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NASA CONTRACTOR REPORT

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FINAL REPORT FOR DESIGN OF AIRCRAFT TURBINE FAN DRIVE GEAR TRANSMISSION SYSTEM

By: E. Dent, R. A. Hirsch, and V. W. Peterson

Prepared under Contract No. NAS3-12417 by
ALLISON DIVISION OF GENERAL MOTORS
Indianapolis, Indiana 46206

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CONTRACTOR REPORT

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By E. Dent, R. A. Hirsch, and V. W. Peterson

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Allison Division of General Motors
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Allison EDR 6556

Technical Management

Lewis Project Manager: Harold Gold
Lewis Research Advisor: R. L. Cummings

Lewis Research Center Cleveland, Ohio
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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DESIGN OF AIRCRAFT TURBINE FAN DRIVE GEAR TRANSMISSION SYSTEM

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Allison Division of General Motors

SUMMARY

The following three basic types of gear reduction concepts were studied as being feasible power train systems for a low-bypass-ratio, single-spool, geared turbofan engine for general aircraft use:

- Single-stage external-internal reduction
- Gears (offset shafting) (see Figure 1)
- Multiple compound idler gear system (concentric shafting) (see Figure 2)
- Star gear planetary system with internal ring gear final output member (concentric shafting-counterrotation) (see Figure 3)

In addition, studies were made of taking the accessories drive power off both the high-speed and low-speed shafting, using either face gears or spiral bevel gears (see Figure 4). Both antifriction and sleeve-type bearings were considered for the external-internal and star-planet reduction concepts.

NASA Letter dated July 29, 1969, stopped all further design effort on sleeve-type bearings; therefore, only antifriction bearings were considered in the layshaft designs. The correspondence cited also eliminated the star-planet designs since the available space envelope restricted the planet bearings to sleeve-type bearings. Although it incorporates the minimum of parts, the internal-external concept (offset shafting) complicates the air inlet flow paths, has poor growth potential, and is higher in cost than the layshaft arrangement approved for detailing.

Two dual reduction layshafts were designed through the layout phase. The first one consisted of dual helical gear sets, fixed input sun, floating input quill shaft, floating output sun with thrust unbalance taken on the input sun bearing. This arrangement was highest in cost. The second concept consisted of a floating input sun, spur input gears, helical output gears, fixed output shaft with thrust unbalance taken on layshaft bearings. The second layshaft configuration was lowest in cost of all arrangements studied and was approved by NASA for detailing.

Tables I and II show the relative merits of the three basic concepts and the objections to hydrodynamic bearings.

INTRODUCTION

NASA Lewis Research Center at Cleveland, Ohio, has initiated a program for the design and development of a low-bypass-ratio, single-spool, geared turbofan engine for general aircraft use. The objective of this program is to provide a demonstrated study of the technology required to design, develop, and produce an extreme cost reduction type of gas turbine engine.

This design report is written in response to the contractual obligations specified in NASA contract NAS3-12417, dated February 25, 1969.

The objective of this program was to investigate potential reduction gear system arrangements and to make a final design and detail of the arrangement approved by the NASA project manager.

The program was divided into two phases, as shown in Table III. The original program was for nine months ending November 25, 1969, but the contract was amended to 13 months to provide more time to optimize the fanshaft and layshaft bearing loads. Upon approval by the NASA project manager, detail drawings were made of a three-layshaft reduction gear system. This arrangement is lowest in cost of four arrangements studied, possesses good growth potential, and meets the original objectives.

DESIGN SPECIFICATIONS

TYPE AND MODEL

This design specification defines the Allison modular concept Model AGB 2652 fan drive gear transmission for aircraft engine use. The transmission incorporates fan reduction gearing, lubrication system, fan rotor mounting flange, fan shroud attachment flange, fan stator support, bypass duct and gasifier inlet flow path and supporting struts, main engine mounts, gasifier case attachment flange, and accessory drive pad.

APPLICABLE DOCUMENT

The following specification, incorporated herein by reference and made a part hereof, forms a part of this specification to the extent specified herein:

MIL-L-7808G, Lubricating Oil

TECHNICAL DATA

Performance Characteristics

The ratings herein are based on the use of MIL-L-7808 oil.

Ratings

The performance ratings are as follows:

Rating	Input shaft hp	Gasifier rpm	Fan shaft rpm	Fan shaft hp
Takeoff	675	28,000	15,000	650
Cruise	360	28,000	15,000	340

Oil Consumption

The transmission total oil consumption shall not exceed 0.02 gal/hr at takeoff conditions.

OPERATIONAL REQUIREMENTS

Lubrication System

The lubrication system shall adequately lubricate the transmission, using MIL-L-7808 oil throughout its operating range. The minimum flow requirement is 2.4 gal/min at takeoff condition.

Heat Rejection

The heat rejection of the transmission to the oil shall not exceed 350 Btu/min corrected to standard sea level static conditions at takeoff fanshaft horsepower.

Oil Flow Interruption

The transmission shall operate continuously with no detrimental effect during and after a period of two seconds in which only air is supplied to the inlet of the oil pump.

Scavenging System

The scavenging system shall adequately scavenge the transmission under the operating conditions specified herein.

Oil Pressure and Temperature

The operating oil pressure and temperatures shall be 60 psig and 185°F maximum.

Time Between Overhaul

The transmission shall operate satisfactorily for 300 hours at takeoff condition based on a 2000-hour duty cycle.

PHYSICAL DESCRIPTION

Drawings and Data

The following Allison drawing forms a part of this specification:

6873811, Fan Reduction Gear and
Inlet Housing Assembly

Overall Dimensions

The overall dimensions of the transmission shall be as shown on the Fan Reduction Gear and Inlet Housing Assembly.

Dry Weight of Complete Transmission

The dry weight of the Fan Reduction Gear and Inlet Housing Assembly shall not exceed 65 pounds.

Weight of Residual Fluids

The estimated weight of residual fluids after transmission operation and drainage in the horizontal position is 0.5 pound.

Accessory Drive Pad

One power takeoff pad shall be provided which shall be designed to transmit 14 hp at a gasifier speed of 14,000 rpm with clockwise rotation of the drive shaft as viewed from the exterior face of the pad. The accessory pad shall be designed to support a maximum accessory weight of 65 lb with a maximum overhung moment of 355 lb-in.

