.

In the development of a design that will satisfy the fixed nominal center-distance requirements, the effects of thermal contraction resulting from differing materials or thermal gradients are accounted for so that negative or excessive backlash or tip interference does not occur. Changes in center distance are compensated for in design by providing sufficient tip clearance to avoid interference at minimum center distance. As noted previously, involute gears tolerate moderate variations in center distance. Lead modifications such as crowning of the teeth (sec. 2.2.5.5) are employed to accommodate shaft tilt resulting from distortions of the gear case. Gear load capacity is reduced or allowable gear stress levels are decreased to allow for the effects of misalignment (ref. 8, sec. 6).

2.2 GEAR DETAIL

Adequate gear strength depends on tooth size, pressure angle (see sketch below), number of



teeth, and face width. Tooth profiles often are modified to compensate for expected elastic deflections of the teeth under load. Surface textures are specified to avoid scoring and fatigue failures. Rim and web proportions are designed to avoid both excessive vibration and excessive deflection from tooth loads.

Although the gear system configuration often dictates gear diameter, some changes in this gear dimension may be necessary to satisfy the combined requirements of component spacing, speed ratio, and minimum pinion size to achieve adequate tooth strength. Nonstandard center distances sometimes are used to achieve specific speed ratios. The results of detail design may indicate that a change of diameter is required to obtain sufficient strength; the design procedure is then iterated until a satisfactory design emerges.

2.2.1 Pressure Angle

Relatively high (25°) pressure angles are favored for turbopump gears because the larger radius of curvature reduces contact stress and the wider base increases beam strength of the tooth. High pressure angles also permit the use of fewer pinion teeth without excessive undercutting (ref. 10, pp. 14-17).

2.2.2 Number of Teeth

The number of teeth is chosen to satisfy the following conditions:

(1) The speed ratio shall be that specified by system requirements.
(2) The tooth size shall provide bending strength adequate for the design loads.
(3) The frequency of tooth meshings must not coincide with natural frequencies of the gear system. In cases of resonance, changes are made to the gears or other system components, or damping methods are employed (refs. 21 through 23).
(4) The pinion teeth must not have excessive undercutting.
(5) Tooth contact must not result in excessive compressive stresses, which will occur if too few teeth are used.

Iteration of the design analysis is required until all the above conditions are met. A detailed analysis of root bending stress and face compressive stress based on the AGMA methods (refs. 3, 8, and 11) is performed for each turbopump gear.

The maximum allowable number of teeth is a function of manufacturing and inspection costs; the practical limit is regarded as approximately 100. To ensure hunting-tooth action and maximize life, the number of teeth in pinion and gear are selected so that no common factors exist.

2.2.3 Contact Ratio

Contact ratio can be visualized as the average number of teeth in contact. A high (1.5) contact ratio contributes to a smooth transfer of load from one tooth to the next (ref. 10, p. 55). In the Mark 4 turbopump (Atlas sustainer), a stub-tooth design with low contact ratio was replaced with a full-depth design; the resulting increase in contact ratio contributed to a great improvement in life and reliability of the gear set. A low contact ratio increases the severity of dynamic loads and causes premature tooth breakage.

2.2.4 Face Width

Experience has shown that width of a spur gear tooth should be limited to 0.5 to 0.7 times the gear pitch diameter. The accuracy of alignment required to prevent load concentration near the ends of wider teeth is difficult to achieve. Some designers contend that accurately mounted and machined double helical gears can have a total face width twice the pitch diameter. A rule of thumb used by some designers is to limit the face width to six times the circular pitch.

2.2.5 Tooth Proportions

Involute gear tooth profiles are defined by proportions specified in standardized dimensional systems such as the standards for coarse-pitch (P_d = 1 to 19.99) involute spur gears (ref. 24) and for fine-pitch ($P_d \geq 20$) involute spur and helical gears (ref. 25). Summaries of gear proportion systems are given in reference 2 (ch. 5). Modifications to the proportions given in the standards generally are made for high-power gears in order to achieve maximum tooth strength and durability. These profile modifications are compromises made to achieve solutions to particular problems and are used only when necessary. The terms and symbols used in the following sections are illustrated in figure 10; a more complete compilation of definitions and terms with corresponding symbols and abbreviations is given in reference 1.

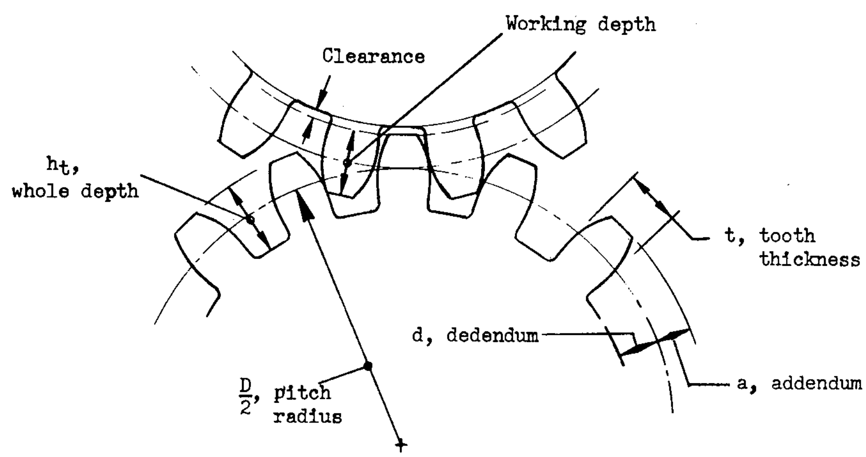


Figure 10. – Sketch illustrating terms and symbols for tooth proportions.

2.2.5.1 WHOLE DEPTH

Full-depth tooth forms ($h_t \geq 2.00/P_d$) are preferred for turbopump gears because smoother action is obtained with the resulting higher contact ratio. Some designers, however, believe that the smaller length of action on a stub tooth ($h_t < 2.00/P_d$) represents a reduction in scoring risk.

The following factors are considered in determining the tooth whole depth:

- Strength
- Contact ratio
- Maximum fillet radius
- Grind stock
- Addenda proportioning to avoid undercut and achieve balanced strength
- Sliding velocity and scoring risk
- Availability of cutters.

2.2.5.2 TOOTH THICKNESS

Tooth thickness must be established to provide both desired tooth strength and backlash; therefore, it is one of the important design calculations made. If pinion tooth modifications are made to avoid undercutting, required backlash is obtained by thinning the gear teeth.

2.2.5.3 ADDENDA

Pinion tooth addendum often is increased and gear tooth addendum is decreased from the standard proportions to (1) eliminate undercutting of pinion teeth, (2) balance the bending strengths of the pinion and gear teeth, and (3) equalize (thereby reducing the maximum) sliding velocities. A compromise must be made in the effort to satisfy all three of these objectives; most designers give priority to balancing pinion and gear strength. The addendum requirements for avoiding undercut of pinion teeth and achieving equal sliding velocities have been developed (refs. 2 and 10). Figure 11 presents addendum values required to obtain equal strength for spur gears with 20° PA.

2.2.5.4 ROOT FILLET

Fillet radii are maximized to reduce root-bending-stress concentration. Photoelastic studies have shown the importance of fillet radii (ref. 26). To determine the maximum allowable radii, enlarged (20X to 100X) layouts are made of the gear tooth. The allowable fillet size also is controlled by the manufacturing method; the gear tool manufacturer must review the fillet radii chosen.

2.2.5.5 TOOTH-FORM MODIFICATION

Gear teeth with loads in excess of 1000 ppi often are modified from a pure involute form to compensate for errors of manufacture, mounting deflections under load, and tooth deflections. The goal is to achieve a perfect involute profile under load, and tooth

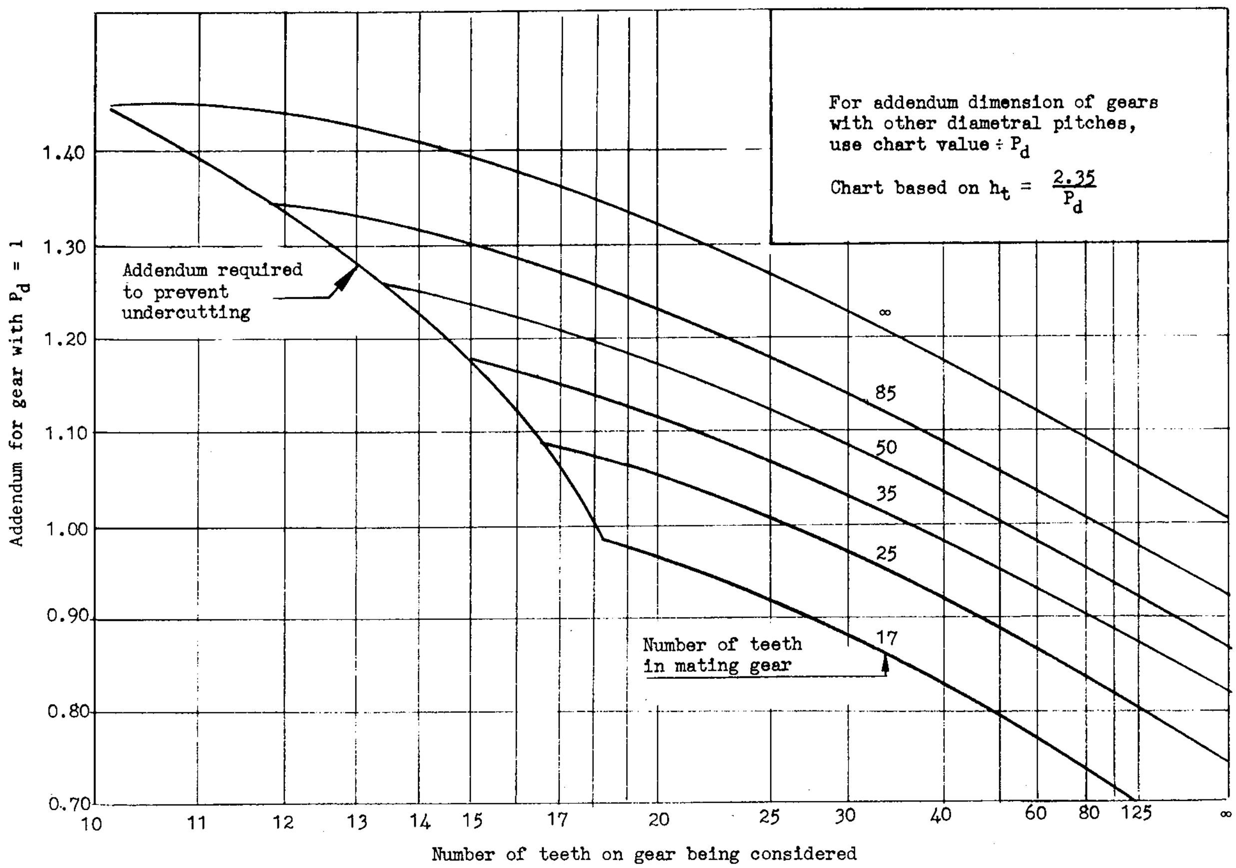


Figure 11. – Addenda values for equal strength vs number of gear teeth (20° PA spur gears).

modification usually takes the form of tip relief or a combination of tip and flank relief. Methods for calculating required modifications are presented in reference 27. Load concentrations near the ends of teeth caused by misalignment are minimized with lead corrections (axial modifications) that consist of crowning or end easing the teeth (i.e., making them circumferentially thinner near the ends than in the center). The amount of crowning or end easing depends on the design loads and expected misalignment. Crowning may range from 0.0005 to 0.002 in. (ref. 2, ch. 5). In high-precision gears where good alignment is maintained, crowning greater than 0.0006 in. is avoided because it results in a reduction of gear load capacity.

Lead modifications also are used to provide more even load distribution on teeth with torsional windup on gears with face widths of 1 inch or more. Lead modifications generally are used to correct scoring noted in service and therefore are based on experience.

2.2.5.6 SURFACE TOLERANCES

Since it is not possible to produce a perfect gear tooth surface, tooth involute profiles and axial surfaces (lead profiles) are allowed to deviate from the nominal by an amount dependent on the gear load, speed, and smoothness of operation required by the application. The allowable deviations are defined on the gear drawing, which includes sample gear inspection charts. Lead and involute surfaces (fig. 12) are translated by

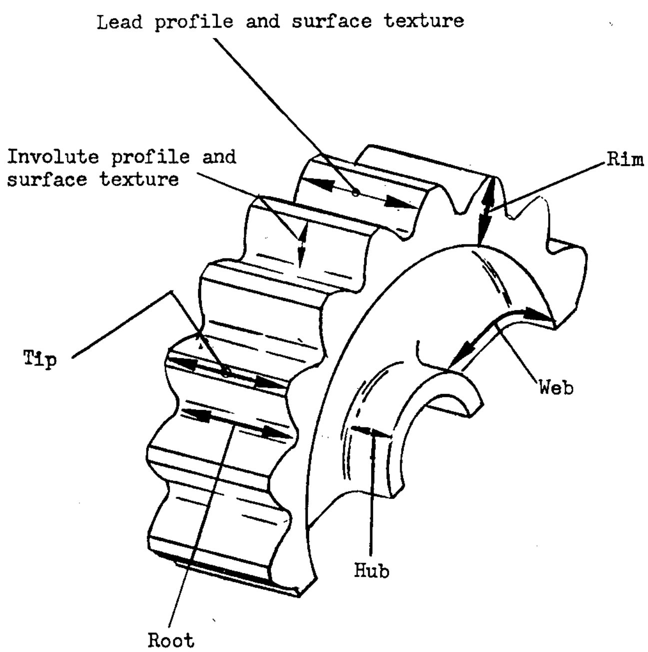


Figure 12. – Sketch illustrating terms for gear surfaces.

inspection machines to lines traced onto chart paper from the actual gear tooth. The rolloff chart translates an involute curve to a straight line, so that deviations of the tooth surface from a perfect involute are readily observed. Desired profile tolerance bands including profile or lead modifications are reproduced on the sample inspection charts (fig. 13); gear tooth traces are required to fall within the tolerance band to be acceptable. Additional restrictions placed on the involute and lead profiles are that they should not be concave and that the rate of reversal (change in direction of the surface trace) should not exceed a rate determined by the severity of service. The normal limits for power gears with diametral pitch in the range 8 to 12 are as follows:

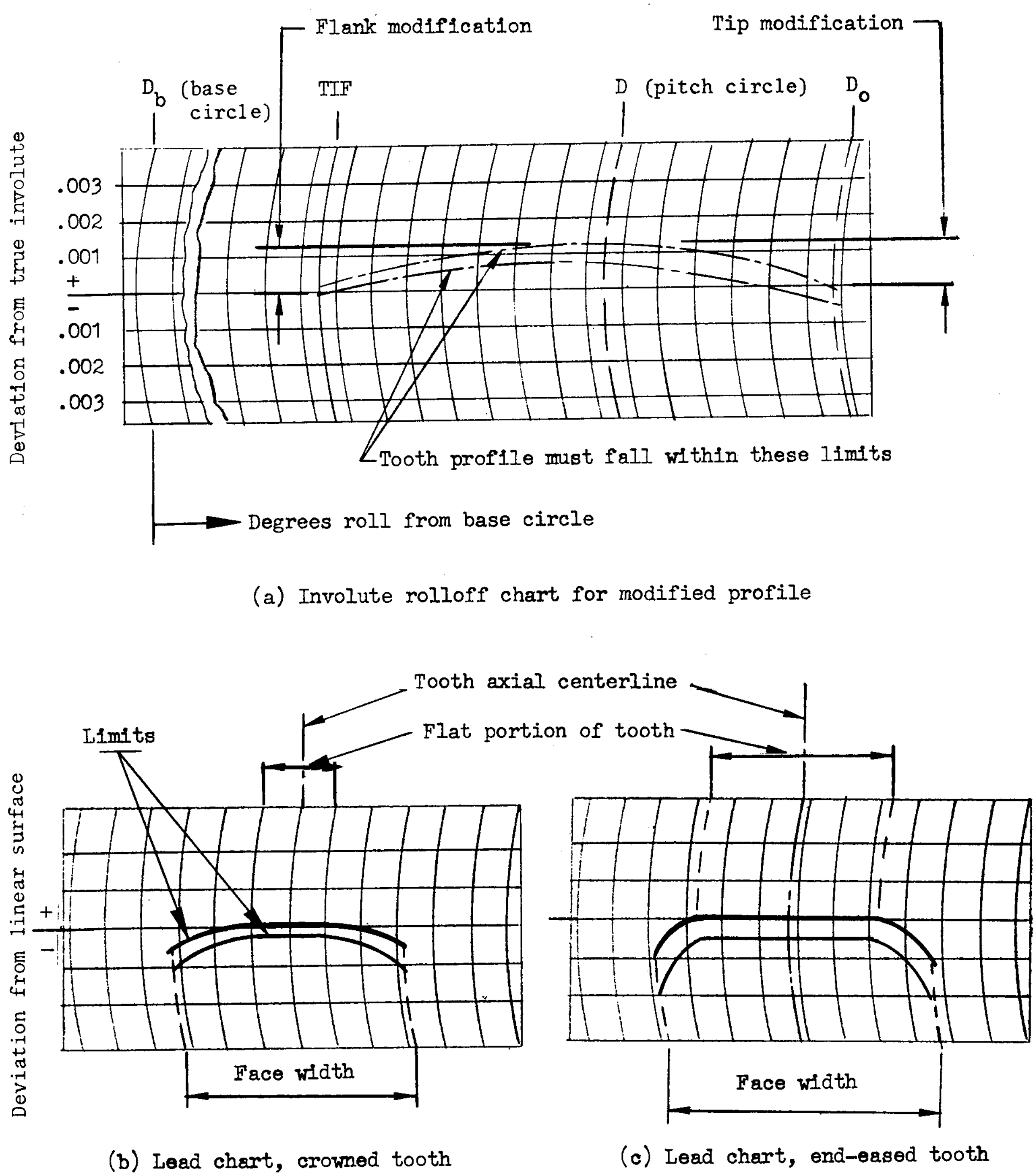


Figure 13. – Sample inspection charts for involute and lead.

Type of gear	Allowable rate of reversal, in. for any 25% of active profile
Main power (heavy loads)	0.0002
Accessory and lightly loaded power	0.0003

2.2.5.7 SURFACE TEXTURE

The surface textures of both active and nonactive gear surfaces greatly influence the life of gears and therefore are carefully controlled by design. Roughness values of 6 μin. AA (arithmetic average) to 20 μin. AA for gear tooth contact surfaces have been found to give satisfactory service and are within the production capability of gear manufacturers. Both finer and coarser finishes tend to score and are avoided. Waviness limits are not presently used, but limited testing indicates that the maximum peak-to-valley height should not exceed 50 μin. When defined, waviness specifications will control surface quality in the peak-to-peak frequency range between surface roughness and the rate of reversal requirements.

The surface quality of noncontacting areas (rims, webs, lands, tooth ends, and roots) has an effect on fatigue life. Tool marks, surface tempering, and gouging in the root fillet are examples of detrimental conditions. Similar marks on lands and tooth ends must not lap over onto active contacting surface. Reference 28 discusses the beneficial effects of maintaining surface integrity.