OPERATIONAL AND DESIGN DETAILS

Design Torque

The design torque shall be at the maximum steady-state operating condition. This corresponds to 650 fan shaft hp at 28,000 rpm gasifier speed.

Limit Torque

The limit torque shall be 150% of the design torque. No permanent deformation is permissible.

Ultimate Torque

The ultimate torque shall be 200% of the design torque. No parts fracture may occur. Parts may deform permanently.

Design Speed

The design speed shall be the maximum steady-state operating speed. This corresponds to a gasifier speed of 28,000 rpm and a fan shaft speed of 15,000 rpm.

Limit Speed

The limit speed shall be 110% of the design speed without affecting the endurance characteristics or structural integrity of the transmission.

Ultimate Speed

The ultimate speed shall be 141% of the design speed without failure of the transmission drive train components or structure.

Attitude Conditions

The fan drive transmission shall be designed to operate satisfactorily under the following attitude conditions:

1. Level position (horizontal) with transmission inclined 20 degrees to either side.

2. Level position (horizontal) with the transmission inclined 45 degrees to either side for 15 seconds.

3. 90 degree diving angle for 10 seconds.

4. Zero to 30 degrees below horizontal with up to 10 degrees inclination on either side.

5. Zero to 45 degrees above horizontal with up to 10 degrees inclination on either side.

6. 90 degree climbing angle for 10 seconds.

7. Negative "g" or inverted flight for 10 seconds.

8. Zero "g" operation for 10 seconds.

Ambient Temperature Conditions

The transmission shall operate satisfactorily in any of the attitudes listed under Attitude Conditions of this specification under the following conditions:

1. Operate satisfactorily on the ground after a soaking period of 10 hours at an ambient temperature of 160°F.

2. Operate satisfactorily on the ground and in flight after a soaking period of 10 hours at ambient temperature of -20°F.

3. Operate satisfactorily throughout an ambient air temperature range of from -65°F to 250°F.

STRUCTURE

Mass Moment of Inertia of Rotating System

The combined moment of inertia of the fan output shaft assembly rotating parts is 39.4 lb-in.^{2}.

Gyroscopic Moments

At maximum rated speed, the fan transmission shall withstand a gyroscopic moment imposed by a steady angular velocity of 2 radians per second in yaw combined with a vertical load factor of ±1 for a period of 30 seconds.

Flight Maneuver Forces

The transmission and its supports shall be compatible with the FAA requirements and as established on NASA drawing CDL-11270.

Transient Torque Loads

The transient torque load factor on the transmission mounts is 1.5.

THREE-LAYSHAFT MECHANICAL DESIGN

MECHANICAL FEATURES

The three-layshaft design selected by NASA for design detailing is shown in Figure 5. The design incorporates the following salient features:

- Designed for four potential plane/engine installations
- Three layshafts for more uniform load sharing
- Dual reduction train:
  - First reduction gear set—spur
  - Second reduction gear set—helical
- Integral input sun gear, accessories drive pinion, and splined power input shaft (floating)
- Integral accessories drive gear and shaft
- Fan thrust reacted by helical gear set
- Fan-helical gear thrust unbalance reacted by layshaft rear bearing (ball)
- Power output shaft loads reacted by preloaded back-to-back angular contact ball bearings
- Integral seal support, fan shaft bearing support, and layshaft gear front bearing support
- Fan wheel attached to power output with Curvic coupling
- Bearing life exceeds 2000-hr duty cycle L2 life
- Simplified oil system

DESCRIPTION OF LAYSHAFT CONFIGURATION

Figure 5 shows the layshaft configuration which was approved by NASA for design detailing. Torque is transmitted from an internal spline in the gasifier rotor shaft to the floating power input sun gear. The accessories drive pinion is integral with the sun gear, and accessories drive power is taken off through a mating face gear. Three equally spaced layshafts supported in antifriction bearings radially position the sun gear. The first reduction gear set (spur) reduces the 28, 000-rpm gasifier speed to 24, 000 rpm (layshaft), and the second reduction gear set (helical) provides a fan shaft speed of 15, 000 rpm.

The helical fan output gear is supported in preloaded angular contact bearings, and the fan rotor is driven through a fixed Curvic coupling which provides positive drive with precision centering and high load carrying capacity. Lubrication of the gears and bearings is covered a little later in this report under "Heat Rejection and Lubrication System."

WEIGHT SUMMARY

Weights of the major components are shown in Table IV. The weight of the reduction gear and inlet housing casting is 31 lb and the weight of all other components is 29.344 lb, making a total weight of 60.344 lb. This weight should reduce to about 55 lb with weight reduction studies.

MATERIALS SUMMARY

Table V shows the major component materials. The O-rings are made from nitrile rubber and are satisfactory down to -40°F. The fanshaft and layshaft bearings are made from CEVM-52100 steel with iron-silicon-bronze separators. The accessories drive bearings are made from CVD-52100 steel with stamped metal separators. The studs are made from 160,000-psi chrome moly steel.

DESCRIPTION OF DESIGN DETAILS

Reduction Gear and Inlet Housing

The reduction gear and inlet housing is fabricated from a high-strength casting of AMS-4215 aluminum. The basic components are inner and outer shells and a splitter ring separating the bypass and gasifier gas paths connected by four hollow, equally spaced radial struts.

The outer shell contains the accessories drive gearbox mount pad, engine mount pads, flanges for attaching inlet doors, and bypass duct nozzle, and connections for one pressure-oil port and four potential scavenge ports.

The inner shell contains and supports the layshaft reduction gear unit assembly and provides support for the basic engine and gasifier compressor front (thrust) bearing. No provisions have been made for anti-icing; however, these features can be readily incorporated into the design.

Face Gear and Power Input Shaftgear

The 56-tooth, 20-diametral pitch accessories drive face gear was designed to AGMA Standard 203.02. Initially, shimming is required to compensate for the radial stack, but no axial shimming is necessary for the mating pinion. The output shaft male spline stresses are insignificant. The fundamental critical speed was calculated to be 98,000 rpm.

The power input shaftgear is positioned axially by either a shouldered bolt or stud and self-locking nut at the gasifier first-stage compressor wheel. The shaftgear can be assembled or disassembled with the unit assembly in the horizontal position. An O-ring at the splined end ensures that the spline is submerged in oil.

Layshaft Reduction Gear Unit

The layshaft gear and bearing support housing assembly is shown in Figure 6. The complete assembly can be assembled into the reduction gear and inlet housing once the face gear and power input shaftgear have been assembled. One bolt hole is offset at the mounting flange to prevent improper assembly.

The gear and bearing support housing is made from 80,000-psi nodular iron (AMS-5316) and is made up of two matched details doweled together. The front member supports the fanshaft bearings and layshaft roller bearings. The rear member supports the layshaft thrust bearings and provides oil porting from the oil pressure transfer tube to the oil manifold assembly. The layshaft thrust is reacted by a thrust plate sandwiched between the two members. The layshaft bearing bores are the same size, and the rear bore is open to facilitate manufacturing.

The helical gear power output shaft is supported by two back-to-back preloaded angular contact ball bearings separated by a matched inner and outer spacer. The outer spacer incorporates oil orifices for bearing lubrication. The bearing outer races are retained by the labyrinth seal stator and retaining plate which is shimmed for a 0.000 to 0.002-in. clearance. The bearing inner races are secured by a spanner nut which also retains the rotating labyrinth seal.

After the roller bearing inner races and rollers have been pressed onto the three layshafts, the thrust bearing shaft ends are passed through the shaft clearance holes in the thrust plate prior to assembling and securing the ball bearings. The resulting layshaft and thrust plate assembly, including the oil manifold assembly, support housing rear member, and bearing retaining rings, is assembled into the front housing as a unit while the index marks on the layshaft helical gears are aligned. The rear support housing is retained by six self-locking nuts.

Gears

The first reduction gear set is spur and the second reduction gear set is helical. This arrangement permits balancing most of the fan thrust by the helical gear set. The remaining thrust unbalance is reacted by the layshaft thrust bearings. All gear teeth are divisible by the number of layshafts for equal load sharing, and the layshaft cluster gears are index marked for proper assembly. A 20-degree pressure angle was selected in lieu of 25 degrees to reduce bearing loads. Specific gear data are shown in Table VII.

The gears are made from AMS-6370 (SAE 4130) and are nitrided all over except for balance correction areas, threads, and fixed Curvic coupling. The helical output gear, spur input gear, accessories drive pinion gear, and spur layshaft gear are designed for hobbing and shaving. The helical layshaft gear is shaped and shaved and the accessories drive face gear may be hobbed, shaped, or ground.

Bearings

The fanshaft ball bearings and layshaft roller and ball bearings are manufactured to ABEC 5/RBEC 5 tolerances and the rings and balls/rollers are made from CEVM-52100 steel. The separators are machined all over from iron-silicon-bronze. The accessories drive ball bearings are manufactured to ABEC 1 tolerances and are made from CVD-52100 steel with stamped sheet metal separators. Bearing geometry and AFBMA capacities are shown in Table VI.

DESIGN CALCULATIONS

The gear loads, bearing loads, and bearing life calculations are for a fan output gear helix angle of 12 degrees. This angle was selected to provide optimum layshaft ball bearing life. The bearing, gear, and layshaft positions are shown in Figure 7.

System Operating Conditions

Gasifier	Input shaft hp	rpm
Takeoff	675	28,000
Cruise	360	28,000

Fanshaft	hp	Thrust (lb)	rpm
Takeoff	650	400	15,000
Cruise	340	200	15,000

Accessory drive	hp	rpm	Gasifier rpm
Takeoff	14	12,000	28,000
Cruise	14	6,000	14,000

Gear Tooth Stress Calculations

Gear tooth bending and crushing stresses for single-tooth contact were calculated by an Allison computer program using the conventional Lewis and Hertz formula. High cycle fatigue based on 2000 hours determined the acceptable stress levels. The gears were sized based on duty cycle power

(15% takeoff and 85% cruise power). The gear design calculations are shown in Table VII. Table VIII compares these gear stresses with current aircraft practice.

Gear Loads

Takeoff

No. 1 & 2 gears
First reduction

$$hp = \frac{657.9550}{3} = 219.3183$$

$$T = 219.3183\left(\frac{63025}{28000}\right) = 493.662 \text{ lb-in.}$$

$$W_t = \frac{493.662}{0.9} = 548.5134 \text{ lb}$$

$$W_r = 548.5134\ (0.36397) = 199.6424 \text{ lb.}$$

No. 3 & 4 gears
Second reduction

$$hp = \frac{654.327}{3} = 218.109$$

$$T = 218.109\left(\frac{63025}{24000}\right) = 572.7633 \text{ lb-in.}$$

$$W_t = \frac{572.7633}{0.75} = 763.6842 \text{ lb}$$

$$W_r = 763.6842\ (0.36397) = 277.9582 \text{ lb}$$

Cruise

No. 1 & 2 gears
First reduction

$$hp = \frac{344.6757}{3} = 114.8919$$

$$T = 114.8919\left(\frac{63025}{28000}\right) = 258.6093 \text{ lb-in.}$$

$$W_t = \frac{258.6093}{0.9} = 287.3436 \text{ lb}$$

$$W_r = 287.3436\ (0.36397) = 104.5844 \text{ lb}$$

No. 3 & 4 gears
Second reduction

$$hp = \frac{342.6733}{3} = 114.2244$$

$$T = 114.2244\left(\frac{63025}{24000}\right) = 299.9581 \text{ lb-in.}$$

$$W_t = \frac{299.9581}{0.75} = 399.9441 \text{ lb}$$

$$W_r = 399.9441\ (0.36397) = 145.5677 \text{ lb}$$

Accessories Drive Pinion Gear Loads

Accessories drive hp = 14

$$\text{Torque} = \frac{14 \text{ hp } (63025)}{28,000 \text{ rpm}} = 31.5125 \text{ lb-in.}$$

$$\text{Tangential load } (W_t) = \frac{\text{Torque}}{\text{Pitch radius}} = \frac{31.5125}{0.6} = 52.52 \text{ lb}$$

$$\text{Separating load } (W_r) = W_t \text{ (cos press. angle)} = 19.11 \text{ lb}$$

The accessories drive pinion gear loads are reacted at the gasifier bearing and through the sun gear to the layshaft bearings. (Refer to Figure 8.)