Tests have shown that the surface roughness changes during operation. Surfaces with roughness values of less than 4 μin. AA increased to 10 to 12 μin. AA during operation. Other gears with a surface roughness of 26 to 32 μin. AA smoothed out to 10 to 14 μin. AA during a run-in process. Scoring occurred on gears with roughness over 32 μin. AA. Surface-roughness improvement has been noted in operation with fuel-additive (RP-1 plus 2% Oronite 262) lubricant, but mixed results have been obtained with petroleum and diester-based oils. Shot peening, vapor blasting, or special grind control of critical noncontacting gear surfaces has resulted in marked improvement in gear life (ref. 28). Directions of surface texture measurements usually are specified on the drawing (fig. 12) to ensure that the design goals are achieved.

2.2.6 Rim and Web

Rim and web designs often are dictated by the operating vibration spectrum. Thin rims and webs ($< h_t$) in particular are prone to vibration problems. Large radii are used to blend rims to webs and webs to hubs to reduce vibration-caused fatigue failures of the web. Natural frequencies coinciding with operating forcing frequencies are avoided by altering rim and web shape, mass distribution, and size, number, and shape of lightening holes. Shot peening of rims and webs sometimes is effective in preventing web fatigue failures. A web fatigue problem that occurred on uprating of the Mark 3 gear train was eliminated by increasing the web thickness, increasing rim cross section and blend radius, and shot peening the web surfaces (fig. 14).

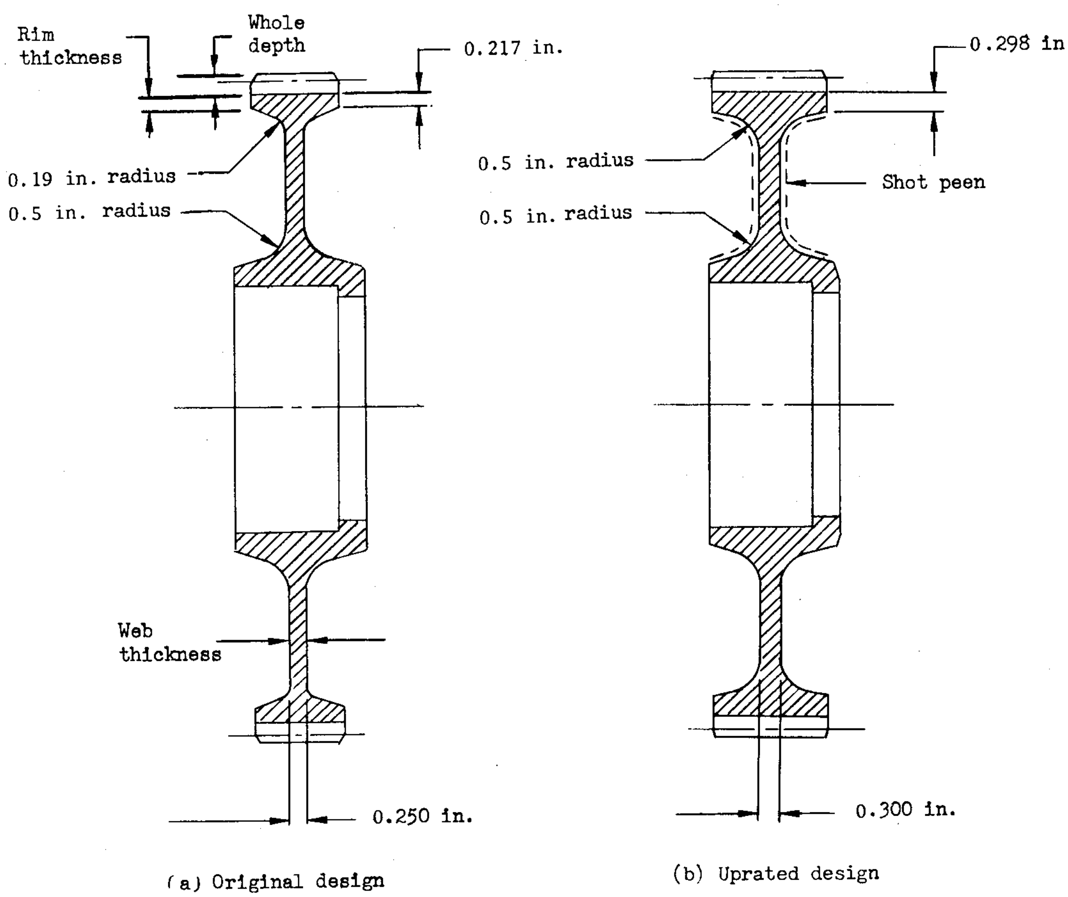


Figure 14. – Sketches illustrating rim and web dimensions (original and uprated designs).

On highly stressed power gears, the rim and web dimensions are minimized to achieve adequate power capacity and stiffness at minimum weight. Rim and web are sized to avoid resonant frequencies. The proportions usually are based on experience and are given as a proportion of the tooth whole depth. Changes are made if testing indicates a weakness. For example, tooth breakage failures in Titan II idler gears were traced to a resonant condition between the gear wheel natural frequency and the mesh frequency. The wheel contour was changed to alter the wheel natural frequency, and the problem was eliminated.

Rims and webs of small gears and those with low stress levels normally are sized to achieve manufacturing ease and low cost. Lightening holes often are placed in the webs of low-cost or lightly loaded gears. Care is taken to ensure that the spokes between the lightening holes are not too thin. Normal spoke circumferential dimension is from 0.5 to 1.5 times the size of the lightening holes.

The natural frequencies of gears can be estimated by analytical methods, but normally the frequencies also are determined experimentally by shaking simulated or actual gears and observing resonances (ref. 21). Sand or "popcorn" salt[1] is used to reveal mode patterns. Interference diagrams (also called Campbell diagrams) are plotted for the various modes to determine whether any modal-shape standing-wave frequencies coincide with gear meshing frequency.

2.2.7 Tolerances

As part of the effort to hold down gear costs, tolerances are made as large as possible consistent with the required gear reliability. The AGMA has established tolerance classes suitable for different applications (ref. 4). The entire spectrum of current aerospace gears requires tolerances that are a combination of AGMA classes 9 through 14. In general, rocket engine turbopump power gears require tighter tolerances than other aerospace power gears. Most turbopump power gears meet class 13 limits, while accessory gears are made to modified class 10 limits. Present manufacturing capabilities to meet tolerances are summarized in table VII. Reduction of the tolerances below the values shown in the table will result in a corresponding increase in cost. Achieving the levels of accuracy shown requires close liaison and special coordination of measuring-instrument calibration between the manufacturer and the gear user.

The accuracy with which gear dimensions can be measured should be a factor of 10 finer than the manufacturing tolerance, although this ratio is not always achieved. Table VIII lists the limits of equipment used for measuring gears produced in quantity in production runs,

[1]Popcorn salt is sodium chloride that is finer than table salt but coarser than talc. The smaller grain size provides visual evidence of vibrations with frequencies higher than those that can be seen with sand.

Table VII. – Gear Manufacturing Tolerances[a,b]

Tolerances on tooth elements					
	Finishing method				
Dimension	Hobbing	Shaping	Shaving	Grinding	Honing[c]
Involute	5	5	3	2	2
Lead, in./in.	3	3	2	1	1
Tooth-to-tooth spacing	15	15	15	5 to 10	5+
Whole depth	30	30	30	5	–
Fillet radii	50	50	50	10	–
Circular tooth thickness	30	30	30	15	5
Out-of-roundness	25	25	25	10	10
Concentricity	25	25	20	5	5
Surface roughness, μin. AA	125	63	32	16	20

Tolerances on gear body elements			
	Finishing method		
Dimension	Machining	Grinding	Grinding and polishing
Journal diameter	5	1	1
Bore diameter	5	1	1
Journal concentricity	10	2	2
Bore concentricity	10	2	2
Journal-to-bore concentricity	15	2	2
Tooth element concentricity	20	10	10
Taper	5	1	1
Parallelism	20	2	2
Hub dimensions	20	1	1
Web dimensions	20	5	5
Rim dimensions	20	2	2
Fillet dimensions	50	10	10
Surface roughness, μin. AA	63	16	4

[a]Except as noted, tolerances in this table represent total ranges and are expressed in units of one ten-thousandth inch (2 = 0.0002 in.).

[b]Highly specialized gear configurations can be produced for specific applications to tighter values for production measurement accuracy. Rejection rate in this specialized field is seldom less than 75 percent.

[c]Tolerances for honing are for corrective machining of values in excess of table values. Honing is not recommended for hardened rocket engine gears because of resultant surface texture.

Table VIII. – Accuracy of Measuring Equipment Used to Inspect Production Runs of Gears[a]

Tooth elements	
Dimension	Measurement accuracy
Involute	1
Lead (axial), in./in.	0.5
Lead (helical), in./in.	1
Tooth-to-tooth spacing	0.5
Accumulative pitch	1
Whole depth	5
Fillet radii	10
Pitch diameter	2
Out-of-roundness	2
Concentricity	2
Surface roughness, μin. AA	2 (linear surface) 4 (curved surface)

Gear body elements	
Dimension	Measurement accuracy
Journal diameter	0.5
Bore diameter	0.5
Journal concentricity	1
Bore concentricity	1
Journal-to-bore concentricity	1.5
Tooth element concentricity	5
Taper	0.5
Parallelism	1
Hub dimensions	2
Web dimensions	2
Rim dimensions	2
Fillet dimensions	2
Surface roughness, μin. AA	2 (linear surface) 4 (curved surface)

[a]Except as noted, values in this table represent total ranges and are expressed in units of one ten-thousandth inch (2 = 0.0002 in.).

Table IX. – Accuracy of Metrology Laboratory Measurements on Gears[a]

Tooth elements	
Dimension	Measurement accuracy
Involute	0.1
Lead, in./in.	0.1
Tooth-to-tooth spacing	0.1
Accumulative pitch space	0.1
Whole depth	0.1
Fillet radii	1.0
Pitch diameter	1.0
Out-of-roundness	1.0
Concentricity	1.0
Surface roughness, μin. AA	2 (linear surface) 4 (curved surface)

Gear body elements	
Dimension	Measurement accuracy
Journal and bore diameter, taper, roundness	0.05
Concentricities and normalities of journals, bores, and tooth elements	0.1
Hub, rim, web dimensions	0.1
Fillet dimensions	1.0
Surface roughness, μin. AA	2 (linear surface) 4 (curved surface)

[a]Table lists the accuracy of measurement (in inches) that can be performed on gear elements. These measurements are used primarily to calibrate production measurement equipment. Units shown are one ten-thousandth inch (2 = 0.0002 in.) unless otherwise noted.

while table IX presents the limits achievable in the most specialized metrology laboratories. The laboratory equipment is used primarily to calibrate production measuring equipment.

2.3 MATERIALS

2.3.1 Gears

The materials used in turbopump gear systems are similar to those developed for aircraft gear practice with some additional refinements. Some attempts have been made to select gear materials suitable for lubrication by rocket propellants; these materials were selected on the basis of chemical compatibility rather than strength and have very limited usefulness.

A summary of the materials used in turbopump gears is presented in table X.

Table X. – Materials for Turbopump Gears

Application	Gear material	Lubricant/coolant	Comments
Critical, highly loaded power gears	AMS 6265	MIL-L-6086 oil MIL-L-7808 oil MIL-L-25336 oil Fuel-additive[a]	Used where high capacity and reliability are required; corrosion protection required.
Moderately loaded gears	AMS 6260 AISI 9310 AISI 4620 AISI 8620	Same as above	Used in applications less critical than those above.
	AMS 6470 (nitrided)		Used for wear resistance; must have smooth edge radii to avoid edge chipping; must have protection from moisture corrosion.
Lightly loaded gears	Any of the above, plus AISI 4340 AISI 4140	Same as above	Used for accessory gears.
Propellant-cooled gears, moderately or lightly loaded	AMS 6260 AMS 6265	LH_2, LO_2, RP-1	Gear material must be protected from corrosion by moisture.
	AISI 440C	LH_2, LO_2, IRFNA, N_2O_4	Gear material very brittle; has some corrosion resistance; in experimental status only.
	Beryllium copper (Berylco 25)	Ethylene diamine, UDMH, N_2H_4, LH_4, GH_2	Gear material low in hardness; has inherent corrosion resistance; in experimental status.

[a]RP-1 plus Oronite 262 (2 to 3% concentration in service, up to 10% during run-ins).

Deep-carburized case-hardened steels similar to vacuum-melted AISI 9310 (AMS 6265) have been found to possess the best combination of properties for power gears; the hardened outer surface resists compressive stress and wear, while the tough ductile core provides resistance to shock loads and has good resistance to bending-stress cycle fatigue. Use of vacuum-melted steel has extended the fatigue life of gears.

The use of corrosion-resistant materials for gears has been limited to experimental programs, because no corrosion-resistant material possesses the combination of hardenability and toughness required for highly loaded power gears. The 300-series steels and the Inconels cannot be sufficiently hardened to withstand high compressive stresses; in addition, these materials tend to score and gall excessively. The truly hardenable stainless steels, such as 440C, are too brittle to withstand dynamic tooth loads.

Beryllium-copper alloy can be heat treated to approximate the tensile strength attained in the core of carburized steel gears, but the surface cannot be hardened to withstand the compressive stresses encountered in power gears. The lower modulus of beryllium-copper alloy allows greater bending deflections but at the same time reduces the peak stresses caused by dynamic loads.

In one experimental program (ref. 18), nitrided 410 CRES was tested as a candidate gear material that would combine corrosion resistance, a hardened surface, and a ductile core. Although the gears were superior ro 440C gears, the extremely brittle hard case chipped at the tooth corners.

2.3.1.1 MATERIAL GRADES

To obtain the load capacity and reliability required for turbopump gears, steel heat lots and many details of the manufacturing processes must be specified. The properties of carburizing steels vary over a wide range, the extent depending on the grade specified. The AGMA has recognized the need for grading materials and processes; two grades are defined in reference 3. Three grade levels for aerospace gear materials have been defined, and a fourth appears potentially useful:

<u>Grade 1 (minimum quality).</u> – Process control is moderately tight; small deficiencies in quality are acceptable. This grade is used for aircraft accessory gears.

<u>Grade 2 (normal quality).</u> – All material properties are required to meet high standards. Rigid inspection and process control (aircraft quality) guarantee that the specified high quality is in fact achieved. This grade is used in aircraft power gears.

Grade 3 (premium quality). – Best material grade that aircraft-quality gear manufacturers can presently provide in production quantities; it is expensive to achieve. Rigid inspection and process control are required. In addition, randomly chosen parts out of production runs are tested destructively. Use of mill lots of steels is specified. This grade is used in turbopump power gears.

Grade 4 (ultimate quality). – Best material that can be manufactured by present technology; no expense is spared to achieve the optimum. Rigid laboratory inspection and process control, including mill lot control of the steel, process analysis, and random destructive testing of parts, are employed. Reduction in cost will follow more widespread use.

2.3.1.2 METALLURGICAL PROPERTIES

Hardnesses achieved in the case and core are major factors in the strength and durability of a gear. The hardenability is dependent partly on the chemical composition of the steel, which is allowed to vary within limits specified by the material designation. To ensure attainment of adequate strength, hardenability tolerances (H-bands) are specified in addition to the material composition and cleanliness requirements. Materials for turbopump power gears are bought in registered mill lots of AMS 6265, a vacuum-melted low-alloy carburized steel. Compliance to the specified hardenability (H-band) limits is documented and certified by the steel producer; the certification is based on end-quench hardness-traverse tests made on a representative sample of the lot of steel.

Banding (segregation of alloy constituents into nearly parallel bands aligned in the direction of metal working) has unknown effects on material properties and thus is considered undesirable in gear materials. Alloy banding has been reduced by hot soaking to induce solution of alloy bands; carbide banding is not amenable to this process because of the high solution temperature required. Vacuum remelting has a beneficial effect in reducing banding because of the smaller mass of a vacuum-melted ingot. Requirements intended to reduce or eliminate banding often are included in gear material specifications.

2.3.2 Gear Case

Gear-case materials are chosen to provide a lightweight, rigid structure for mounting the gears. The materials must be strong, easily fabricated, dimensionally stable, and chemically compatible with the service environment. Lightweight, high-strength castable aluminum alloys (such as A356-T61 or Tens-50) are used for most turbopump gear cases. The advantage of light weight is considered to outweigh the disadvantages of the larger coefficient of thermal expansion (sec. 2.1.9).

Magnesium alloy has been used for gear cases but generally has been replaced by aluminum to avoid electrolytic action and consequent corrosion. Retention of threaded steel inserts also has been a problem with magnesium gear cases.

2.4 FABRICATION

Control of the fabrication processes is necessary to attain desirable material properties and therefore must be considered part of the designer's responsibility. It is especially important to maintain consistency of processes such as forging and shot peening; these processes have great influence on the performance of the gear, but their effects cannot be verified by nondestructive testing.

2.4.1 Forging

Critical turbopump gears are made from forged blanks in order to improve grain orientation; noncritical gears are made from bar stock. Most turbopump power gear forgings are made in specially designed, accurately dimensioned closed forging dies. A typical gear is made from a blank forging in which the outer shape is made 1/16 to 1/4 in. larger in all dimensions than the finished gear shape; after forging, excess stock is removed and the teeth are cut in the blank. For maximum strength, the grain flow approximates the outline of the finished gear shape. High-energy-rate forging has been used to forge complete gear blanks including gear teeth, webs, rims, and hubs. Tests run on these gears have shown increased strength and life due to the grain flow of the metal around the gear-tooth-root fillet (refs. 17, 29, 30, and 31). Plastic-flow tooth forming by gear rolling also has been used to improve grain flow.

2.4.2 Tooth Cutting

Gear teeth are cut in a blank by hobbing, shaping, or green grinding before the gear is carburized, hardened, and finished. It is easier to obtain a superior surface texture by hobbing gear teeth than by other means, but more axial cutter clearance is required. Other machining methods must be carried out with additional care to achieve the surface texture obtained by hobbing. Short-pitch hobs are sometimes used in preshave and pregrind cutting to obtain the maximum undercut and fillet radii while providing some grinding stock at the pitch diameter. These hobs must be designed for a specific gear and are not suitable for cutting a range of number of teeth, as are standard hobs. Reference 10 presents information on gear tooling design. Gear tool suppliers often can furnish capable consultation as well as specialized tooling.

Shaping of gear teeth is used primarily when gears and splines are in axial proximity to other components. Shaping generally does not generate as smooth a finished root as hobbing and therefore is the second choice of tooth-cutting methods.

Grinding in place of a cutting operation often is called "green grinding." Green grinding of tooth forms from solid blanks is possible when gears finer than 20 pitch are made; coarser pitch gears generally require some type of pregrind cutting operation. Green grinding of complete gear forms has some advantages; it is used to obtain a slightly better texture, especially on hard materials, and may be required for materials that work harden. Green grinding is not used in producing turbopump power gears.