Sun Gear Loads Due to Accessories Drive

$$\text{Tangential load } (W_t) = 52.52\left(\frac{4.6}{6.4}\right) = 37.7493 \text{ lb}$$

$$\text{Separating load } (W_r) = 19.11\left(\frac{4.6}{6.4}\right) = 13.7396 \text{ lb}$$

The resultant of the preceding loads is represented by vector AC in Figure 9. Its component vectors are either additive to or subtractive from the first reduction gear loads, depending on the respective layshaft positions.

Layshaft First Reduction Gear Resultant Loads

Takeoff

Layshaft No.	Tangential	Separating
	548.5134	199.6424
	+37.7493	+13.7396
1	586.2627 lb	213.3820 lb
	548.5134	199.6424
	-30.7735	+25.8220
2	517.7399 lb	225.2644 lb
	548.5134	199.6424
	- 6.9758	-39.5617
3	541.5376 lb	160.0807 lb

Cruise

Layshaft No.	Tangential	Separating
	287.3436	104.5844
	+37.7493	+13.7396
1	325.0929 lb	118.3220 lb

Cruise (cont)

Layshaft No.	Tangential	Separating
	287.3436	104.5844
	-30.7735	+25.8220
2	256.5701 lb	130.4064 lb
	287.3436	104.5844
	- 6.9758	-39.5617
3	280.3678 lb	65.0227 lb

Output Helical Gear Thrust

Takeoff

hp = 654.3271

T = Thrust

$$T = \frac{\text{hp}\ (63025)(\tan \text{helix angle})}{\text{rpm}\ (\text{pitch radius})}$$

$$T = \frac{654.3271\ (63025)(0.21256)}{24000\ (0.75)}$$

T = 486.9780 lb

Cruise

hp = 342.6734

$$T = \frac{342.6733\ (63025)(0.21256)}{24000\ (0.75)}$$

T = 255.0321 lb

Fan Rotor Assembly-Fanshaft Helical Gear Thrust Unbalance

Takeoff

Fanshaft helical gear thrust, lb	486.978
Fan thrust, lb	-400.000
	86.978

Cruise

Fanshaft helical gear thrust, lb	255.0321
Fan thrust, lb	-200.0000
	55.0321

Fanshaft Bearings

Calculation of Fanshaft Bearing Dynamic Capacity

AFBMA formula (for balls smaller than 1-in. diameter):

$$C = f_c\,(i \cos\alpha)^{0.7}\, Z^{2/3}\, D^{1.8}$$

where

i = number of rows of balls, 1

α = contact angle, $\cos\alpha = 0.90631$ for 25-degree contact angle

Z = number of balls, 19

D = ball diameter, 0.3125 in.

d_m = pitch diameter, 2.126 in.

f_c = factor = function of $D \cos\alpha / d_m$

C = basic load rating for 1 million revolutions

C = 4419.5 (0.93345)(7.120)(0.12323)

C = 3620 lb

Fanshaft Duplex Bearing Axial and Radial Loads

Axial Loads

Takeoff

	Fwd bearing	Rear bearing
Preload	150.0000	150.0000
Thrust unbalance	+86.9780	-86.9780
	236.9780 lb	63.0220 lb

Cruise

	Fwd bearing	Rear bearing
Preload	150.0000	150.0000
Thrust unbalance	+55.0321	-55.0321
	205.0321 lb	94.9679 lb

Radial Loads

Assumptions:

Fanshaft and rotor assy weight 12.3 lb

Axial dimension from center of gravity of fanshaft and rotor assy to intersection of rear bearing 25-degree contact angle line and shaft horizontal centerline 3.55 in.

Calculated effective bearing span =

0.9914 +0.5 (spacer) +0.5906 (bearing width) = 2.082 in.

Fwd bearing	Rear bearing
$12.3\left(\frac{3.550}{2.082}\right) = 20.9726$ lb	$20.9726 - 12.3 = 8.6726$ lb

Fanshaft Bearings Equivalent Load

$$P = XVF_r + YF_a$$

where

X = a radial factor

V = a rotation factor

Y = a thrust factor

Analyzing the bearings as individual bearings where $F_a/VF_r > e$, e = 0.68 for a 25-degree contact angle:

$X = 0.41$

$Y = 0.87$

$V = 1.0$

Takeoff

Fwd bearing

$$P = 0.41\,(20.97) + 0.87\,(236.978) = 214.768 \text{ lb}$$

Rear bearing

$$P = 0.41\,(8.67) + 0.87\,(63.022) = 58.385 \text{ lb}$$

Cruise

Front bearing

$$P = 0.41\,(20.97) + 0.87\,(205.0321) = 186.976 \text{ lb}$$

Rear bearing

$$P = 0.41\,(8.67) + 0.87\,(94.968) = 86.178 \text{ lb}$$

Fanshaft Bearing Duty Cycle Loads

Front bearing

$$\left[(FSTOF)^3\ 0.15 + (FSCRF)^3\ 0.85\right] = 191.5757 \text{ lb}$$

Rear bearing

$$\left[(FSTOR)^3\ 0.15 + (FSCRR)^3\ 0.85\right] = 83.0634 \text{ lb}$$

where

FSTOF, FSTOR, FSCRF, FSCRR = Front and rear fanshaft bearing equivalent radial load at takeoff and cruise and 0.15 and 0.85 factor
= percent of time at takeoff and cruise.

Fanshaft Bearing Life

Front bearing

$$\left(\frac{3620}{191.5757}\right)^3 \times \frac{16670}{15000} = 7591.64 \text{ L10 hr} = 12.640 \text{ L2 hr with matl factor of 5}$$

Rear bearing

$$\left(\frac{3620}{83.0634}\right)^3 \times \frac{16670}{15000} = 93138 \text{ L10 hr} = 155,075 \text{ L2 hr with matl factor of 5}$$

These bearing lives are for 12-degree helix angle. Figure 10 shows fanshaft bearing life over a range of fan output gear helix angles.

Gyroscopic Loads and Bearing Life

I, fan rotor assy (slug ft^2)	= 0.032
rpm, 15000×1.10	= 16,500
Ω(radians/sec)	= 2.0
T, torque (lb-ft) = $I\,\omega\,\Omega = 0.032\,(1727.88)(2)$	= 110.648 lb-ft
Rotor CG to intersection rear bearing contact angle and rotor horizontal centerline (in.)	= 3.55
e (effective bearing span) (in.)	= 2.082

Load front bearing (highest load) = $\dfrac{110.648 \times 12 \times 3.55}{2.082}$ = 2264 lb

$$\text{Life} = \left(\frac{3620}{2264}\right)^3 \left(\frac{5 \times 16670}{16500}\right) \times 0.32 = 6.6\ \text{L2 hr}$$

Fanshaft Bearing Life (Failed Fan Blade)

Fan blade weight (lb)	= 0.15
CG blade, rotor horizontal centerline (in.)	= 4.86
rpm	= 15,000
CG blade, intersection of rear bearing contact angle line with rotor horizontal centerline (in.)	= 3.550
e (effective bearing span) (in.)	= 2.082

$$\text{CF blade} = \frac{W}{g}\,(\bar{R})\,(\omega)^2$$

$$\omega = \frac{2\,\pi\,N}{60}$$

$$\text{CF} = \frac{0.15}{32.2}\left(\frac{4.86}{12}\right)\left(\frac{2\,\pi \times 15000}{60}\right)^2 = 4655\ \text{lb}$$

$$\text{Front bearing load} = 4655\left(\frac{3.55}{2.082}\right) = 7937.2 \text{ lb}$$

$$\text{Bearing life} = \left(\frac{3620}{7937.2}\right)^3 \left(\frac{5 \times 16670}{15000}\right) = 31.6 \text{ min}$$

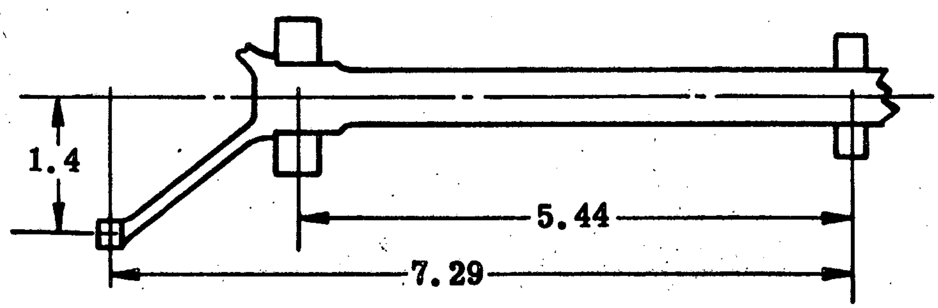
Layshaft Bearings

Figure 11 shows a schematic of a layshaft cluster gear with the tangential, separating, and thrust loads imposed on the gears. It should be noted that the separating gear loads are in the same direction while the tangential gear loads are in the opposite direction and that the helical gear thrust couple load is additive to the roller bearing separating load and subtractive from the ball bearing separating load. Gear loads are transferred to the bearings in the conventional manner. Table IX shows the layshaft roller and ball bearing loads and life. The curve labeled "LS" in Figure 12 shows that 12.15-degree helix angle is optimum for maximum layshaft ball bearing life; however, 12 degrees is considered maximum since the ball bearing life deteriorates if the fan thrust or other factors entering into bearing optimization are in error. Above 12.15-degree helix angle, the equivalent radial load is greater than the radial load, resulting in decreased ball bearing life (above 12.15 degrees for 20-degree ball bearing contact angle, the equivalent radial load = 0.43 radial load plus axial thrust load). Figures 12 and 13 show that minor changes in layshaft span dimensions appreciably improve layshaft bearing life and should be considered for improved endurance and growth potential. These figures were completed too late for incorporating the span advantage into the existing program.

Accessories Drive Bearings

Accessories Drive Shaft Bearing Loads and Life

The accessories drive shaft bearing and gear span dimensions are as shown in the following sketch.



6556-15

$$W_t = 52.52\left(\frac{7.29}{5.44}\right) = 70.38 \text{ lb}$$

$$W_r = 19.11\left(\frac{7.29}{5.44}\right) = 25.61 \text{ lb}$$

$$P = (W_t^2 + W_r^2)^{1/2} = 74.895 \text{ lb}$$

$$T = 19.11\left(\frac{1.4}{5.44}\right) = 4.918 \text{ lb}$$

$$\frac{F_a}{VF_r} = \frac{4.918}{74.895} = 0.0656 \qquad \frac{F_a}{C_o} = \frac{4.918}{1000} = 0.00492$$

$$\therefore X = 1, \; Y = 0$$

No. 5 Bearing

$$\text{L2 life (hours)} = \left(\frac{1650}{74.895}\right)^3 \times \left(\frac{3 \times 16670}{12000}\right) \times 0.32 = 14,260 \text{ hr}$$

No. 6 Bearing

$$W_t = 52.52\left(\frac{1.85}{5.44}\right) = 17.86 \text{ lb}$$

$$W_r = 19.11\left(\frac{1.85}{5.44}\right) = 6.499 \text{ lb}$$

$$P = (W_t^2 + W_r^2)^{1/2} = 19.00 \text{ lb}$$

$$\text{L2 life (hours)} = \left(\frac{965}{19.00}\right)^3 \times \left(\frac{3 \times 16670}{12000}\right) \times 0.32 = 174,722 \text{ hr}$$

Gasifier Bearing

Gasifier Bearing Radial Load

$$\text{Tangential load } (W_t) = 52.52\left(\frac{1.8}{6.4}\right) = 14.77 \text{ lb}$$

$$\text{Radial load } (W_r) = \frac{W_t}{\text{cos press. angle}} = \frac{14.77}{\cos 20°} = 15.72 \text{ lb}$$

CRITICAL SPEEDS

The critical speeds and response to unbalance of the fan rotor assembly were calculated utilizing the Allison digital computer program BB60. In the computer program, the fan rotor assembly was divided into a number of discrete elements. The stiffness, mass, pitch inertia, and gyroscopic characteristics of each element were defined. Matrix techniques were used to compute the calculations. The scope of the representation included the fanshaft back to the 48-tooth helical power output gear. The shaft was represented as being supported by a spring at each of the bearings. A conservative spring rate of 500, 000 lb/in. was used in this application.