2.4.3 Heat Treatment

Power gears are deep case carburized to increase the surface durability of the teeth. Carburization is obtained by exposing the parts to a carbon-rich high-temperature environment; carburizing procedures recommended by AGMA are given in reference 32. Critical power gears are subjected to gas-type carburization under rigidly controlled conditions of time, temperature, furnace-wall materials, and carburizing-medium composition. Variation in the carbon concentration of the carburizing atmosphere programmed into the heat-treatment cycle often is used to obtain a desired gradient in carbon content. Copper plating frequently is used to prevent carburization of areas that are not heat treated. Acid stripping of the copper is avoided because of the potential for introducing hydrogen embrittlement. Case hardening also may be obtained by nitriding the gears. This process produces a superior wear-resistant surface, but it is more expensive than carburizing and results in a more brittle case. Nitriding has not been used for turbopump main-power-train gears.

2.4.4 Tooth Finishing

Most turbopump gears are finished by grinding in order to obtain the best texture and conformance to tolerances. Form grinding has the advantage of permitting the use of a small wheel, thereby requiring only a small amount of axial clearance between parts on the gear shaft. Generation grinding is done with larger wheels and requires more generous axial spacing to give grinding-wheel clearance.

Grinding is used to obtain the required accuracy of all critical contacting profile surfaces. In addition, the roots of accessory and lightly loaded power gears generally are ground. When roots are ground, care must be taken to ensure that a minimum of hardened stock is removed from the critical area of the root. Sometimes only the sides of noncritical gears are

ground, and a greater blending tolerance may be made between the profiles and roots. For heavily loaded (above 2000 ppi) turbopump gears, unground roots are specified in order to avoid removal of hardened case, as it is difficult to determine the amount of case removed by grinding in the roots; grinding also induces a residual tensile stress undesirable in the root section. Residual tensile stress on a gear tooth profile surface may be desirable, since in service this area is loaded in compression.

The required allowance for grinding stock depends on the amount of distortion of the gear due to processing and heat treatment. Ideally, the minimum amount of case should be removed. Enough grinding stock is allowed to ensure that 100 percent of the contacting surface of each tooth is finished.

Many factors contribute to distortion during carburizing and hardening. These factors include gear size, shape, forging design, heat-treating technique, and unrelieved stresses induced in the manufacturing processes preceding heat treatment. Distortion generally is proportional to gear size and face width. Distortion is reduced in large gears by carburizing the complete gear and removing the carbon from nonhardened areas only (webs and hubs). The parts then are quenched so that the heat flow produces minimum distortion; sometimes quenching dies are used for this purpose. Distortion is controlled to 0.003 to 0.005 in., the limit depending on gear size. The "as-carburized" case depth is made approximately 20 percent deeper than the desired "as-finished" depth to allow for finish grinding.

Three areas of grinding are defined on critical hardened gears (fig. 15). The gear profile (zone A, fig. 15) is ground completely to ensure that the entire surface conforms to required design dimensions and that the entire surface is cleaned. Zone B, between the profile and root, may or may not be ground; but no sharp steps or stress risers are allowable. The root fillet, zone C, normally is not ground; this area is protected from grinding and the resulting residual tensile stresses. Grinding on accessory gears may not be critical and may be allowed in the roots of lightly loaded gears.

Grinding abuses that cause retempering and rehardening can be avoided by proper selection of wheel grit, coolants, feeds, and speeds. Nital-etch inspection is used to check for grinding burns on finished gears.

2.4.5 Shot Peening

Shot peening of tooth surfaces and other gear surfaces extends gear tooth fatigue life; increases of 130 to 400 percent have been noted (refs. 33 and 34). Reference 34 defines shot peening terms and processing in detail.

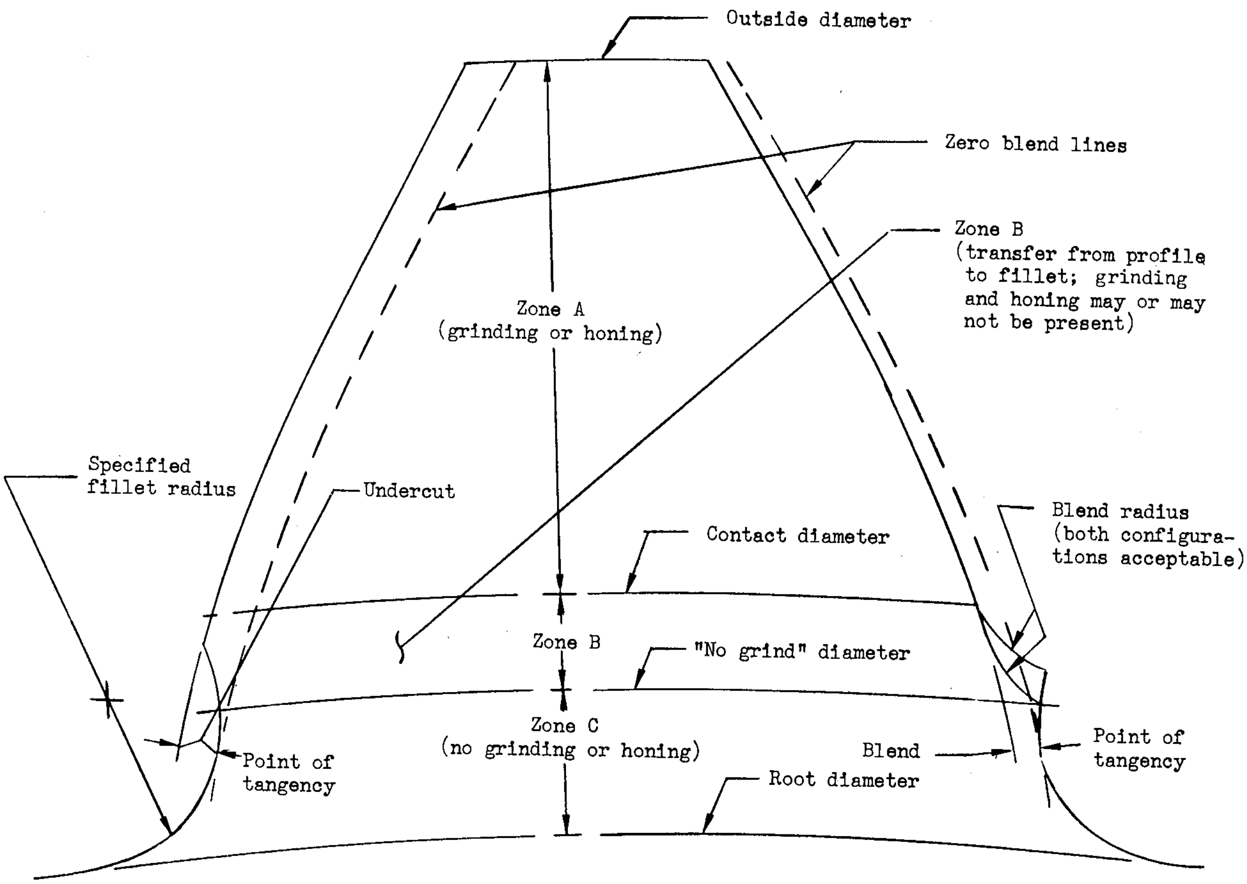


Figure 15. – Sketch illustrating grinding zones on critical hardened gears.

The root fillets of gears are peened to increase the residual compressive stress and to reduce the effects of surface discontinuities of the teeth.

Gear rims and webs of critical gears are shot peened to prevent fatigue cracks from forming in tool marks, nicks, and other surface imperfections that may occur during manufacture. A significant increase in the average fatigue life of the Mark 3 gear train was attributed in part to the effects of shot peening the web.

Shot peening is known to retard stress-corrosion cracking by reducing the tensile stress on the surface. Shot peening before plating helps prevent plating cracks from extending into the base material; however, it is not a substitute for reducing embrittlement potential by baking the part. Multiple peening the same part with different-size shot has further increased

fatigue life (refs. 35 and 36). There is some indication that extremely intense peening creates a very deep residual compressive stress that will reduce subsurface shear failures under compressive stresses (ref. 33). Corrosion resistance and resistance to fretting have been increased by peening with stainless steel or glass. Some stainless steel parts have exhibited decreased corrosion resistance when peened with non-corrosion-resistant cast iron or steel shot. Light peening appears to improve the lubrication performance of active surfaces provided that the surface roughness is not increased greatly.

The effectiveness of peening depends on the preparation of the surface before peening. Minute decarburization, oxidation, and scale on a gear are detrimental and are removed before peening.

Shot size uniformity has an effect on peening effectiveness and is therefore controlled for processing of turbopump gears by invoking specification MIL-S-13165.

An indirect method is used to gage the effectiveness of a shot-peening treatment. An Almen test strip (a piece of spring steel 3/4 in. wide, 3 in. long, in one of three standard thicknesses) is exposed to the same peening as the work piece. When released from the holding fixture, the strip bows into an arc proportional in height to the residual compressive stress on the peened side. For turbopump gears, a peening callout of 0.015A is used, indicating an arc height of 0.015 in. for an A strip (thickness = 0.051 ±.001 in.).

In addition to the arc height, an exposure time is specified. It is expressed as multiple of the time required to achieve indenting of 98 percent of the surface. Turbopump gears are subjected to an exposure of 4X.

2.4.6 Configuration Control

To achieve the necessary control of gear physical characteristics, the gear drawing includes or refers to specifications that include the following items:

- A detailed three-view drawing of the gear
- A data block (see table XI for an example)
- Sample involute roll-off chart based on degrees of roll from base circle (fig. 13(a)) with limits
- Sample lead chart (figs. 13(b) and 13(c))
- Root fillet detail (fig. 15)

This design document is essential in conveying the designer's intent to the manufacturer and to quality-control personnel. Reference 2 (ch. 11) discusses gear drawings and data requirements.

Table XI. – Sample Data Block for Gear Drawing

Tooth element requirements	Turbopump gear dimensions (representative values)	
	Power gears	Accessory gears
Number of teeth	33	70
Diametral pitch, in.$^{-1}$	11	12
Pressure angle, deg	25	20
Pitch diameter, in.	3.000 ref.[a]	5.833 ref.
Involute form (profile)	modified	standard
Base circle diameter, in.	2.71892	5.48152 ref.
Outside diameter, in.	$3.241^{+0.000}_{-0.002}$	$5.966^{+0.000}_{-0.005}$
Root diameter, in.	2.803 ref.	5.664 ref.
True involute form (TIF) diameter, in.	2.890 max.	5.720 max.
Fillet radius, in.	0.028 min.	0.025 min.
Addendum, in.	0.1205 ref.	0.0667 ref.
Whole depth, in.	0.219 max.	0.150 max.
Circular thickness at pitch diameter, in.	0.1650 – 0.1677	0.1269 ref.
Measuring pin diameter[b], in.	0.17454 ref.	0.14400 ref.
Measurement over pins[b], in.	3.3117 – 3.3167	$6.025^{+0.000}_{-0.006}$
Maximum involute profile error[c], in.	±0.0002 (see chart)	±0.0003 (see chart)
Maximum cumulative pitch error (any two nonadjacent teeth)	0.0010	0.0015
Lead error, in. per in. of face width	0.0002 (see chart)	0.0003
Tooth-to-tooth spacing error, in.	0.0002	0.0003
Crown, in.	0.0002 (see chart)	None
Rate of reversal, inches in any 25 percent	0.0002	0.0003
Backlash when assembled, in.	0.003 – 0.007	0.006 – 0.010

[a]"ref." means dimension given for reference only.

[b]These values may be established by the quality assurance department and may not be included in the gear drawing.

[c]All gear values are measured from the axis of the part as designated or from an axis determined by its mounting diameters.

2.5 TESTING

2.5.1 Acceptance Testing

Quality-assurance tests for gear acceptance depend on the gear property considered, the severity of service, the reliability required, and the service history of the gear.

Quality-assurance requirements for turbopump main-power gears are a combination of 100-percent inspection, sampling inspection, and certification as summarized below:

(1) All gears are subjected to the following checks:
- Dimensional inspection including surface texture
- Hardness check of case and exposed core
- Magnetic particle inspection for cracks and flaws
- Visual inspection (with magnification if required) for nicks, tool marks, and other surface flaws
- Surface etch to determine presence of surface tempering from grinding abuse

(2) Heat-treat specimens or sample gears are tested destructively for the following metallurgical properties:
- Case hardness
- Core hardness
- Chemical composition (may be certified by steel manufacturer)
- Case depth
- Microstructure (esp. grain size and banding)
- Retained austenite
- Hardenability (may be certified by steel manufacturer)
- Cleanliness

(3) Intrinsic qualities for which no tests exist are ensured by process control and certification of proper processing for
- Shot peening
- Hydrogen-embrittlement relief (if required)
- Material source
- Heat treatment (type of furnace, dew-point control, etc.)

Reference 2 (ch. 23) presents a discussion of gear inspection and the devices for gear inspection.

2.5.2 Performance Testing

Testing has been used extensively as a design and diagnostic tool in the development of turbopump power-gear systems. Several types of loading devices are used, including those that supply and absorb full power such as dynamometers; these devices are sophisticated, accurate, and expensive to procure and run. Other devices use loading methods that impose torque as a function of speed. One of the most useful test fixtures has been the back-to-back tester in which a fixed torque is locked by torsional windup of the shafts into a closed loop formed by the two gear trains (fig. 16). The tester prime mover need supply only friction losses; thus a gear load of several thousand horsepower can be simulated with a back-to-back tester driven by a 100-hp electric motor.

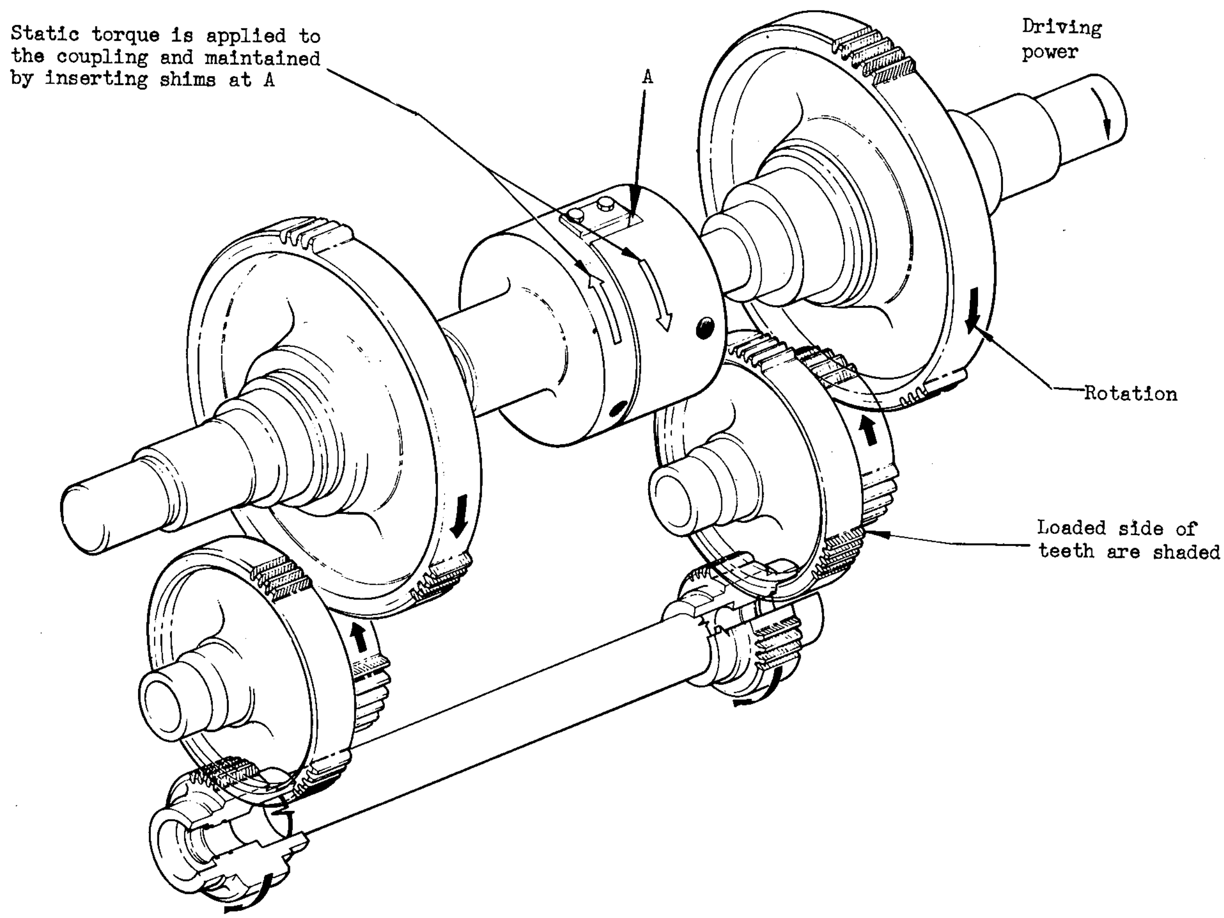


Figure 16. – Sketch of gear arrangement in a back-to-back gear tester.

The back-to-back gear tester (also used to run-in production gear case assemblies) has been used in the identification and ultimate correction of problems in profile modification, vibration, fatigue, tolerance limits, processing uniformity, and material adequacy. For example, lubrication problems arising from low ambient pressures in a gear system were simulated by evacuating the back-to-back gear enclosure to an absolute pressure of 2 mm Hg prior to starting the test. High-speed motion pictures were used to observe lubricant circulation and foaming behavior. Successful solutions to the difficulties were developed.

Gear operational data have been obtained during turbopump hot-fire and static engine tests and on rare occasions from telemetered vehicle flight data. Since at each higher assembly the difficulty and expense of gathering information increases, engine and flight tests are used to feed back gear data only for initial testing of a new system and for trouble shooting.

3. DESIGN CRITERIA and Recommended Practices

3.1 GEAR SYSTEM

The gear system shall satisfy the requirements for speed, direction of rotation, spacing, load capacity, and mounting rigidity.

The gear designer should review preliminary design layouts of the turbopump to ensure that the gear system requirements of power, speed, and life can be met by proper selection of gear type and size. The mechanical designer primarily responsible for overall turbopump design should make layouts showing the position, direction of rotation, and size of pumps, turbine, gears, and bearings. The effect on gear case design of the positions of ducts, manifolds, volutes, and accessories must be considered at this preliminary design stage.

The responsible project engineer should ensure cross-feed of information between the gear designer and other specialists so that necessary design adjustments are made with minimum time lag. Important decisions should be documented with timely distribution to those concerned in order to avoid wasted effort and to provide data for future reference.

Initial estimates of gear size may be modified as the intercomponent influences become evident and the gear detail-design calculations proceed. The design-configuration selection cycle may be shortened by a review of previously used or existing gear systems and by incorporation of design elements already shown to be successful.

As the gear-system configuration becomes firm, attention should be directed toward details such as the lubricant delivery system, accessibility of fasteners, and design of special tooling required for assembly. Tolerance stackups must be performed to ensure adequate axial clearance for quill shafts used in the power train and accessories.

3.1.1 Speed Ratio

The gear system shall satisfy the turbopump requirements for speed ratio.

Design the gear system to have a speed ratio that results in optimum turbine and pump speeds. Perform weight and efficiency studies to determine the optimum configuration considering hardware weight and propellant consumption improvements. For recommended practices in selecting system speed, consult reference 37.

Design accessory gear ratios to obtain the speed required for the specific accessory (e.g., the hydraulic pump or the electric generator). Observe Air Force/Navy Design (AND) Standard requirements for speed, and for speed tolerance if required.

The reduction ratio of each mesh should be kept below 5. The following table indicates the number of reductions recommended for overall reductions ratios:

Overall gear-train ratio	1 to 4	2 to 9	3 to 12
Number of reductions	1	2	Planetary

Utilize the following sequence to determine the proper gear diameters:

(1) Choose the minimum gear and pinion diameters that (a) achieve the desired speed change within the limits for maximum reduction ratio per mesh and (b) satisfy the spacing requirements of pumps and turbines.
(2) Determine whether the pinion diameter chosen will accommodate the required minimum number of teeth (sec. 3.2.2). Increase the pinion diameter if required.
(3) Check tooth strength by the methods given in section 3.1.7.
(4) Enlarge gears as necessary to achieve acceptable life and reliability.

3.1.2 Speed Capability

The gear system shall operate satisfactorily at the speed required in the application.

When possible, spur gears should be designed for pitchline velocity less than 20 000 ft/min. Higher speeds require special attention to measurable and inherent gear quality characteristics, lubricant capability, and details of tooth design in order to maintain reliability.

When PLV in a gear design exceeds 10 000 ft/min, special attention must be given to lubricant delivery. One or more of the following practices should be applied:

- Direct the lubricant stream to the disengaging side of the mesh. Ensure that the velocity (speed and direction) of the lubricant stream is sufficient to penetrate to the pitch line of the tooth as it passes the jet (refs. 20 and 21).
- Use baffles and deflectors to ensure delivery of lubricant to the teeth and to scavenge oil thrown from the teeth.
- In designs with PLV greater than 25 000 ft/min, use double helical gears with a face overlap of at least 2 in preference to spur gears.
- Minimize the sliding velocity between gear tooth surfaces, preferably to 60 ft/sec or less. Calculate sliding velocities by the method shown in reference 2 (ch. 14, sec. 14-1) or that in reference 10 (p. 55).

Sliding velocity can be reduced by any of the following practices:

- Use the smallest diametral pitch that will result in a gear with adequate root bending strength (sec. 3.1.7).
- Minimize pitchline velocity.
- Use a large pressure angle (25°).
- Use long addendum of driving pinion (sec. 3.2.5.3.3).

3.1.3 Gear Type

The gear type shall be suitable for the application.

Involute spur gears on coplanar parallel shafts are recommended for most main-power-train and accessory drives. Utilize speed-reducing gear trains rather than speed-increasing designs whenever possible.

Involute helical or bevel gears are suitable choices for accessory drives when the loads are light.

For applications that exceed the capacity of spur gears, double helical gears are recommended because they have higher load-carrying capacity, higher speed capability, balanced axial-load component, and smooth operation. The decision to use double helical gears should be coordinated with potential suppliers.

3.1.4 Gear Mounting

The gear mounting method shall provide accurate location and alignment of the gear.

Utilize straddle mounting in preference to overhung mounting. Base the selection of bearing and gear spans on deflection calculations that account for shaft bending, bearing deflections, casing deflections, and thermal effects. As a general rule, follow the minimum proportions shown in figure 3. Place an overhung gear as close to the nearest bearing as possible. Ensure that the bearing mounting surface can be machined properly by providing adequate tool clearance. A highly loaded gear should be made integral with its shaft (fig. 4).

3.1.5 Gear Attachment

Relative motion of the gear and shaft shall not cause excessive deflections or fretting.

Make the gear integral with the shaft when possible. Incorporate roller-bearing inner races on the gear shaft, if feasible, to eliminate separate races. The heat-treatment requirements of the raceways are the same as those for the gear teeth. See reference 38 for recommended raceway configurations. When the gear must be removable from the shaft, utilize two tight-fitting pilots straddling the gear web to locate the gear radially. Transfer driving torque by a loose-fitting involute spline centered under the gear web. Figure 5 presents the recommended configuration.

3.1.6 Backlash

Backlash shall not become negative.

Provide sufficient backlash by design to prevent tooth contact on the nondriving side of the teeth and to allow for a lubricating film. In selecting the minimum backlash, consider the effects of the following factors:

- Center distance changes caused by thermal contraction of the gear case
- Gear tooth bending deflections
- Bearing eccentricities, which reduce minimum theoretical gear center distances
- Gear eccentricity
- Gear tooth thickness tolerance
- Gear tooth spacing tolerance

Backlash may be provided by increasing center distance or by reducing tooth thickness. When a long-pinion addendum is used, it is recommended that backlash be obtained by thinning the gear tooth rather than the pinion tooth. Otherwise, part of the improved strength balance arising from the long addendum will be lost (ref. 2, ch. 5).

3.1.7 Load Capacity

Gear load capacity shall be adequate to transmit the design loads.

Calculate steady-state gear loads from torque and pitch radius as outlined in reference 2 (ch. 12). Preliminary designs of the gears may be roughed out by utilizing recommended values

of unit load for bending stress and K values for compressive stress. Verify that the gear design satisfies the criteria 3.1.7.1 through 3.1.7.3 set forth below.

3.1.7.1 TOOTH ROOT BENDING STRENGTH

The gear tooth root shall possess adequate bending strength to achieve the required service life under the applied loads.

Use the face load and unit load limits shown in table XII for preliminary design calculations and layouts for root bending strength.

Table XII. – Preliminary Design Limits for Face Load and Unit Load

Gear service	Material grade for carburized steel gears	Face load, ppi	Unit load, psi
Main-power train	3[a] or better	7 000	42 000
Main-power train	AGMA 2	2 500	25 000
Accessory drive	AGMA 1	1 500	12 400
Nonlubricated (propellant-cooled)			
Noncryogenic	AGMA 1 or 2	1 000	12 000
Cryogenic	AGMA 1 or 2	500	6 000

[a]This material grade is equivalent to that used in aerospace gearing but has not been adopted as an official designation by AGMA.

For reverse loading (gear loaded in both directions of rotation), reduce the allowable load capacity to 0.7 of the values shown in table XII. The preliminary design must be refined by applying modifying factors to account for dynamic loading (ref. 8).

Check the strength adequacy of the preliminary design selected on the basis of unit load by calculating the bending stress as follows:

(1) Lay out a large-scale (10X to 20X) tooth profile to select the tooth form factor Y that leads to the selection of the geometry factor J. A detailed procedure for finding Y for turbopump gears is given in reference 8; base Y on the load at the highest point of single tooth contact (fig. A-1, ref. 8).

(2) Use the form and geometry factors found in (1) above to calculate the root bending stress; then apply the formulas given in reference 8 for stress concentration and allowable stress modification for load, tooth size, and stress distribution. Table XIII gives values for dynamic factor K_v for various operating conditions. The factor K_v allows for tooth-geometry errors, speed, inertia and stiffness of rotating elements, load, and tooth stiffness.

Table XIII. – Dynamic Load Factor K_v for Various Operating Conditions

Operating condition	Dynamic load factor K_v	
	Range	Recommended value
Start	0.5 to 0.83	0.77
Turbine overshoot	0.5 to 0.77	0.77
Steady state	0.9 to 1.1	0.91
Transient	0.77 to 1.3	0.83
Shutdown	0.83 to 1.0	0.91

Table XIV presents values for overload factor K_o that account for the operational characteristics of the driving and driven elements.

Table XIV. – Overload Factor K_o Related to Pump and Turbine Type and Pumped Fluid

Turbine type	Overload factor K_o	
	Many-stage gas and light-liquid axial-flow pumps	Heavy-liquid centrifugal pumps
Many stages	1.00	1.25
Velocity-compounded, 1 or 2 stages	1.25	1.50

(3) Determine whether tooth bending strength is adequate for the intended life and gear quality by utilizing a plot of allowable root tensile stress vs number of cycles of stressing, as given in figure 6.

Modify the design as required to meet allowable root bending stress values.

Determine gear reliability by consulting figure 7, which gives the probability of tooth breakage during 1 million cycles as a function of root bending stress for four grades of gear material quality.

Consider providing 20-percent excess load capacity to allow for future uprating of the turbopump drive system.

Redesign the gear mountings or gear support if the load distribution factor K_m exceeds 1.5.

Specify shot peening of root fillets (sec. 3.4.5) as a means of improving tooth bending fatigue life if high reliability is required of highly loaded gears, or if fatigue life is less than that required.

3.1.7.2 TOOTH FACE COMPRESSIVE STRENGTH

The gear shall not be subject to surface pitting failure due to compressive stress.

For preliminary design purposes, estimate the compressive strength of gear teeth by using the tooth pitting index K given in equations (2a) and (2b). Recommended limiting values of K are listed in table XV.

Table XV. – Preliminary Design Values for Pitting Index K

		K value limits	
Gear type	Lubricant/Coolant	AGMA material grade 1 and 2	Material grade 3[a] or better
Main power	Oil or fuel-additive[b] Propellant	1000 200	2500 500
Accessory	Oil or fuel-additive[b] Propellant	600 200	1000 500

[a]This material grade is equivalent to that used in aerospace gearing but has not been adopted as an official designation by AGMA.
[b]RP-1 plus 2% Oronite 262.

For preliminary design estimates of compressive stress, use $S_c = 6500\ K^{1/2}$ for steel turbopump gears with 25° PA; use $7100\ K^{1/2}$ for gears with 20° PA.

For detail design, use the methods of reference 10 (p. 51) or of reference 11 to calculate compressive stress at the pitchline and at the pinion's lowest point of single tooth contact. For design evaluation, use the pitchline value; however, if the contact stress at the lowest point is more than 10 percent greater than the stress at the pitchline, then use the maximum value.

Redesign the gear if the calculated gear surface compressive stress exceeds the allowable values given in figure 8 or if the probability of failure (fig. 9) exceeds design requirements. Compressive stresses can be reduced by widening the face, increasing the gear pitch diameter, or modifying the addendum proportions (sec. 3.2.5).

When the allowable compressive stress is not known (as for a new material), consult reference 11 (table 5), or use the larger of the following values as an estimate of the allowable compressive stress:

$$S_{ac} = \frac{S_{tu}}{1.8}$$

or

$$S_{ac} = \frac{S_{ty}}{1.5}$$

where

S_{ac} = allowable compressive stress, psi

S_{tu} = ultimate tensile strength for lowest case hardness value, psi

S_{ty} = yield tensile strength for lowest case hardness value, psi

Initiate fatigue testing to establish values for S_{ac}.

Deep peening of gear contact surfaces to improve compressive strength is not recommended for turbopump gears, because limited testing has shown that this practice can cause an increase in scoring tendency.

Recommended case depths to ensure adequate compressive strength are shown in table XVI.

Table XVI. – Recommended Case Depth to Ensure Adequate Compressive Strength (20-Percent Stock Removal Allowance)

Pitch	Finished case depth, in.	Case prior to finishing, in.	Maximum grinding stock[a] (one side), in.
20	0.012 to 0.020	0.015 to 0.024	0.003 to 0.005
15	0.015 to 0.025	0.018 to 0.030	0.003 to 0.005
12 to 9	0.025 to 0.035	0.030 to 0.042	0.005 to 0.007
8	0.025 to 0.040	0.040 to 0.048	0.005 to 0.008
6	0.030 to 0.050	0.036 to 0.059	0.005 to 0.009

[a]Use lowest possible values.

The effective case depth for carburized or nitrided turbopump power gears should be twice the depth to the point of maximum subsurface shear stress. Subsurface shear stress can be calculated by methods devised by Buckingham (ref. 39, p. 529) or Dudley (ref. 10, p. 48). Effective case depth should be considered the depth at which the hardness is 50 R_c or is 10 R_c points lower than the outer surface case hardness, whichever condition is more demanding.

3.1.7.3 CHIPPING RESISTANCE

Tooth tips shall not chip because of excessive brittleness or stress concentrations.

Limit case depth at the tip to twice the case depth on the flank (fig. 17) or to one-half the addendum, whichever is greater.

The minimum tooth-tip width should be 0.25/P (ref. 2, ch. 5).

Provide smoothly blended tip, end, and edge radii (fig. 18) on the gear tooth in accordance with table XVII; X refers to end and edge radii, and Y refers to tip radii. X radii should be applied before carburization and should have surface roughness of 63 μin. AA max.; X and Y should be more generous for nitrided gears and should be applied before nitriding so that corner buildup is precluded. Avoid any discontinuities in radii application, particularly in tooth root areas.

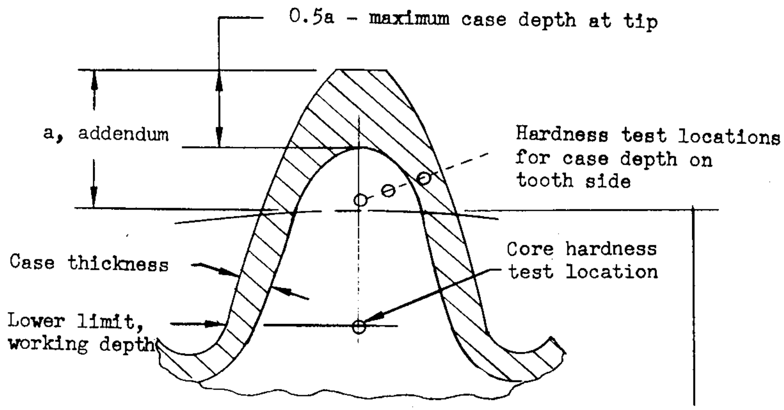


Figure 17. – Sketch showing locations for case depth and hardness tests.

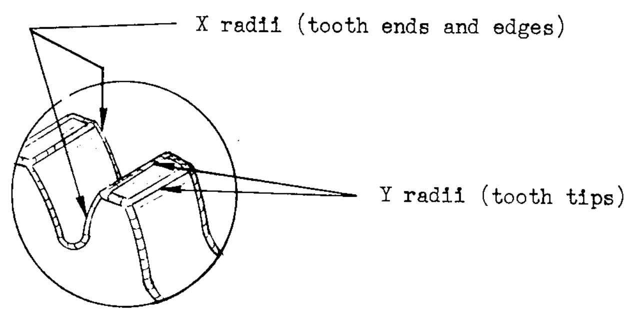


Figure 18. – Sketch illustrating gear-tooth tip, edge, and end radii.

Table XVII. – Recommended Tip, End, and Edge Radii for Gear Teeth

Hardening method	Pitch range	Radius, in.		Surface roughness of radii, μin. AA	
		X	Y	Main-power gears	Accessory drive gears
Carburizing	5 to 8	0.015 to 0.025	0.010 to 0.020	20	40
	10 to 12	0.010 to 0.020	0.005 to 0.010	20	40
	16 to 20	0.005 to 0.010	0.003 to 0.008	20	40
Nitriding	6 to 10	Max. possible $\leq$ 0.050	Max. possible $\leq$ 0.050	20	40
	12 to 20	Max. possible $\leq$ 0.030	Max. possible $\leq$ 0.030	20	40

3.1.8 Lubrication and Cooling

Deterioration of gear tooth surfaces by friction and wear shall not reduce gear life.

Lubricate and cool the gear tooth surfaces with solid, liquid, or gas coolants/lubricants. Recommended procedures and guidelines are set forth in sections 3.1.8.1 through 3.1.8.4.

For bearing lubrication practices, consult reference 38.

3.1.8.1 HEAT REMOVAL

The gear lubrication system shall maintain the gear system within a temperature range that will not result in degradation of material properties.

Provide sufficient lubricant flow to balance heat input from gear and bearing inefficiency as well as from external sources such as turbine heat soakback. Select the minimum flowrate of coolant that will maintain an equilibrium temperature within the capabilities of all materials involved. To achieve this objective, proceed as follows:

(1) Determine the rate at which heat is to be removed by summing the heat inputs from gears, bearings, and external sources. Assume a gear efficiency loss per mesh of 0.5 to 0.7 percent of the power transmitted in spur gear trains, or consult reference 2 (ch. 15) or reference 12 for detailed methods of estimating gear power loss. Add any heat input from external sources such as turbine gas leakage into the gear case or heat radiation to the gear case.

(2) Select the maximum lubricant temperature allowed (assume 210° F in the absence of specific requirements) and determine the maximum inlet temperature expected.

(3) Calculate the lubricant flowrate required by dividing the heat rate found in (1) above by the product of temperature difference found in (2) and the lubricant specific heat (assume it to be 0.42 Btu/lbm-° F in the absence of explicit data). Convert the value found into basic flowrate units and check the value found against the following estimates:

Lubricant type	Minimum flowrate, gal/min/hp/mesh
Oil	6.67×10^{-4}
Fuel-additive[a]	5.0×10^{-4}

[a]RP-1 plus 2% Oronite 262.

Ensure that the lubricant system supply is adequate to maintain the required flowrate for full run duration over the total environmental temperature range. Most systems will flow more at higher temperature because the fluid viscosity is lower.

Provide heaters if necessary to maintain more nearly constant flow by eliminating the higher viscosity existing at temperatures at the low end of the range. The quantity of lubricant circulated should be sufficient to ensure mission completion despite a single malfunction of an associated component (e.g., failure of a shaft seal).

3.1.8.2 SCORING PREVENTION

The contacting tooth surfaces shall not experience destructive scoring.

Maintain a scoring index within the ranges shown in table XVIII. The scoring indexes are presented in descending order of preference. PVT should be used only as a measure of scoring risk and not as a design limit.

Table XVIII. – Recommended Limits on Three Types of Scoring Index

Method	Scoring index	Use	Recommended value
Bodensieck (ref. 13)	$\lambda = h/s$[a]	Diester oils Fuel-additive[b]	1.5 min. 1.3 min.
AGMA flash temperature[c] (ref. 14)	Flash temperature index, °F	MIL-L-7808 oil Fuel-additive[b]	300 max. 350 max.
PVT calculated at tooth tips (refs. 1 or 3)	Contact pressure (psi) x sliding velocity (ft/sec) x distance from pitch line to point where contact pressure is calculated (in.)	For all turbopump gears	3.0×10^6 max.

[a] h = lubricant film thickness
s = surface roughness

[b] RP-1 + 2% Oronite 262.

[c] Calculate using surface finish factor of 55/(55-s) instead of 50/(50-s) as shown in reference 14, where s = surface roughness, μin. AA.

When scoring risk is considered too high, any of the following steps are recommended as a means to reduce scoring:

- Reduce lubricant inlet temperature.
- Increase lubricant flow (sec. 3.1.8.1).
- Use high lubricant pressures (60 to 600 psi).
- Redesign gears to obtain larger radii of curvature of teeth.
- Raise scoring-preventing properties of lubricant by using EP (extreme pressure) additives or change lubricant; see table VI. Use only extensively tested EP additives; these chemically active compounds sometimes become corrosive during storage, and this effect is not predictable.
- Reduce overload factors K_o by reducing torque peaks.
- Modify profile and lead to compensate for elastic deflections.
- Increase number of gear reductions (2 or more).
- Increase face width.
- Use finer pitch.
- Redesign the gear system to lower the loads.
- Lower the pitchline velocity.
- Lower the sliding velocity (sec. 3.1.2).