Figure 15 shows plots of the three lowest critical speeds with a 0. 300-in. spacer between the 25-degree contact angle preloaded angular contact ball bearings (effective bearing span = 1. 875 in.). The final design effective span was increased to 2. 0875 inches (0. 500-in. spacer). Figure 16 shows the lower critical speed to be 28, 000 rpm with this span. The higher critical speeds were not calculated since they would be out of the operating range.

Figure 17 shows the calculated response for a fan wheel and blade unbalance of 0. 02 lb-in. Maximum predicted response is approximately two mils at 15, 000 rpm. This response is quite low for rotor motion and has been calculated for extreme conditions of unbalance and dampening.

HEAT REJECTION AND LUBRICATION SYSTEM

A computer program available at Allison was used to calculate the bearing friction heat generated. This program takes into account bearing geometry and oil properties at temperature, speed,and load. It calculates heat generated due to rolling and sliding for rolling element bearings and load, viscous, and spin heat generated for ball bearings. Gear losses were assumed to be 0. 5% per mesh. The gear loss was increased an additional 5% to take care of possible windage losses. The heat rejection losses are shown in Table X.

The circulating and dry sump-type lubrication system is designed for an external reservoir and heat exchanger. It is capable of furnishing adequate lubrication, scavenging, and cooling as needed for bearings, splines, and gears at all operating and flight conditions.

The system is designed to use MIL-L-7808 oil at 60 psig and at 185°F oil inlet temperature. Jet lubrication is provided to the fanshaft, layshaft, and gasifier thrust bearings and to the out-of-mesh side of the gears. Accessories drive bearings and gears are lubricated by oil thrown off the jet-lubricated components. Pressure oil at the reduction gear and inlet housing/accessories drive gearbox housing interface is conveyed to a manifold assembly within the layshaft gear and bearing support housing via a transfer tube.

The power input spline and gasifier thrust bearing are lubricated through the hollow power input shaftgear by means of an orifice located on the aft side of the oil manifold assembly. Two orifices on the OD of the manifold assembly tube supply oil to the out-of-mesh side of the sun gears and the No. 2 layshaft gears. Four orifices located on the manifold laminated sheet metal structure supply oil to the remaining gear meshes. The forward end of the manifold tube supplies oil to an annulus surrounding the fanshaft bearing outer spacer. Two orifices in the spacer meter oil to the fanshaft bearings. Three equally spaced holes from the annulus distribute oil to the layshaft bearings.

The oil is scavenged from one of the four hollow struts and, depending upon the plane/engine installation, is picked up either externally at the lower strut or at the accessories drive pad interface. A schematic of the engine lubrication system is shown in Figure 20. The oil distribution requirements are listed in Table XI.

SHAFTS, SPLINES, AND SEALS

The torsional shear stress of the power input shaft, layshaft, and power output shafts were calculated by the following formula:

$$S_s = \frac{16\ T\ D_o}{\pi(D_o^4 - D_i^4)}$$

where

T = torque, lb-in.

D_o = outside diameter, in.

D_i = inside diameter, in.

The spline crushing stress was calculated by the following formula:

$$S_c = \frac{2\ T}{(PD)^2\ L}$$

where

T = torque, lb-in.

PD = spline pitch diameter, in.

L = spline length, in.

Table XII shows the horsepower, torque, allowable torsional shear strength, stress at design torque, limit torque, ultimate torque, and margin of safety.

Seals

The seals are straight labyrinth type of conventional design. A "blow-down" seal may be required to prevent oil leakage under flight operating conditions.

COST ANALYSIS

Introduction

The fan drive reduction gear program required that reliability and cost be a first consideration and that weight be secondary. Therefore, value engineering cost studies were applied to each component of the three-layshaft design selected for design detailing in order to arrive at the lowest cost while retaining total reliability. Design trade-off studies, new processes, materials, standard parts, quality control, etc, were all considered in arriving at the lowest cost configuration.

Team Effort

In order to assist in decision making, Design and Value Engineering utilized the knowledge and assistance in team efforts of personnel from Metallurgy, Procurement, Processing, Reliability, Tooling, and Manufacturing departments as well as from outside specialty vendors.

The use of sleeve bearings vs antifriction bearings (ball and roller) was studied. A definite cost advantage would be possible with the use of sleeve bearings, such as Clevite, if the reliability were assured. However, a program of design, build, test, and redesign would be necessary to get the right lubrication system with a series of sleeve bearings in the design. After due consideration, NASA and Allison engineers agreed that because of high frictional power losses, starting torque limitations, and possible oil interruption, only ball and roller bearings would be used, thus ensuring cold weather reliability.

The materials to be used for gears and shafts in this application were also studied:

1. AMS-6260 carburizing

2. AMS-6381/6415/6470/6370 nitriding

3. AMS-6431 D6 vacuum melt

4. EMS-65040 maraging

The cost of the carburizing and nitriding materials was determined to be essentially the same; that of the D6 material was three times as much; and that of the maraging material was ten times as much. Further, the machinability (SAE 1112=100) of the carburizing and nitriding materials in the core-treated condition is "50" as compared with "6" for D6 and maraging steels. Since the higher core hardness and strength were not required in this application, only the nitriding and carburizing steels were considered. AMS-6370 (SAE 4130) material was selected for nitriding because it was less subject to chipping and had a sufficient case hardness (15N88 min) to support the required loads. AMS-6415 (SAE 4340) does not have sufficient case hardness and AMS-6470 is more susceptible to chipping. AMS-6260 carburizing material was not selected because of the added labor cost of grinding after hardening and because some parts could not be ground after hardening.

The use of aluminum castings or nodular iron castings for the gear housing was also evaluated. It was determined that the cost of the aluminum castings would be higher, that they would require bearing cages which would add to the cost, and that concentricity and fit controls would add producibility and reliability problems due to the operating temperature spread and difference in the coefficient of expansion of the two metals. Therefore, it was decided that both improved reliability and reduced cost could well be exchanged for the increased weight.

Cost Estimates

Following is a summary of value engineering cost estimates based on 2000 units per year at a labor rate of $12.50 per hour:

Raw material	$ 30.24
Purchased parts—Miscellaneous	22.89
Bearings	143.67
Estimated standard hours—30.1	376.40
Total	$573.20

This estimate is $387 less than that of an earlier layshaft design described in Appendix A. Of this reduction, $179 is in the cost of bearings. This was made possible through quotes from a new source, elimination of two bearings, and by changing three layshaft roller bearings to ball bearings. Other reductions were made by design trade-offs plus quotes from vendors for standard parts. Build and test of this unit will result in further reliability, quality, and value improvements. Detailed estimated cost data are shown in Table XIII.

The cost of the inlet housing, based on size, and indicated complexity, is estimated at $160.00 for the aluminum casting and $325.00 for machining (26 hours at $12.50/hr).

ENGINE MOUNT PADS AND ACCESSORY GEARBOX PAD LOADS AND STRESSES

Engine mount loads based on MIL-E-5007C and representative of the most severe conditions for right-side-mounted and bottom-mounted engines were furnished to Allison per NASA drawing CDL 11270. Figure 21 shows the appropriate features of this drawing.

The stresses are a maximum in the fore-aft direction on the upper mount pad of the right-hand engine installation (Case II) and on the right-hand pad of the bottom-mounted engine (Case III). The maximum moment that can be supported by the accessories drive pad is 355 lb-in. The engine mount and accessories drive pad stresses are shown in Table XIV.

TABLE I—COMPARISON OF GEAR ARRANGEMENTS.

	Internal	Planetary	Layshaft
Cost	Medium	Poor	Good
No. parts	Good	Medium	Medium
Noise and vibration	Poor	Medium	Good
Growth potential	Poor	Medium	Good
Offset inlet duct and shafting	X	---	---
Type bearings	Antifriction	Sleeve and Antifriction	Antifriction

TABLE II—OBJECTIONS TO HYDRODYNAMIC BEARINGS.

Oil interruption
Cavitation—erosion development
Starting torque (6-hp starter limit)
Heat rejection/cooling requirements—fuel cooling
Deflection and misalignment control

TABLE III—PROGRAM SCOPE.

Two Phase Program

Phase I	Phase II
Coordination of design requirements	Refine selected design
Studies of reduction gear arrangements	Value engineering trade-off studies of selected configuration
Value engineering cost comparison studies	Mount stress analysis
Selection of optimum arrangement	Detailing
Layout of optimum arrangement	Final report

TABLE IV—WEIGHT SUMMARY.

Part number	Title	Weight (lb)
6873812	Housing, Reduction Gear & Inlet	31.00
6875363	Cage, Bearing	0.30
6875364	Cage, Bearing	0.26
6873814	Housing, Gear & Bearing Support	16.20
6873818	Gearshaft, Helical Fan	1.50
6875360	Bearing, Ball Duplex (40 X 68 X 30)	0.90
6873819	Spacer Assy, Bearing (Matched set)	0.32
6873820	Seal, Labyrinth Rotating	0.23
6875357	Nut, Spanner	0.08
6875358	Ring, Lock	0.0025
6873816	Seal, Labyrinth Stator	0.242
6873821	Plate, Bearing—Retaining	0.120
6873817	Gearshaft, Cluster—Layshaft	4.430
6873815	Plate, Bearing Thrust	1.000
6875361	Bearing, Ball (15 X 42 X 13)	0.540
6873822	Manifold Assy, Oil Transfer	0.310
6875362	Bearing, Ball (15 X 32 X 9)	0.068
6873824	Gearshaft, Face—Accessory Drive	0.966
6875502	Bearing, Ball (17 X 40 X 12)	0.150
6873823	Gearshaft, Spur Power Input	1.144
6873825	Tube, Oil Transfer—Pressure	0.012
6875359	Bearing, Roller Cylinder (20 X 42 X 12)	0.570
	Total	60.344

TABLE V—COMPONENT MATERIALS.

Part number	Title	Material
6873812	Housing, Reduction Gear & Inlet	AMS 4215 Aluminum casting
6875363	Cage	AMS 6370
6875364	Cage	AMS 6370
6873814	Housing, Gear & Bearing Support	AMS 5316 Nodular iron casting
6873818	Gearshaft, Helical Fan	AMS 6370
6873819	Spacer Assy, Bearing	AMS 6322
6873820	Seal, Labyrinth—Rotating	AMS 6360
6875357	Nut, Spanner	AMS 6370
6875358	Ring, Retaining	AMS 5688
6873816	Seal, Labyrinth—Stator	AMS 6360
6873826	Shim	AMS 5510
6873821	Plate, Bearing—Retaining	AMS 6350
6873824	Gearshaft, Face—Accessory Drive	AMS 6370
6875503	Shim	AMS 5510
6873823	Gearshaft, Spur—Power Input	AMS 6370
6873825	Tube—Oil Transfer Pressure	AMS 4070
6873817	Gearshaft, Cluster—Layshaft	AMS 6370
6873822	Manifold Assembly—Oil	AMS 5513
6873815	Plate, Bearing Thrust	AMS 6350

TABLE VI—BEARING GEOMETRY AND AFBMA CAPACITIES.

Position No.	Size mm	No. of rolling elements	Size of rolling elements	Contact angle	AFBMA static capacity C_o (lb)	AFBMA dynamic capacity C (lb)
1	40 × 68 × 15	19	0.3125	25°	3260	3620
2	40 × 68 × 15	19	0.3125	25°	3260	3620
3	20 × 42 × 12	9	0.2756 × 0.2756	---	2067	3664
4	15 × 42 × 13	10	0.34375	20°	1875	2616
5	17 × 40 × 12	8	0.2656	---	1000	1650
6	15 × 32 × 9	9	0.1875	---	565	965

TABLE VII—LAYSHAFT GEAR DATA.

Position No.	No. teeth	Diametral pitch		Helix		Pitch dia (in.)	Face width (in.)		Pitch line velocity (ft/min)	Profile contact ratio		Face contact ratio		Bending stress* (psi)	Crushing stress* (psi)	Pressure angle
		Plane of rotation	Norm	Angle	Hand		Min	Max		Min	Max	Min	Max			
1	36	20	20	0	---	1.8	0.870	0.890	13195	1.320	1.857	---	---	14415	127,248	20°
2	42	20	20	0	---	2.1	0.740	0.780	13195	1.320	1.857	---	---	16309	127,248	20°
3	30	20	20.4468	12°	RHCW	1.5	1.075	1.115	9425	1.261	1.793	1.373	1.428	17508	128,829	20°
4	48	20	20.4468	12°	LHCCW	2.4	1.015	1.055	9425	1.261	1.793	1.373	1.428	16377	128,829	20°

*Duty cycle stresses for single tooth contact.
Bending stresses are for beginning of single tooth contact.
Crushing stresses are for operation at pitch point.