- Refine the surface texture but do not reduce the minimum roughness below 6μin. AA.
- Increase tooth surface hardness (60 R_c minimum).
- Improve alignment if required.
- Select lubricant with higher viscosity.
- Redesign gears to have lower contact stress.
- Use contact ratios larger than 1.4

3.1.8.3 LUBRICANT PROPERTIES

The lubricant shall possess adequate load-carrying capacity, acceptable chemical stability, and acceptable viscosity over the range of environmental temperatures of the application.

Determine lubricant load-carrying capacity by testing or, as rough guidelines, use the values shown in table VI. Ryder gear test results should be used to compare the relative capabilities of lubricants. Load values achievable with a new gear design must be established by test under operating conditions. Although in most cases oxidizer cannot be allowed to mix with lubricant, in selecting the lubricant consider its compatibility with turbine gas, fuel, or oxidizer with a view toward avoiding potential sludging that may occur if leakage enters the gear case. Alternatively, incorporate replaceable filter elements and remove accumulated sludge on a regular service schedule.

Avoid lubricants with excessively high viscosity at the lowest operating temperatures. A fuel-additive mixture or MIL-L-7808 synthetic-base lubricant is recommended when temperatures down to -30° F are involved. Avoid MIL-L-6086 petroleum-base lubricant for service at temperatures below 0° F.

The use of a fuel-additive mixture as the gear-system lubricant is recommended for all engines using RP-1. The recommended additive is Oronite 262, a zinc dialkyl dithiophosphate compound, which should be mixed with the fuel at a concentration of 2 to 3 percent by volume. Exercise caution to prevent moisture contamination of this compound.

Individual production batches of MIL-L-25336 synthetic-base oil should be retested at 6-month intervals for retention of specified load-carrying capacity, because the EP properties of this lubricant have been shown to be unstable. Use of this lubricant for a new design is not recommended.

Select lubricants with demonstrated foaming resistance. MIL-L-7808 oils and the fuel-additive mixture are recommended for resistance to foaming at low ambient pressure (less than 2 psia). MIL-L-6086 oil foams extensively and should be avoided for service at low ambient pressure.

With the exception of RP-1, propellants in general should not be used as gear coolants. Consider propellant cooling for applications with short life requirements, loads below 500 ppi, and speeds below 10 000 ft/min PLV. Testing should precede inclusion of propellant lubrication in final design.

Use solid-film lubricants where loads, speeds, and intended life permit. For information on solid-film lubricants, consult reference 40 and section 2.1.8.3

3.1.8.4 LUBRICANT DELIVERY SYSTEM

The lubricant delivery system shall cool and lubricate adequately and reliably and shall weigh as little as possible.

Splashed-oil and grease-packed lubrication systems should be used only for gear systems of less than 100 hp, with pitchline velocity less than 10 000 ft/min.

Once-through (overboard drain) flow should be used for one-duty-cycle, short-duration applications with oil or fuel additive as the lubricant. Provide 10-percent excess capacity for such systems; base the calculated reserve on maximum flow and duty-cycle duration.

Use positive-displacement pumps for feeding lubrication systems that must operate over a wide temperature range, because these pumps will maintain a more constant flowrate over a range of temperature than will a constant-pressure pressurized system. Ensure that the lubrication system is capable of withstanding the pressures resulting from low-temperature operation.

Recirculating lubricant flow systems are recommended when the total weight of tanks, lubricant, feed and scavenger pumps, and heat exchanger is lower than the total weight of tanks, lubricant, feed pump or pressurant, and controls for a once-through system. The lubricant reservoir for the recirculating system should have a capacity of 100 percent (preferred) or 60 percent (minimum) of the volume flowing in 1 minute.

3.1.8.4.1 Lubricant Spray Nozzles

Lubricant delivery nozzles shall deliver an adequate quantity of lubricant to the gear tooth surfaces.

Place lubricant delivery nozzles on the disengaging side of the mesh, with the stream directed radially inward toward the center of each gear.

The lubricant supply pressure should be sufficient to result in a jet velocity that will penetrate to the pitchline of gear and pinion. Methods of calculating the required velocity

are presented in reference 21; a supply pressure of 400 to 600 psi is recommended for gears with PLV of 20 000 ft/min or greater.

The nozzle spray should cover the entire face of both pinion and gear; a rule of thumb is to provide one nozzle for every 1/2 in. of gear face width.

The minimum nozzle outlet orifice should be 0.015 in. in diameter to reduce clogging tendency.

3.1.8.4.2 Foaming of Lubricants

Foaming of lubricant shall not degrade cooling.

Excessive foaming of lubricant, which sometimes occurs at gear case pressures below 2 psia, should be prevented by selecting low-foaming lubricants, by pressurizing the gear case to 5 psia minimum with gaseous nitrogen or gaseous helium, by using baffles, or by adding antifoaming additives to the lubricant.

3.1.9 Gear Case

The gear system envelope (gear case) shall provide rigid mounting with good alignment for gears and bearings.

Make overall turbopump layouts to determine whether pump and turbine spacing is compatible with gear center distance.

Isolate the gear case from external duct loads and thermal loads where practical.

To ensure adequate gear-case structural strength and rigidity, perform detailed stress and deflection analyses considering all internal and external forces and temperature-induced deflections. When possible, utilize the results of measurements taken during actual operation of similar gear cases. Internal forces to be considered include bearing loads arising from gear tangential forces, separating forces, and axial loads from helical gears. Methods of calculating gear reactions are given in reference 2 (ch. 12). Gear-case internal pressure also must be considered and kept as low as feasible (5 to 40 psig). Pump and turbine impeller axial and radial loads must be added when they are supported on the gear-case structure. External loads from ducts should be eliminated when possible, as they must be included in the stress deflection analyses if the ducts are supported by the gear-case structure. Whenever possible, the gear case should be made as symmetrical as possible to reduce possible distortion by thermal or mechanical loads.

Provide clearance for ducts, manifolds, and volutes. Consider duct deflections due to temperatures and pressures.

Fasteners and bolts should be made accessible, so that a minimum number of special tools are required for assembly and disassembly of the gear box, turbopump assembly, and turbopump installation in the engine system. Avoid fasteners inside the gear case.

Test for gear-case strength using brittle lacquer (stress-coat) applied to gear case outside surfaces. Crack pattern during application of full load will reveal the magnitude and direction of gear-case strains and will indicate the required design modification.

Obtain accurate relocation of gear-case components by use of locating pins or dowels with line-to-line or slight press fit. Locate bearing bores by line boring the assembled gear case with locating pins installed.

Provide sufficient clamping force with screws or bolts to prevent relative movement of gear-case halves under loads. Clamp the gear-case components together with a metal-to-metal joint. Use O-rings or other recessed seals (not gaskets) to prevent leakage from parting planes whenever possible.

Anticipate the effects of malfunctions of associated components (e.g., overpressurization due to failure of seals) so that some redundancy or margin of safety is designed into the gear case.

3.2 GEAR DETAIL

3.2.1 Pressure Angle

The pressure angle of the gear shall not detract from the tooth load capacity.

A 25° pressure angle is recommended in preference to lower values for turbopump power gears, particularly if long-addendum pinion teeth are used (see sec. 3.2.5.3).

3.2.2 Number of Teeth

The number of teeth on a gear shall satisfy the gear speed ratio, yet not result in operational or manufacturing difficulties.

Choose the number of teeth required to achieve the specified ratio within the following constraints:

(1) Select gear pitch that provides adequate strength; if necessary, increase tooth bending strength by using coarser pitch and therefore fewer teeth.
(2) Avoid undercutting pinion teeth by adjusting pinion pitch and pitch diameter so that the number of teeth does not fall below the minimum given in the following chart:

Pressure angle, deg	Power gears, minimum teeth	Accessory gears, minimum teeth
20	26	22
22.5	23	19
25	20	16

(3) Establish the number of teeth so that tooth meshing frequency does not coincide with the natural frequencies of the gear system or any of its elements.
(4) Ensure hunting-tooth action and maximum wear life by choosing numbers of teeth in meshing gears so that there are no common factors.
(5) Limit the maximum number of teeth to 100; otherwise, manufacturing and quality-control costs are likely to be excessive.

3.2.3 Contact Ratio

The transfer of tooth loads to successive teeth shall not produce excessive dynamic loads.

The contact ratio, calculated with the formula given in reference 10 (p. 55), should be kept at a value greater than 1.2. Use 1.5 when possible. Avoid stub-tooth designs for highly loaded gears because of the inherently low contact ratio.

3.2.4 Face Width

The width of the gear face shall not result in uneven load distribution across the face.

Keep the ratio of face width to pitch diameter (F/D) less than 1:1 for spur gears. On double helical gears, the total width for both helicals should not exceed twice the pitch diameter.

The following effective values for F/D are recommended for turbopump pinions:

Gear type	Effective F/D values	
	Preferred maximum	Limit
Spur	0.5	0.7
Helical	0.60	0.9
Double helical	1.1	2.0

3.2.5 Tooth Proportions

Tooth proportions shall ensure maximum strength and smooth tooth action.

Use the gear tooth proportion system listed in reference 25 for coarse pitch ($P_d \leqq 19.99$) and in reference 26 for fine pitch ($P_d \geqq 20$) involute gears. Include profile modifications (sec. 3.2.5.5) as required to obtain the maximum load capacities.

3.2.5.1 WHOLE DEPTH

The whole depth shall not sacrifice strength, make manufacturing difficult, or prevent use of adequate fillet radii.

Use full-depth standard tooth form proportions as listed in reference 25 whenever possible. Accessory-drive gear designs should follow the standard proportions. For gear designs where special design goals exist, use the following whole-depth values:

Gear type	Whole depth
Main-power-train gears with adjusted addendum and dedendum	$\frac{2.35}{P_d}$ to $\frac{2.40}{P_d}$
Moderately loaded accessory gears	$\frac{2.25}{P_d}$ to $\frac{2.35}{P_d}$
Lightly loaded accessory gears	$\frac{2.25}{P_d}$ (standard)

Stub teeth (those in which the theoretical working depth is less than $2.00/P_d$) should not be used for accessory and power gears, because the low contact ratios of stub teeth result in sharp transfer of loads. Stub teeth can be used to advantage in internal-gear design to give additional tooth tip clearance.

3.2.5.2 TOOTH THICKNESS

Tooth thickness values shall not reduce tooth strength.

Obtain backlash by thinning the gear tooth rather than the pinion tooth. Consult reference 2 (ch. 5, p. 5-20) for design procedures to be used in selecting tooth thickness.

3.2.5.3 ADDENDA

3.2.5.3.1 Undercutting

Addendum proportions shall not cause undercutting of teeth.

Use the required pinion addendum to avoid tooth undercut. Proper values for spur and helical gears are shown in figure 1 of reference 3.

3.2.5.3.2 Equal Strength

Addendum proportions shall result in equal bending strength.

Use values of addendum shown in figure 11 for spur gears with 20° pressure angle to obtain equal-strength teeth (balanced Y factor) in pinion and gear. Figure 11 also can be used for gears with 25° PA if the same whole depth is used.

In general, increase the addendum of the driving pinion and equally decrease the addendum of the driven gear. For gears different from those for which figure 11 is valid, layouts and tests must be made to generate similar charts. Calculation methods are given in references 8 (App. A) and 25 (information sheet B).

3.2.5.3.3 Equal Sliding Velocities

Addendum proportions shall result in minimum sliding velocities.

To obtain equal sliding velocities above and below the pitch line, choose addenda shown in figure 2 of reference 3 (for 20° PA) or figure 3 of reference 3 (for 25° PA).

In summary, establish the design values of addenda on the basis of the following factors, which are listed in descending order of importance:

(1) No pinion undercutting
(2) Root strength maximized by equalizing pinion and gear strength
(3) Sliding velocities minimized by adjusting radii of curvature
(4) Peak compressive stress minimized.

When untried addenda proportions are used, confirm design adequacy by gear tests.

3.2.5.4 ROOT FILLET

Gear-tooth-root fillet radii shall not result in excessive stress concentrations.

Root fillet radii should be maximized to minimize root bending stress. Recommended values for fillet radii for 25° PA aerospace gears are shown in figure 19. Limiting radius values may be calculated as shown in reference 25 (information sheet B).

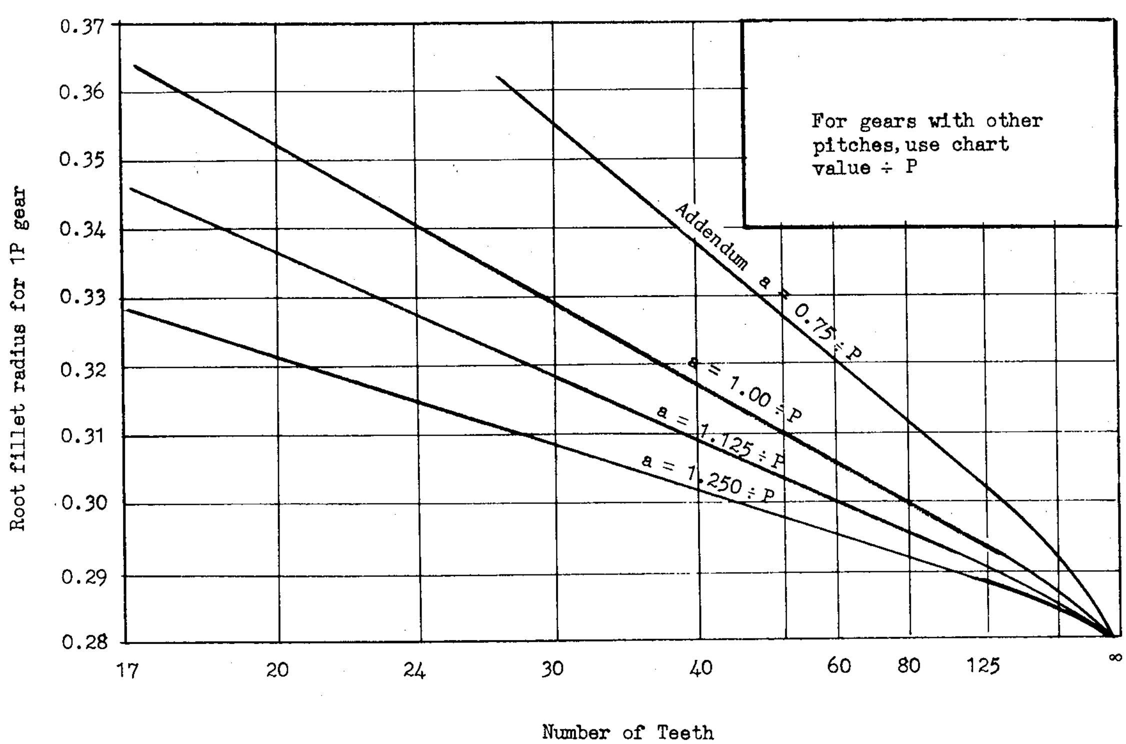


Figure 19. – Recommended root fillet radii vs number of teeth (25° PA spur gears).

A large-scale (20X minimum) layout of the gear tooth profile should be made to ensure that the root fillet radii will not interfere with the mating gear tooth tip.

The gear cutting and grinding tool manufacturer should review the chosen radii to ensure the feasibility of cutter design limits.

3.2.5.5 TOOTH-FORM MODIFICATION

3.2.5.5.1 Involute Modification for Load

Tooth bending of highly loaded gears shall not adversely affect tooth action.

Modify the involute profile of all gears operating at face loads of 2000 ppi and above.

Estimate the required modification by assuming a tip bending deflection of 0.00035 in. per 1000 ppi face load for steel gears. When this estimated deflection exceeds 0.0005 in., consider "barrelled" profile modification (fig. 13(a)) of tip and root. Tip-only modification may be used on lightly loaded power and accessory drive gears. Employ the methods of reference 27 to calculate in detail the required modification.

3.2.5.5.2 Tip Relief for Speed Effects

The gear tooth tips shall not interfere at start of mesh.

To allow for manufacturing errors, provide a tip relief of 0.0005 in. for the first 25 percent of active profile for gears that will operate at speeds exceeding 15 000 ft/min PLV. The profile modification can drop to zero at the 25 percent of active profile depth point if speed is the only factor requiring profile modification. Recalculate contact ratio of gears after tip relief to ensure that the contact ratio does not drop below 1.2.

3.2.5.5.3 Lead Modification

Gear tooth deflection or misalignment shall not result in excessive stress concentration.

Crown the teeth of gears that may be subjected to misalignment. Crowning of 0.0002 to 0.0005 in. is recommended for highly loaded gears. See figure 13(b) for an example of a lead chart for a properly crowned gear tooth.

Gears supported in rigid gear cases, while not requiring crowning, should be provided with end easing as shown in figure 13(c) to prevent stress concentrations at the ends of the teeth.

For gears with F/D values greater than 0.5 or with face widths over 1 in., consider helix correction to compensate for torsional windup of the gear tooth. A preliminary estimate of the required helix correction is 0.0002 in./in. of face width.

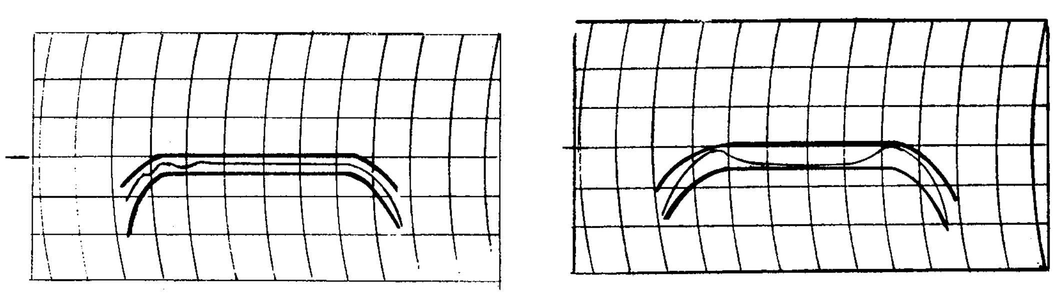
Lead modification values recommended herein should be used as preliminary design values in the absence of testing experience. For a more detailed calculation of theoretically required modifications, consult references 41 or 42.

Normally, lead corrections should be limited to 0.0006 in. maximum to avoid contact stress increase and reduction in load capacity. Test for lead modification required to compensate for misalignment by coating the teeth with flash copperplate, operating the gear mesh briefly under reduced load, and studying the contact pattern created. Repeat this test after applying profile modification to confirm its adequacy. This procedure is especially important for bevel gears; however, to determine proper lead modification for the torsional windup, the gears must be run under full load. The effects of torsional windup may be reduced by driving through splines axially located at the center of the gear face (e.g., the pinion shown in fig. 4).

3.2.5.6 SURFACE TOLERANCES

The shape of the active involute or lead surface shall not result in high local contact stress.

The rate of reversal should not exceed 0.0002 in. for any 25-percent portion of the active involute or lead profiles for 8- to 12-pitch power gears. In addition, the overall lead pattern must not be concave (fig. 20). A dimension of 0.0003 in. per 25 percent of the active profile is acceptable for accessory drive and lightly loaded power gears. Determine the rate of reversal by examining the traces on involute and lead charts.



(a) Acceptable (b) Unacceptable (concave)

Figure 20. – Acceptable and unacceptable lead traces.

3.2.5.7 SURFACE TEXTURE

3.2.5.7.1 Contacting Surfaces

The surface texture of tooth contact surfaces shall not adversely affect scoring resistance and fatigue life.

Grinding is recommended as the final finishing method for tooth contacting profiles.

Measure surface texture in the directions illustrated in figure 12. Peak-to-valley waviness should not exceed 50 μin. Recommended values for surface roughness are shown in table XIX. Do not strive for surface roughness values less than 6 μin. AA, because such surfaces not only are more expensive to manufacture but actually appear to have greater tendency to score than surfaces with roughness values between 6 μin. AA and 20 μin. AA.

Table XIX. – Recommended Values for Gear Surface Roughness

Surface	Surface roughness, μin. AA		Preferred machining operation	Importance of surface condition
Contacting surfaces	min.	max.	Grinding	Very critical
Power gears, PLV < 25 000 ft/min	6	20		
Power gears, PLV > 25 000 ft/min	6	16		
Accessory drive gears	6	40		
Noncontacting surfaces		max.		
Roots and Fillets:				
Before peening		64	(1) Hobbing	Very critical
After peening		100	(2) Shaping (3) Grinding (do not grind after case hardening)	
Mounting surfaces		32	Grinding, turning	Critical
Rims and webs		125	Crush grinding or turning	Important
Hubs		125	Crush grinding or turning	Important
Lands and tips		125		Minimal

A light vapor blast or peening of the gear tooth contacting surface with small-size shot or glass beads is recommended for improving borderline lubrication performance. Surface texture requirements for the finished gear after blasting should be determined by a test program that compares performances of gears with different surface textures.

3.2.5.7.2 Noncontacting Surfaces

Noncontacting gear surfaces shall not contain harmful stress concentrations or residual stresses, yet shall not be excessively expensive to manufacture.

Table XIX presents recommended surface roughness values and production finishing methods for nonactive gear surfaces.

Grinding after carburizing in the root area must not be permitted because this process will produce undesirable residual tensile stresses.

3.2.6 Rim and Web

3.2.6.1 RIM AND WEB PROPORTIONS

The rims and webs on power gears shall carry the required load and shall be of minimum weight and lowest cost.

Web thickness should never be less than 0.10 in. When face widths are between 0.10 and 0.20 in., make the webs the same size as the face and utilize lightening holes to reduce weight. Recommended lightening-hole proportions for lightly loaded power gears are shown in figure 21.

Recommended rim and web proportions derived from experience (up to 25 000 ft/min PLV) are shown in table XX; dimensions refer to figure 14 and are presented as a decimal fraction of the tooth whole depth h_t.

Avoid bolted rims and webs, because joint deflection causes undesirable gear meshing inaccuracy and fretting at the bolted joint. If balancing of gears is required, provide sufficient stock for weight removal as shown in figure 22.

Specify shot peening of gear rims and webs on main-power gears or other gears with high tensile web stress, or on gears that are subject to web fatigue failures.

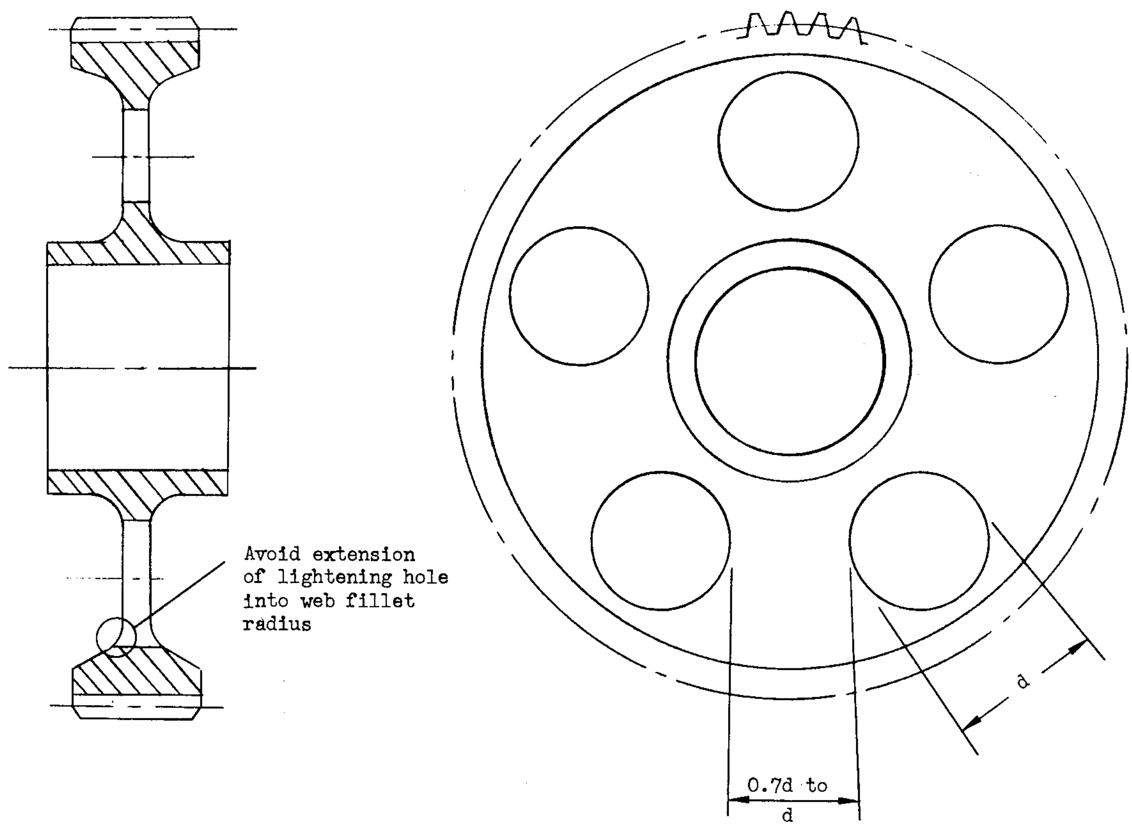
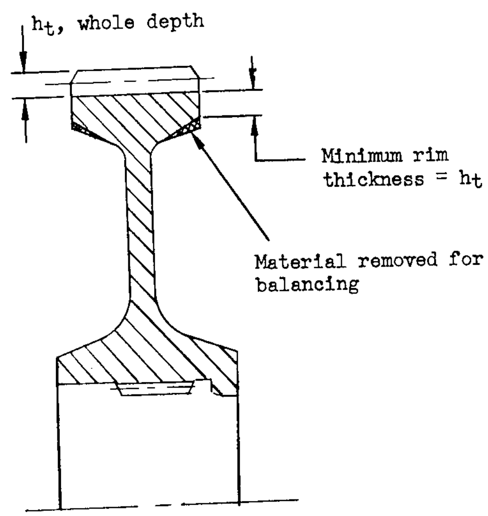


Figure 21. – Sketches illustrating recommendations for design of lightening holes.

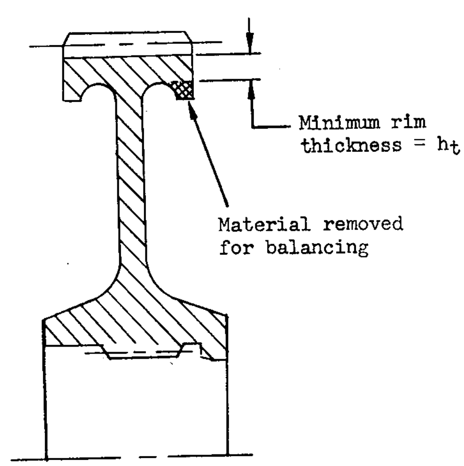
Table XX. – Recommended Rim and Web Thicknesses and Number of Web-Lightening Holes

Item	Main power	Cryogenic and moderate power accessory	Light power gears
Rim thickness*	1	1	0.7
Web thickness*	1	0.7	0.5
Number of lightening holes	None	Odd numbers 5, 7, 9, etc.	Odd numbers 5, 7, 9 or even numbers > 6

*Thickness in inches = factor shown times tooth whole depth.



(a) Typical rim configuration



(b) Alternate rim configuration

Figure 22. – Recommended locations for stock removal for balancing.

3.2.6.2 GEAR RESONANCE

The gear shall not have torsional or lateral resonances in the gear operating range.

The frequency of gear tooth meshings should not correspond to a resonant frequency of any part of the system. Hunting-tooth action is desirable to avoid reinforcing vibrations (sec. 3.2.2). The number of lightening holes in webs of mating gears should not contain common factors.

Determine the natural frequency of any new gear design by calculation or test. The modal shapes and resonant frequencies of rimmed gears can be calculated as outlined in reference 23. These calculated frequencies are reasonably accurate; however, testing is recommended for more precise determination. Shake tables and sand-pattern methods are adequate for low frequencies, but fine-grained table salt (popcorn salt) and piezoelectric sensors (accelerometers) are required for determining gear meshing frequencies.

Campbell diagrams (interference diagrams) should be constructed for each gear to establish forcing frequencies and resonant modes. Frequencies investigated should range from a few cycles per second to 150 percent of the maximum operating mesh frequency.

Avoid designing the gear and shaft system such that its torsional natural frequency is in the expected operating speed range. Resonances may be detected by accelerometers attached to the gear case. Corrective action should be initiated if destructive amplitudes are discovered.

Provide for balancing of the gears or shaft if the calculated imbalance forces are detrimental to bearings or other components. If possible, plan to include balancing provisions on other components rather than on the gears.

Use helical gears in preference to spur gears when vibrational problems are anticipated. Dampers are recommended for reduction of vibrational response in gear rims (ref. 21).

3.2.7 Tolerances

Tolerances specified for gears shall be precise enough to achieve good load capacity and reliability, yet shall not exceed the manufacturer's capabilities.

Use the coarsest tolerances that will satisfy the capacity and life requirements of the application. For critical power gears, use AGMA quality levels 9 to 13 or the values given in table VII. Finer tolerances will require development on the part of the gear manufacturer that the designer must coordinate and evaluate by actual gear test.

3.3 MATERIALS

3.3.1 Gears

The properties of the gear material shall satisfy design requirements.

Material recommendations are summarized in table XXI, and specific material considerations are presented in sections 3.3.1.1 and 3.3.1.2. Ensure consistent material properties by invoking existing standards, or establish new specifications and quality control tests as necessary to achieve consistency.

Materials used in critical turbopump gears should be purchased in mill lots to ensure uniformity over a production run of gears.

It is recommended that no forging or machining work be performed on a main-power gear until it is established that the material will meet the cleanliness, hardenability, and carbon content requirements for the gear.

Recommended power-gear materials are not corrosion-resistant; therefore, corrosion protection must be provided by one of the following means:

(1) Cover gear surfaces during idle periods with preservative compounds such as MIL-C-16173 for systems lubricated with oil and fuel-additive.
(2) Maintain continuous inert-gas protective atmosphere for gears throughout idle periods with gaseous nitrogen per MIL-P-27401 or gaseous helium per MIL-P-27407.
(3) Plate surfaces with chromium per QQ-C-320 or electroless nickel per MIL-C-26074. Chromium plate is preferred for areas such as locating surfaces, which require an accurate, hard surface. The plating thickness should be 0.002 in. oversize, ground back to the required size. Chromium plating of active gear tooth surfaces is not recommended. For the nickel plating, recommended thicknesses are 0.0003 to 0.0005 in. on gears enclosed in cases, or 0.0009 to 0.0011 in. on surfaces exposed to the atmosphere.

Dry-film lubricants (sec. 2.1.8.3 and ref. 40) can offer a low-cost means of corrosion protection.

Lubrication of power gears with propellants alone is not recommended; if however, such use is contemplated, consult section 2.3.1 for materials that have been used. Testing of materials should be an integral part of any gear design project involving propellant-lubricated gears.

Table XXI. – Recommended Materials and Material Requirements for Turbopump Gears

Item	Recommendation			
Material quality grade	AGMA 1	AGMA 2	3[a]	4[a]
Intended use	Accessories	Medium power	Main-power train	Very critical gears
Material (alloy designation)	AISI 9310 9315 3310 8620 4620 4340	AMS 6265 AMS 6260 AISI 9310 9315 8620 4620	AMS 6265[b] 6260[b] AISI 9310[b]	AMS 6265[b] AISI 9310[b]
Carbon content, percent Case Core	 Per material specification Per material specification	 0.75 to 1.00 Per material specification	 0.75 to 0.95 0.10 to 0.13	 0.8 to 0.95 0.11 to 0.14
Hardness, Rockwell "C" scale Case Core Size effect considered	 58 minimum 31 to 44 No	 58 to 63 32 to 42 Optional	 60 to 63 34 to 42 Yes	 61 to 64 38 to 42 Yes
Material purchased in registered mill lots	No	Rarely	Yes	Yes
Cleanliness requirements specified	No	Optional	Yes	Yes – special requirements
Case depth control	Loose	Adequate to ensure uniform case	Adequate to ensure thick, uniform case	Adequate to ensure thick, uniform case
Grain size (per ASTM E 112-63)	3 to 5	5 minimum	6 minimum	7 minimum
Carbide network	Small continuous networks allowable	Networks required to be noncontinuous	No significant networks allowable	No significant networks allowable
Retained austenite, percent maximum (a) Determined by visual examination under magnification (b) Determined by X-ray diffraction	 15 25	 10 15	 5 15	 3 13

[a]As noted, not currently an AGMA-designated grade.
[b]Special hardenability and procurement requirements.

3.3.1.1 MATERIAL GRADES

The quality of gear materials shall meet the specified design requirements.

Gear materials must be graded to achieve the performance required for turbopump service. The general requirements for the grades are listed in table XXI. Grades 1 and 2 are presently recognized by AGMA (ref. 3) for aircraft quality gears. A new designation, grade 3, is recommended for most turbopump mainpower gears; for even more critical applications, a potential grade 4 is listed. Uprating and increasing performance requirements require an upgrading of the gear quality, but cost and schedule considerations require that the designer specify the lowest grade that will meet the requirements of the design.

AMS 6265, a vacuum-melted AISI 9310 carburizing steel, is recommended for most aerospace gearing. Allowable bending and compressive stress levels are given in section 2.1.7. H-bands should be specified for critical gears to control the material hardenability. For the same reason, the carbon content should be specified to tighter limits than standard (table XXI). Reference 43 (pp. 189-216) presents the important aspects of hardenability of steels.

3.3.1.2 METALLURGICAL PROPERTIES

The metallurgical properties of the materials shall meet the requirements of the design.

Specify the hardness requirements on the gear drawings; specify the hardenability requirements on the gear forging drawing. The recommended case and core hardness ranges for carburized steels in the various grades are shown in table XXII; the higher values of case

Table XXII. – Recommended Hardness Values for Carburized Gears

Use	Hardness			
	AGMA grade 1	AGMA grade 2	Grade 3[a]	Grade 4[a]
Case (Rockwell 15N superficial hardness)	89.2 min.	89.2 to 91.5	90.2 to 91.5	90.5 to 91.7
Case[b] (Rockwell C)	58 min.	58 to 63	60 to 63	61 to 64
Core (Rockwell C)				
D = 0 to 6 in.	31 to 44	32 to 42	36 to 42	38 to 42
D = 1 to 12 in.	31 to 44	32 to 42	34 to 42	36 to 42

[a]As noted, not currently an AGMA-designated grade.
[b]Measure with 15N scale on profile of gears to be placed in service.

hardness shown in the table are more desirable than values at the low end of the acceptable band. It is recommended that, when contact stress is critical, the acceptable band of hardness be narrowed by raising the lower limit to the value of the next higher grade. Grade 3 is recommended for most turbopump power gears; detailed recommendations for hardness of Grade 3 gears is shown in table XXIII.

Table XXIII. – Recommended Hardness for Material Grade 3[a] Gears (Carburized)

		Case	
D, in.	Core R_c	Equivalent R_c	15N
0 to 4	36 to 42	60 min.	90 min.
4 to 8	34 to 42	60 min.	90 min.
8 to 12	32 to 42	60 min.	90 min.

[a]As noted, not currently an AGMA-designated grade.

No continuous carbide networks may be allowed, although carbide networks that are fine and discontinuous appear to be beneficial to the performance of turbopump gears. A metallographic specimen magnified 500 times is recommended for evaluation of carbides. Carbide particle sizes up to 0.0003 in. are acceptable. The case structure must not have an excessive amount of retained austenite, but total absence of retained austenite could be undesirable as the gear material may become too brittle. Turbopump gears should have 5 to 15 percent retained austenite (table XXI).

3.3.2 Gear Case

Gear-case materials shall be chemically compatible with the coolants and lubricants used and structurally suitable for supporting the loads.

Cast aluminum alloys A356-T61, A357-T61, and Tens-50 are recommended for most applications. Cast corrosion-resistant steels should be used when the gear case may contact corrosive materials. The specific alloy must be compatible with the propellant. Because of the highly reactive chemical nature of magnesium alloys, these alloys should not be used in contact with oxidizers.

Table XXIV. – Recommended Processes and Process Controls for Fabricating Turbopump Gears

Process or Control	Recommendation			
	Accessories	Medium power	Main-power train	Very critical gears
Forging	Use of forging optional	Open or closed die forging required	Testing and qualification of die and forging source required	Forgings with integral teeth recommended qualification of die and forging source required
Cutter control	No	Only on critical gears	Yes	Yes – cutters are registered
Finishing method for profile surfaces	Grind, hone, lap or shave	Grind required on critical gears	Grind required	Grind required (new methods may be required)
Root treatment after carburization	Grinding allowed	Grinding not recommended, but allowed	No grinding permitted	No grinding permitted
Grinding burn acceptability	Light burns accepted	None accepted	None accepted	None accepted
Detection method	Visual	Nital etch	Mild nital etch	Mild nital etch[a]
Stock removal control	Small percent of run inspected	Large percent of run inspected	100% of run inspected	100% of run inspected
Fillet radii control	Small percent of run inspected	Large percent of run inspected	100% of run inspected	100% of run inspected
Shot peening				
Roots	Special cases only	1X intensity	2X to 4X	4X minimum
Rims	No	Special cases only	2X to 4X	4X minimum
Webs	No	No	2X to 4X	4X minimum
Heat-treatment furnace				
Type	Pit, brick-lined, or better	Brick-lined or vacuum retort	Stainless teeel or vacuum retort	Vacuum retort only
Processing control	Dew point or carburizing pack	Dew point or infrared gas analyzer	Prefer infrared gas analyzer	Infrared gas analyzer only
Magnetic particle inspection	Yes	Yes	Yes	Yes
Analysis of type 1[b] heat-treat sample for carbon content	Optional	Optional	Yes	Yes
Inspection method for carbide network determination	Random sample visual examination of microstructure under magnification; type 1 or optional type 2[c] sample	Examination at 500X of microstructure of type 2 heat-treat sample	Examination at 500X of heat-treat sample type 2 or, when required, type 3[d]	Examination at 1000X of heat-treat sample type 3
Retained austenite determination	Random check by microstructure examination or X-ray diffraction	Microstructure examination or X-ray diffraction on large percent of run	Determination of level on 100% of gears by microstructure examination or X-ray diffraction	Determination of level on 100% of gears by microstructure examination and X-ray diffraction
Inspection of all gear dimensions	Sampling	Required on large percent of gears	Required on 100% of gears[e]	Required on 100% of gears[e]

[a]Hydrogen embrittlement relief required; bake at 325° F for 2 hrs.
[b]Heat-treat sample is a rod of the same material that accompanies the gear through its heat-treatment cycle.
[c]Heat-treat sample is from the same material and has the same cross section as the gear.
[d]Heat-treat sample is a section cut from an actual gear.
[e]Traceability of inspection equipment to standards.

Preparatory to selecting the material for the gear case, consider the effect on the gear alignment, clearances, and backlash of the relative gear and gear-case deflections as governed by the elastic modulus and thermal expansion coefficient of the intended gear-case material. Determine the resulting changes in center distance and backlash, and choose another material if the results are detrimental to gear operation or gear life.

3.4 FABRICATION

Gear fabrication processes and process controls shall minimize conditions that limit gear life.

The recommended fabrication techniques, processes, and process controls are summarized in table XXIV and in sections 3.4.1 through 3.4.5 that follow.

3.4.1 Forging

Material grain orientation shall follow approximately the finished gear outline.

Forgings should be used to produce gear blanks for all turbopump power train gears.

Either bar stock or forgings may be used for accessory gears. Open- or closed-die forgings should be used for grade 2 gears. Accurate closed-die forgings are needed for grade 3 gears. High-energy-rate forgings of gear bodies complete with forged teeth are preferred for grade 4. All main-power gears for turbopumps should be grade 3 or better and should be made from forging blanks in which the direction of grain flow is controlled. These forgings should contain no laps, voids, or banding. The forging supplier and the dies used should be qualified and approved. The grain flow required in the cross section of a typical forging is shown in figure 23.

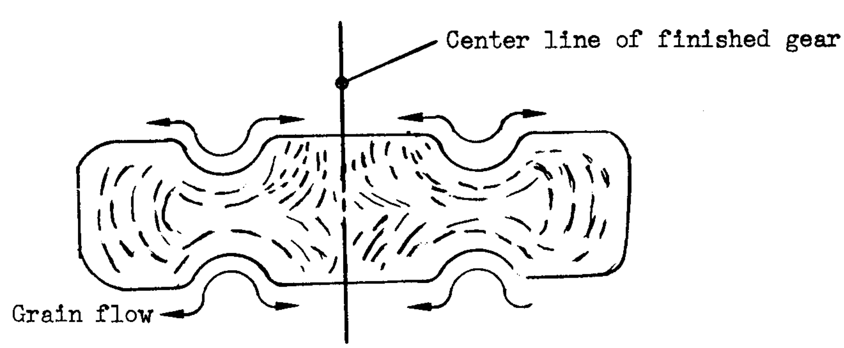


Figure 23. – Cross-section sketch of a forging showing proper grain flow.

3.4.2 Tooth Cutting

The tooth cutting method shall satisfy the gear configuration requirements without reducing gear load capacity.

Hobbing is the preferred cutting method because it produces a good finish; however, axial clearance must exist for the cutter. Shaping should be used to cut gear teeth near other components where axial space is limited. Use short-pitch hobs to obtain optimum fillet radii, required undercut, and grinding stock at the form diameter for specialized turbopump gears. Green grinding should be used for work-hardening metals and for obtaining a good root finish. Gears finer than 20 pitch can be ground from an uncut blank; gears coarser than 20 pitch should be precut before gear grinding. Consult reference 10 (chapters 5 and 6) for a more complete discussion of tooth cutting and tool design. The gear designer and the manufacturer must coordinate and approve the tool form and cutter design. Sufficient lead time should be allowed for procurement and testing of cutters and grinding forms.

3.4.3 Heat Treatment

The heat treatment of gears shall produce the material properties required without inducing defects.

The gear blank, whether bar or forging, should be normalized at 1700° to 1750° F before any machining is started.

The proper material properties listed in section 3.3.1.2 should be obtained by carburizing after tooth cutting. Areas where hardening is not desired should be masked by copper plating.

The pack-type carburizing furnace is not recommended for any gear grade. A dew-point-control, pit-type furnace with an infrared gas control may be used through grade 3 if control and timing on carburized samples is very strict. The type of furnace recommended for carburizing most gears is a vacuum-retort stainless-steel type with an infrared analyzer for control. Duplex carburizing processes that reduce the furnace carbon potential at the end of the cycle should be employed to obtain the desired carbon content gradient and to minimize retained austenite and carbide networks near the outer surface. For critical gears, choose only a gear manufacturer experienced in carburizing procedures.

3.4.4 Tooth Finishing

Finishing operations after carburizing and hardening shall not reduce gear capacity.

The copper place that is used to mask the gear and thus prevent carburization in the areas where it is unwanted should be removed by a process not involving acid, because acid introduces the potential for hydrogen embrittlement.

The close tolerances necessary on gear profiles usually require a finishing operation after hardening, because of distortion resulting from the hardening process. Grinding is the process recommended for correcting this distortion. Abrasive honing may be used on accessory gears if the scoring risk is low and if the pitchline velocity is below 20 000 ft/min. Lapping is not recommended for turbopump gears. Teeth should not be etched to obtain profile or lead modification.

Lightly loaded accessory gears may be ground in the root to maintain a good tooth contour. Highly loaded accessory gears and main-power-train gears should not be ground in the root area after hardening.

Figure 15 illustrates the grinding pattern for an external gear. The tooth is divided into three zones. Zone A, the active profile, is the area in which the finishing operation (grinding or honing if allowed) should be performed. Zone C extends a distance C from the root diameter toward the tip of the tooth. The distance C may be computed as

$$C = \frac{0.250}{P_d}$$

Zone C is the zone in which finishing operations should not be performed on highly loaded gears. Zone B, which lies between Zones A and C, is the zone in which finishing is allowed but not required. The transition from the finished tooth profile to the unfinished root in zone B can result in a mismatch that takes the form of undercut or blend as illustrated in figure 15.

For gears of material quality grades 3 and 4, it is recommended that maximum values be specified for undercut and blend and that a minimum value be specified for blend radius. The values to be specified depend on the application and the manufacturing plan. Typical values for a material grade 3 gear in the 8- to 12-pitch range are as follows:

Undercut (maximize in the range of) . . . . 0.005 to 0.007 in.

Blend (maximize up to) . . . . 0.002 in.

Blend radius (minimize in the range of) . . . . 0.020 to 0.030 in.

The amount of metal removed from the carburized areas of the tooth should not exceed 20 percent of the total case depth available. Table XVI gives recommended maximum values for grinding stock. It is recommended that the amount of material ground away after hardening always be kept to the minimum that allows the gear to meet the dimensional tolerances. Procedures presented in reference 44 should be followed in inspecting the gear for alteration of surface temper by the finishing process.

3.4.5 Shot Peening

Residual compressive stress necessary for maximum fatigue life shall exist in gear surfaces subject to cyclic tensile stresses.

Shot peen the tooth root, rim, and web surfaces of turbopump power gears. The location of peening and the direction of the shot stream should be clearly specified. In peening tooth roots, the shot stream should be directed radially inward toward the gear center. Invoke MIL-S-13165 to control the shot-peening process. Process control is required, since no accurate methods of inspection for shot-peening quality exist. Automated shot-peening control should be specified to eliminate human performance variability. Round cast steel shot is recommended; hardness should be 42 to 55 on the Rockwell C scale. Nominal shot size should not be greater than 1/2 nor less than 1/4 of the fillet radius. Number 130 shot per MIL-S-13165 is recommended for rocket-engine-gear tooth roots with fillet radii of 0.038 in. and larger. For radii of 0.025 to 0.030 in., use number 110 size shot. Shot should be checked continuously during processing. Undersize and broken shot should be eliminated.

Shot-peening requirements for the roots of turbopump gears of 8 to 12 pitch should specify an Almen strip arc height of 0.015A with an exposure time of 4X. Surface roughness in the root should be specified (in μin. AA) before and after shot peening as shown in table XIX.

To ensure proper shot peening of critical areas (the most important area is the tooth-root fillet area), the entire gear tooth including sides, tip, and root should be peened after all heat-treatment and pregrind manufacturing operations are complete. Subsequent grinding should be confined to profile finishing and noncritical areas (sides and tip).

Multipeening (peening the same surface with different size shot) should be used to obtain the maximum beneficial residual compressive stress (ref. 35).

Rim and web surfaces should be shot peened perpendicular to the web surface; use as control an Almen strip height of 0.010A to 0.015A with an exposure time of 4X (ref. 34). Surface roughness after peening should not exceed 125 μin. AA.

3.4.6 Configuration Control

Information for control of gear configuration shall be fully descriptive of gear manufacturing requirements.

A supplementary data block should be added to the gear drawing to present information necessary to control the gear configuration. Table XI is an example of data-block information for two types of turbopump gears.

Sketches of measuring instrument traces showing required involute profile and lead modifications, allowable deviations, and tolerance bands should be made part of the gear drawing (e.g., fig. 13).

For ground main-power gears, where grinding of fillets and roots is not permitted, include on the gear drawing detailed dimensional sketches of the fillets and roots 5 to 10 times size (fig. 15).

Hardness test locations should be indicated on the drawing.

3.5 TESTING

3.5.1 Acceptance Testing

The quality-assurance tests performed prior to acceptance of the gears shall demonstrate conformance to the material and processing requirements but shall not have a detrimental effect on the gears.

Use only a 15N hardness indentor to measure hardness at the pitch line of the gear teeth. Rockwell C scale hardness may be used for testing core hardness or for testing case hardness on heat-treat samples and destructively tested gears. Core hardness requirements must be met anywhere on the uncarburized part of the sample gear, on a section removed from a test gear, or on a test sample. Case depth should be established by a hardness traverse (fig. 17) taken on an actual tooth section. On a carburized gear, the case depth is considered to be the distance from the outer surface to the point at which the hardness has decreased to 50 R_c.

Retained austenite content of accessory and medium power gears should be measured by either visual examination or X-ray diffraction. In cases of conflict, the measurements made by X-ray diffraction should be used. The austenite content of critical gears such as main-power turbopump gears should always be measured by X-ray diffraction.

Grain size should be determined by the methods noted in ASTM E-112-63.

All gears intended for further service should be magnetic-particle inspected per MIL-M-11472, and those with flaws should be rejected.

3.5.2 Performance Testing

Performance tests shall verify that the gear system will operate satisfactorily at actual turbopump operating conditions.

In all tests conducted to demonstrate the adequacy of a new design, the gear train should be operated at full speed while loaded by dynamometer, water brake, or back-to-back tester. The following series of tests is recommended:

(1) Short run, low load
(2) Short run, full load
(3) Full duration, full load
(4) Required qualification life
(5) Ten-percent overload, qualification life
(6) Ten-percent overload, duration to failure
(7) Lubricant flow limits
(8) Simulated failure: plugged lubrication jet, leaking seal, or other realistic failure modes.

After any design modifications indicated by the back-to-back test series are incorporated, the gear evaluation should be continued by observation of test data and examination of gears after testing in turbopump hot-fire and static engine runs. Gear tooth surface condition usually can be monitored throughout the test series by periodic visual examination of the gears through the lubricant jet mounting port in the gear case.

The use of the back-to-back test arrangement also is recommended for trouble shooting problems that may become apparent after the gear system is operational.

Gear tests should be conducted with instrumentation adequate to measure accurately the following parameters:

(1) Shaft speed (input or output).

(2) Lubrication system:
   Flow rate
   Inlet temperature
   Outlet temperature
   Inlet pressure to distribution manifold
   Individual jet pressures for critical jets.

(3) Bearing temperatures on critical bearings.

(4) Vibration. – One sensitive accelerometer located anywhere on the case will detect major vibration. However, if resonances are expected, the use of multiple accelerometers located close to the support bearings for each shaft in question is a greater aid in locating incipient problems.

(5) Shock intensity. – A shock pulse meter or other tuned response meter can be of great aid in discovering bearing fatigue or similar trouble as it develops. Stopping the test prior to failure will prevent extensive damage that might otherwise mask the actual cause of the failure.

(6) Prime mover torque. – This variable often may be the first criterion of increasing load, heat generation, or other degradation of gear condition.

(7) Audible sound. – Change in pitch often accompanies a developing trouble and may alert the test crew to watch other parameters closely.

APPENDIX A

Conversion of U.S. Customary Units to SI Units

Physical quantity	U.S. customary unit	SI unit	Conversion factor[a]
Force	kgf	N	9.807
	lbf	N	4.448
Length	ft	m	0.3048
	in.	cm	2.54
	mil	μm	25.4
Mass	lbm	kg	0.4536
Power	hp	W	745.7
Pressure	mm Hg	N/m^2	133.3
	psi ($lbf/in.^2$)	N/m^2	6895
Rotational speed	rpm	rad/sec	0.1047
Specific heat	$\frac{Btu}{lbm\text{-}°F}$	$\frac{J}{kg\text{-}K}$	4184
Stress	psi ($lbf/in.^2$)	N/m^2	6895
Temperature	°F	K	$K = \frac{5}{9}(°F + 459.67)$
Tensile strength	psi ($lbf/in.^2$)	N/m^2	6895
Thermal energy	Btu	J	1054
Torque	in.-lbf	N-m	0.1130
Viscosity	centistokes	m^2/sec	1.00×10^{-6}
Volume	gal	m^3	3.785×10^{-3}

[a]Multiply value given in U.S. customary unit by conversion factor to obtain equivalent value in SI unit. For a complete listing of conversion factors, see Mechtly, E. A.: The International System of Units. Physical Constants and Conversion Factors. Second revision, NASA SP-7012, 1973.

APPENDIX B

GLOSSARY

Symbol	Definition
AA	arithmetic average
a	addendum
D	pitch diameter of gear
D_b	base circle diameter of gear
D_o	outside diameter of gear
d	(1) dedendum (2) pitch diameter of pinion (3) overhang (fig. 3)
d_b	base circle diameter of pinion
d_o	outside diameter of pinion
EP	extreme pressure
F	face width of gear
h	lubricant film thickness
h_t	tooth whole depth (total depth)
J	geometry factor for bending strength
K	tooth pitting index (preliminary design value representative of compressive stress)
K_m	modifying factor for load distribution
K_o	modifying factor for overload
K_s	modifying factor for size
K_t	modifying factor for temperature
K_v	modifying factor for dynamic load

Symbol	Definition
m_G	gear ratio
P	pitch; Hertz contact pressure
P_d	diametral pitch, $P_d = \dfrac{\text{number of teeth}}{\text{pitch diameter, in.}}$
PA	pressure angle
PLV	pitchline velocity
PV	scoring index, defined in note on Table V
PVT	scoring index, defined in note on Table V
ppi	pounds per inch (of face width)
R_c	hardness on Rockwell "C" scale
ref.	dimension given for reference only; not to be measured
S	bearing span
S_{ac}	allowable compressive stress
S_c	compressive stress
S_{tu}	ultimate tensile strength
S_{ty}	yield tensile strength
s	surface roughness
t	tooth thickness
TIF	true involute form
U_L	unit load (preliminary design value representative of bending stress)
W_t	total tangential tooth load
Y	tooth form factor
λ	h/s

Symbol	Definition
15N	hardness scale for the superficial Rockwell hardness test (use of the 15N scale results in a very small indentation)

Material	Identification
A356-T61	high-strength cast aluminum alloy, temper T61
A357-T61	high-strength cast aluminum alloy in which special properties can be developed by careful control of casting and chilling techniques; temper T61
AISI 410 440C	AISI designations for corrosion-resistant hardenable steels
AISI 3310 4140 4340 4620 8620 9310 9315	AISI designations for low-alloy carbon steels
AMS 5630	wrought corrosion- and heat-resistant steel per AMS 5630
AMS 6260 6265 6470	wrought low-alloy steels per AMS 6260, 6265, and 6470 respectively
Berylco 25	beryllium-copper alloy made by Kawecki Berylco Industries, Inc.
carburizing steel	low-carbon-content steel that can be treated to cause the metal to absorb carbon into the surface, thereby increasing surface hardenability while maintaining a weaker, softer, but tougher core
CRES	corrosion-resistant steel
GH_2	gaseous hydrogen
Inconels	trade name of International Nickel Co. for austenitic nickel-base alloys
IRFNA	inhibited red fuming nitric acid, propellant grade per MIL-P-7254
LH_2	liquid hydrogen, propellant grade per MIL-P-27201

Material	Identification
LO_2, LOX	liquid oxygen, propellant grade per MIL-P-25508
N_2H_4	hydrazine, propellant grade per MIL-P-26536
N_2O_4	nitrogen tetroxide, propellant grade per MIL-P-26539
nital	solution of concentrated nitric acid in alcohol
nitriding steel	steel alloyed with nitride-forming elements such as aluminum, chromium, molybdenum, vanadium, and tungsten. Exposure of the alloy to active nitrogen results in a thin hard case that is especially wear resistant. Precautions are necessary to avoid chipping. Cost is higher than that for carburizing.
Oronite 262	zinc dialkyl dithiophosphate additive, Oronite Div., Chevron Chemical Co.
RP-1	kerosene-base fuel, propellant grade per MIL-P-25576
Tens-50	high-strength aluminum alloy for casting
UDMH	unsymmetrical dimethylhydrazine, propellant grade per MIL-P-25604

Specification	Title
Materials[1]	
AMS 3132	Varnish, Synthetic Resin Corrosion Preventive.
AMS 3170	Thinner, Alcohol-Ester
AMS 5630	Bars and Forgings – 17 Cr-0.5 Mo (0.95-1.20 C).
AMS 6260	Bars, Forgings, and Mech. Tubing – 3.25 Ni-1.2 Cr-0.12 Mo (0.07-0.13 C)
AMS 6265	Bars, Forgings, and Mech. Tubing – 3.25 Ni-1.2 Cr-0.12 Mo (0.07-0.13 C) Premium Quality Consumable Electrode Vacuum Melted.
AMS 6470	Bars, Forgings, and Mech. Tubing, Nitriding – 1.6 Cr-0.35 Mo-1.13 Al (0.38-0.43 C)

[1] Specifications designated AMS are published by Society of Automotive Engineers, Inc., 2 Pennsylvania Plaza, New York, NY 10001. Military specifications are published by the Department of Defense, Washington, DC 20025.

Specification	Title
Materials	
MIL-C-16173	Corrosion Preventive Compound, Solvent Cutback, Cold-Application.
MIL-L-6081 (ASG)	Lubricating Oil, Jet Engine.
MIL-L-6086	Lubricating Oil, Gear, Petroleum Base.
MIL-L-7808	Lubricating Oil, Aircraft Turbine Engine, Synthetic Base.
MIL-L-25336	Lubricating Oil, Aircraft Turbine Engine, Synthetic Base, High Film Strength.
MIL-P-25576	Propellant, Kerosene.
MIL-P-27401	Propellant Pressurizing Agent, Nitrogen.
MIL-P-27407 (USAF)	Propellant Pressurizing Agent, Helium.
Processes and Test Methods	
ASTM D-2596-69	Extreme-Pressure Properties of Lubricating Grease (Four Ball Method), Measurement of. ASTM 17 (1969).
ASTM E112-63	Estimating Grain Size of Metals. ASTM, 1963.
Federal Test Method Std. No. 791, Method 6503.1	Load Carrying Capacity (Mean Hertz Load). Jan. 15, 1969. Contained in FTM Std. No. 791B, Change Notice 1, Oct. 15, 1969.
Federal Test Method Std. No. 791, Method 6508.1	Load Carrying Capacity of Lubricating Oils (Ryder Gear Machine). Jan. 15, 1969. Contained in FTM Std. No. 791B, Change Notice 1, Oct. 15, 1969.
MIL-C-26074	Coatings, Electroless Nickel, Requirements for.
MIL-M-11472 (ORD)	Magnetic Particle Inspection; Process for Ferromagnetic Materials.
MIL-S-13165	Shot Peening of Metal Parts.
QQ-C-320a (Federal Specification)	Chromium Plating (Electrodeposited).

Vehicles, Pumps, and Engines	Identification
Agena	upper stage for Atlas and Thor launch vehicles; uses LR81-BA-11 engine
Atlas (SLV-3)	launch vehicle using MA-5 engine system containing 2 booster, 2 vernier, and 1 sustainer engines; boosters provide 330 000 to 370 000 lbf thrust; sustainer, 60 000 lbf thrust; uses LOX/RP-1; engine system manufactured by Rocketdyne Division, Rockwell International Corp.
Centaur	high-energy upper stage for Atlas and Titan launch vehicles; uses 2 RL10 engines
H-1	engine for S-IB; 200 000 lbf thrust; uses LOX/RP-1; manufactured by Rocketdyne
LR81-BA-11	engine for Agena upper stage; 15 000 lbf thrust; uses IRFNA/UDMH; manufactured by Bell Aerospace Company, Division of Textron
LR-87-AJ-5	Aerojet engine for the first stage of the Titan II; uses N_2O_4/A-50* and develops 215 000 lbf thrust
LR-91-AJ-5	Aerojet engine for the second stage of the Titan II; uses N_2O_4/A-50* and develops 100 000 lbf thrust
Mark 3	turbopump for the engines in the Atlas, Thor, and Saturn IB boosters; manufactured by Rocketdyne
Mark 4	turbopump for the Atlas sustainer engine; manufactured by Rocketdyne
RL10	engine for Centaur upper stage; 15 000 lbf thrust; uses LOX/LH_2; manufactured by Pratt & Whitney Aircraft Division of United Aircraft Corp.
S-IB	first stage (booster) of the Saturn IB vehicle; uses a cluster of eight H-1 engines
Thor	launch vehicle using MB-3 engine system; 170 000 lbf thrust; uses LOX/RP-1; engine system manufactured by Rocketdyne
Titan II	launch vehicle using the LR-87-AJ and LR-91-AJ series of rocket engines developed by Aerojet Liquid Rocket Co.

*50/50 mixture of UDMH and hydrazine.

Abbreviation	Identification
AGMA	American Gear Manufacturers Association
AISI	American Iron and Steel Institute
AMS	Aerospace Material Specifications (published by SAE)
ASA	American Standards Association
ASLE	American Society of Lubrication Engineers
ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
SAE	Society of Automotive Engineers

REFERENCES

1. Anon.: Terms, Definitions, Symbols, and Abbreviations. AGMA 112.04, AGMA, Aug. 1965.

2. Dudley, D. W., ed.: Gear Handbook. McGraw-Hill Book Co., 1962.

3. Anon.: Design Procedure for Aircraft Engine and Power Take-Off Spur and Helical Gears. AGMA 411.02, AGMA, Sept. 1966.

4. Anon.: AGME Gear Handbook. Vol. 1 – Gear Classifications, Materials, and Measuring Methods for Unassembled Gears. AGMA 390.03, AGMA, Jan. 1973.

5. McIntire, W. L.; and Malott, R. C.: Advancement of Spur Gear Design Technology. USAAVLABS Tech. Rep. 66-85, U.S. Army Aviation Lab. (Fort Eustis, VA), 1966.

6. McIntire, W. L.; and Malott, R. C.: Advancement of Helical Gear Design Technology. USAAVLABS Tech. Rep. 68-47, U.S. Army Aviation Lab. (Fort Eustis, VA), 1968.

7. Coleman, W.; Lehmann, E. P.; Mellis, D. W.; and Peel, D. M.: Advancement of Straight and Spiral Bevel Gear Technology. USAAVLABS Tech. Rep. 69-75, U.S. Army Aviation Lab. (Fort Eustis, VA), Oct. 1969.

8. Anon.: Information Sheet for Strength of Spur, Helical, Herringbone and Bevel Gear Teeth. AGMA 225.01, AGMA, Dec. 1967.

9. Seabrook, J. B.; and Dudley, D. W.: Results of 15-Year Program of Flexural Fatigue Testing of Gear Teeth. J. Eng. Ind. Trans. ASME, Series B, vol. 86, 1964, pp. 221-239.

10. Dudley, D. W.: Practical Gear Design. McGraw-Hill Book Co., 1954.

11. Anon.: Information Sheet for Surface Durability (Pitting) of Spur, Helical, Herringbone, and Bevel Gear Teeth. AGMA 215.01, AGMA, Sept. 1966.

12. Costomiris, G.; Daley, D.; and Grube, W.: Heat Generated in High Power Reduction Gearing. PWA-3718, Pratt & Whitney Aircraft Div., United Aircraft Corp. (East Hartford, CT), June 1969.

13. Bodensieck, E. J.: Specific Film Thickness – An Index of Gear Tooth Surface Deterioration. Paper presented at 1965 Aerospace Gear Comm. Tech. Div. Meeting, AGMA (Denver, CO), Sept. 1965.

14. Anon.: Information Sheet – Gear Scoring Design Guide for Aerospace Spur and Helical Power Gears. AGMA 217.01, AGMA, Oct. 1965.

15. Borsoff, V. N.; and Godet, M. R.: A Scoring Factor for Gears. ASLE Trans., vol. 6, no. 2, 1963, pp. 147-153.

16. Lemanski, A. J.: A Comparison of Gear Scoring Indices. Vertol Div., Boeing Co. (Morton, PA), Feb. 1965.

17. Hartman, M. A.: Advances in Aerospace Power Gears. Power Transmission Design, vol. 9, no. 11, Nov. 1967, pp. 40-47.

18. Butner, M. F.: Propellant Lubrication Properties Investigation, Final Report. Rep. WADD-TR-61-77, Pts. I and II (AD 259143), June 1962.

19. Lorvick, R. R.: Lubricating Gears. Mach. Des., vol. 42, no. 17, July 1970, pp. 108-117.

20. McCain, J. W.; and Alsandor, E.: Analytical Aspects of Gear Lubrication on the Disengaging Side. ASLE paper 65-LC-16, ASLE and ASME Lubrication Conf. (San Francisco, CA), Oct. 18-20, 1965.

21. McIntire, W. L.: How to Reduce Gear Vibration Failures. Allison Div., General Motors Corp. (Indianapolis, IN), Feb. 1964.

22. Dudley, W. M.; and Hall, Ira K., Jr.: Analysis of Lateral Vibrations of Gears and Rimmed Wheels. ASME paper 66-MD-4, ASME Design Eng. Conf. and Show (Chicago, IL), May 9-12, 1966.

23. Reiger, N.: Vibration Characteristics of Geared Transmission Systems. AGMA 109.14, AGMA, Oct. 1964.

24. Anon.: Tooth Proportions for Coarse-Pitch Involute Spur Gears (ANSI B6.1-1968). AGMA 201.02, AGMA, Aug. 1968.

25. Anon.: Tooth Proportions for Fine-Pitch Involute Spur and Helical Gears (ANSI B6.7-1967). AGMA 207.05, AGMA, June 1971.

26. Dolan, T. J.; and Broghamer, E. I.: A Photoelastic Study of the Stresses in Gear Tooth Fillets. Univ. of Illinois Eng. Expt. Sta. Bull. 335, Univ. of Illinois (Urbana, IL), Mar. 1942.

*27. Hartman, M. A.: Profile Modification for Heavily Loaded Gears Under Dynamic Conditions. Rocketdyne Div., North American Rockwell Corp. (Canoga Park, CA), Nov. 1967 (unpublished).

28. Pollack, C.: Better Surface Integrity – Secret of Part Reliability. Machinery, vol. 75, no. 3, Nov. 1968, pp. 78-80.

29. Lavoie, F. J.: High Velocity Forging of Gears. Mach. Des., vol. 40, no. 28, Dec. 1968, pp. 146-151.

30. Burroughs, L. R.; and Fitzgerald, P. C.: Evaluation of Advanced Gear Forging Techniques. USAAVLABS Tech. Rep. 69-11, U.S. Army Aviation Lab. (Fort Eustis, VA), Apr. 1969.

*Dossier for design criteria monograph "Liquid Rocket Engine Turbopump Gears." Unpublished. Collected source material available for inspection at NASA Lewis Research Center, Cleveland, Ohio.

31. Parkinson, F. L.: Evaluation of High-Energy-Rate Forged Gears With Integral Teeth. USAAVLABS Tech. Rep. 67-11, U.S. Army Aviation Lab. (Fort Eustis, VA), Mar. 1967.

32. Anon.: Recommended Procedure for Carburized Industrial Gearing. AGMA 246.01A, AGMA, Nov. 1971.

33. Straub, J. C.: Shot Peening in Gear Design, 1964. AGMA 109.13, AGMA, June 1964.

34. Anon.: Shot Peening. Wheelabrator Corp. (Mishawaka, IN), 1965.

35. Anderson, B. N.; and Hartman, M. A.: Multipeening of Surfaces To Extend Fatigue Life. Technology Utilization Docket NAR 50477, Rocketdyne Div., North American Rockwell Corp., Mar. 1966.

36. Bush, J. J.; Mattson, R. L.; and Roberts, J. G.: Shot Peening Treatments. U.S. Patent 3,073,022, assigned to General Motors Corp., issued Jan. 1963.

37. Anon.: Turbopump Systems for Liquid Rocket Engines. NASA Space Vehicle Design Criteria Monograph, NASA SP-8107 (to be published).

38. Anon.: Liquid Rocket Engine Turbopump Bearings. NASA Space Vehicle Design Criteria Monograph, NASA SP-8048, Mar. 1971.

39. Buckingham, E.: Analytical Mechanics of Gears. McGraw-Hill Book Co., 1949.

40. Campbell, M. E.; Loser, J. B.; and Sneegas, E.: Solid Lubricants. NASA SP-5059, May 1966.

41. Anon.: Maag Gear Book. Maag Gear Wheel Co. (Zurich, Switzerland), Dec. 1965.

42. Sigg, H.: Profile and Longitudinal Corrections on Involute Gears. AGMA 109.6, AGMA, Oct. 1965.

43. Lyman, T., ed.: Metals Handbook. Vol. 1: Properties and Selection of Metals. Eighth ed., American Society for Metals (Metals Park, OH), 1961.

44. Anon.: Surface Temper Inspection Process. AGMA 230.01, AGMA, Mar. 1968.

NASA SPACE VEHICLE DESIGN CRITERIA MONOGRAPHS ISSUED TO DATE

ENVIRONMENT

SP-8005	Solar Electromagnetic Radiation, Revised May 1971
SP-8010	Models of Mars Atmosphere (1967), May 1968
SP-8011	Models of Venus Atmosphere (1972), Revised September 1972
SP-8013	Meteoroid Environment Model–1969 (Near Earth to Lunar Surface), March 1969
SP-8017	Magnetic Fields–Earth and Extraterrestrial, March 1969
SP-8020	Mars Surface Models (1968), May 1969
SP-8021	Models of Earth's Atmosphere (90 to 2500 km), Revised March 1973
SP-8023	Lunar Surface Models, May 1969
SP-8037	Assessment and Control of Spacecraft Magnetic Fields, September 1970
SP-8038	Meteoroid Environment Model–1970 (Interplanetary and Planetary), October 1970
SP-8049	The Earth's Ionosphere, March 1971
SP-8067	Earth Albedo and Emitted Radiation, July 1971
SP-8069	The Planet Jupiter (1970), December 1971
SP-8084	Surface Atmospheric Extremes (Launch and Transportation Areas), May 1972
SP-8085	The Planet Mercury (1971), March 1972
SP-8091	The Planet Saturn (1970), June 1972
SP-8092	Assessment and Control of Spacecraft Electromagnetic Interference, June 1972

SP-8103	The Planets Uranus, Neptune, and Pluto (1971), November 1972
SP-8105	Spacecraft Thermal Control, May 1973

STRUCTURES

SP-8001	Buffeting During Atmospheric Ascent, Revised November 1970
SP-8002	Flight-Loads Measurements During Launch and Exit, December 1964
SP-8003	Flutter, Buzz, and Divergence, July 1964
SP-8004	Panel Flutter, Revised June 1972
SP-8006	Local Steady Aerodynamic Loads During Launch and Exit, May 1965
SP-8007	Buckling of Thin-Walled Circular Cylinders, Revised August 1968
SP-8008	Prelaunch Ground Wind Loads, November 1965
SP-8009	Propellant Slosh Loads, August 1968
SP-8012	Natural Vibration Modal Analysis, September 1968
SP-8014	Entry Thermal Protection, August 1968
SP-8019	Buckling of Thin-Walled Truncated Cones, September 1968
SP-8022	Staging Loads, February 1969
SP-8029	Aerodynamic and Rocket-Exhaust Heating During Launch and Ascent May 1969
SP-8030	Transient Loads From Thrust Excitation, February 1969
SP-8031	Slosh Suppression, May 1969
SP-8032	Buckling of Thin-Walled Doubly Curved Shells, August 1969
SP-8035	Wind Loads During Ascent, June 1970
SP-8040	Fracture Control of Metallic Pressure Vessels, May 1970
SP-8042	Meteoroid Damage Assessment, May 1970
SP-8043	Design-Development Testing, May 1970

SP-8044	Qualification Testing, May 1970
SP-8045	Acceptance Testing, April 1970
SP-8046	Landing Impact Attenuation for Non-Surface-Planing Landers, April 1970
SP-8050	Structural Vibration Prediction, June 1970
SP-8053	Nuclear and Space Radiation Effects on Materials, June 1970
SP-8054	Space Radiation Protection, June 1970
SP-8055	Prevention of Coupled Structure-Propulsion Instability (Pogo), October 1970
SP-8056	Flight Separation Mechanisms, October 1970
SP-8057	Structural Design Criteria Applicable to a Space Shuttle, Revised March 1972
SP-8060	Compartment Venting, November 1970
SP-8061	Interaction with Umbilicals and Launch Stand, August 1970
SP-8062	Entry Gasdynamic Heating, January 1971
SP-8063	Lubrication, Friction, and Wear, June 1971
SP-8066	Deployable Aerodynamic Deceleration Systems, June 1971
SP-8068	Buckling Strength of Structural Plates, June 1971
SP-8072	Acoustic Loads Generated by the Propulsion System, June 1971
SP-8077	Transportation and Handling Loads, September 1971
SP-8079	Structural Interaction with Control Systems, November 1971
SP-8082	Stress-Corrosion Cracking in Metals, August 1971
SP-8083	Discontinuity Stresses in Metallic Pressure Vessels, November 1971

SP-8095	Preliminary Criteria for the Fracture Control of Space Shuttle Structures, June 1971
SP-8099	Combining Ascent Loads, May 1972
SP-8104	Structural Interaction With Transportation and Handling Systems, January 1973

GUIDANCE AND CONTROL

SP-8015	Guidance and Navigation for Entry Vehicles, November 1968
SP-8016	Effects of Structural Flexibility on Spacecraft Control Systems, April 1969
SP-8018	Spacecraft Magnetic Torques, March 1969
SP-8024	Spacecraft Gravitational Torques, May 1969
SP-8026	Spacecraft Star Trackers, July 1970
SP-8027	Spacecraft Radiation Torques, October 1969
SP-8028	Entry Vehicle Control, November 1969
SP-8033	Spacecraft Earth Horizon Sensors, December 1969
SP-8034	Spacecraft Mass Expulsion Torques, December 1969
SP-8036	Effects of Structural Flexibility on Launch Vehicle Control Systems, February 1970
SP-8047	Spacecraft Sun Sensors, June 1970
SP-8058	Spacecraft Aerodynamic Torques, January 1971
SP-8059	Spacecraft Attitude Control During Thrusting Maneuvers, February 1971
SP-8065	Tubular Spacecraft Booms (Extendible, Reel Stored), February 1971
SP-8070	Spaceborne Digital Computer Systems, March 1971
SP-8071	Passive Gravity-Gradient Libration Dampers, February 1971
SP-8074	Spacecraft Solar Cell Arrays, May 1971

SP-8078	Spaceborne Electronic Imaging Systems, June 1971
SP-8086	Space Vehicle Displays Design Criteria, March 1972
SP-8096	Space Vehicle Gyroscope Sensor Applications, October 1972
SP-8098	Effects of Structural Flexibility on Entry Vehicle Control Systems, June 1972
SP-8102	Space Vehicle Accelerometer Applications, December 1972

CHEMICAL PROPULSION

SP-8087	Liquid Rocket Engine Fluid-Cooled Combustion Chambers, April 1972
SP-8081	Liquid Propellant Gas Generators, March 1972
SP-8109	Liquid Rocket Engine Centrifugal Flow Turbopumps, December 1973
SP-8052	Liquid Rocket Engine Turbopump Inducers, May 1971
SP-8110	Liquid Rocket Engine Turbines, January 1974
SP-8048	Liquid Rocket Engine Turbopump Bearings, March 1971
SP-8101	Liquid Rocket Engine Turbopump Shafts and Couplings, September 1972
SP-8094	Liquid Rocket Valve Components, August 1973
SP-8097	Liquid Rocket Valve Assemblies, November 1973
SP-8090	Liquid Rocket Actuators and Operators, May 1973
SP-8080	Liquid Rocket Pressure Regulators, Relief Valves, Check Valves, Burst Disks, and Explosive Valves, March 1973
SP-8064	Solid Propellant Selection and Characterization, June 1971
SP-8075	Solid Propellant Processing Factors in Rocket Motor Design, October 1971
SP-8076	Solid Propellant Grain Design and Internal Ballistics, March 1972
SP-8073	Solid Propellant Grain Structural Integrity Analysis, June 1973

SP-8039	Solid Rocket Motor Performance Analysis and Prediction, May 1971
SP-8051	Solid Rocket Motor Igniters, March 1971
SP-8025	Solid Rocket Motor Metal Cases, April 1970
SP-8041	Captive-Fired Testing of Solid Rocket Motors, March 1971

NASA-Langley, 1974