No. teeth	Helix angle	Bending stress (psi)		Crushing stress (psi)	
		TO	CR	TO	CR
36	---	22242	11652	158,066	114,405
42	---	25165	13183	158,066	114,405
30	12°	27020	15740	160,040	115,817
48	12°	25274	14723	160,040	115,817

TO = Takeoff Power
CR = Cruise Power

TABLE VIII—COMPARISON OF THREE LAYSHAFT GEAR STRESS LEVELS WITH CURRENT AIRCRAFT PRACTICE.

Design			Duty Cycle		
	BSTC	PP (OP)		BSTC	PP (OP)
Three layshaft (first reduction)			**Three layshaft (first reduction)**		
hp = 219.318			hp = 142.134		
Bs (psi)	22,242-25,165		Bs (psi)	14,415-16,309	
Cs (psi)		158,066	Cs (psi)		127,248
Three layshaft (second reduction)			**Three layshaft (second reduction)**		
hp = 218.109			hp = 141.332		
Bs (psi)	25,274-27,020		Bs (psi)	16,377-17,508	
Cs (psi)		160,040	Cs (psi)		128,829
Typical engine No. 1			**Typical engine No. 1**		
hp = 5,000			hp = 2,360		
Bs (psi)	31,300		Bs (psi)	14,774	
Cs (psi)		160,000	Cs (psi)		109,924
Typical engine No. 2			**Typical engine No. 2**		
hp = 250			hp = 188		
Bs (psi)	16,500		Bs (psi)	12,380	
Cs (psi)		121,200	Cs (psi)		105,000
Typical engine No. 3			**Typical engine No. 3**		
hp = 317			hp = 227		
Bs (psi)	20,962		Bs (psi)	15,000	
Cs (psi)		136,687	Cs (psi)		115,600
Typical engine No. 4			**Typical engine No. 4**		
hp = 400			hp = 324		
Bs (psi)	24,000		Bs (psi)	19,450	
Cs (psi)		149,000	Cs (psi)		134,200

BSTC = beginning of single tooth contact
PP (OP) = pitch point (operating)
Bs = bending stress
Cs = crushing stress

TABLE IX—LAYSHAFT BEARING LOADS AND LIFE.

Size, mm	20 × 42 × 12			15 × 42 × 13		
Dynamic capacity, lb.	3664			2616		
Contact angle	---			20°		
rpm	24,000			24,000		
Layshaft No.	1	2	3	1	2	3
Thrust (takeoff), lb	---	---	---	← 162.326 →		
Radial load, lb	497.5	514.7	498.31	289.64	260.92	240.04
Equivalent radial load, lb	---	---	---	289.64	274.52	265.54
Thrust (cruise), lb	---	---	---	← 85.010 →		
Radial load, lb	257.7	274.8	259	164.74	137.31	115.66
Equivalent radial load, lb	---	---	---	164.64	144.06	134.74
Equivalent duty cycle load, lb	326	341.37	327	195.16	177.96	169.62
L2 life (hr)	3,677	3,154	3,634	2,785	3,674	4,242

12° helix angle Material factor = 5

TABLE X—HEAT REJECTION (BTU/HR).

	Bearing No.				Gear mesh		Total
	1	2	3	4	1-2	3-4	
Takeoff	610	445	1,380	820	8,792	8,741	20,788
Cruise	580	475	1,100	590	5,226	6,817	14,788

TABLE XI—OIL DISTRIBUTION.

Requirement	Jet diameter (in.)	No. jets	Total flow (lb/min)
	Pressure = 60 psig		
*No. 1 bearing	0.030	1	0.977
*No. 2 bearing	0.030	1	0.977
*No. 3 bearing	0.030	3	2.931
*No. 4 bearing	0.030	3	2.931
*No. 5 bearing & spline	0.030	1	0.977
**Mesh of No. 1 & 2 gears	0.039	3	4.884
**Mesh of No. 3 & 4 gears	0.039	3	4.884
			18.56

*Flow controlled by 0.030 diameter orifice size
**Based on 60°F oil temperature rise

TABLE XII—SHAFT AND SPLINE STRESSES.

	Power input shaft			Layshaft			Power output shaft			Power input spline		
Type torque	100%	150%	200%	100%	150%	200%	100%	150%	200%	100%	150%	200%
Horsepower	675						650			675		
Torque, lb in.	1519.35			572.76			2731			1519.35		
Allowable torsional shear strength, psi	62000	62000	62000	62000	62000	62000	62000	62000	62000	*15000	15000	15000
Stress, psi	22857	34286	45714	2976	4464	5952	5901.54	8851.5	11803	10804	16206	21608
Margin of safety	1.71	0.81	0.36	19.8	12.9	9.4	9.5	6.0	4.25	0.388	-0.075	-0.306

*Allowable crushing strength to prevent fretting

Material AMS 6370 26 Rc min

Tensile strength = 140,000 psi min
Yield strength = 124,000 psi min

TABLE XIII—VALUE ENGINEERING ESTIMATED COSTS.

Item No.	Part number	Quantity per unit	Description	Estimate cost ($) Material	Labor hr/piece	Labor hr/unit	$/piece at 3.70	$/unit at 3.70	$/piece at 12.50	$/unit at 12.50	Total at 3.70	Total at 12.50
1	6873826	1	Outer seal	0.65	0.80	0.80	2.96	2.96	10.00	10.00	3.69	10.65
3	6873818	1	Gear-out	5.00	2.95	2.95	10.92	10.82	36.88	36.88	13.92	39.88
4/6	6873814	1	Bearing housing	14.20	4.55	4.55	16.84	16.84	56.88	56.88	31.04	71.08
5	6873823	1	Gear-drive	2.00	1.72	1.72	6.36	6.36	21.50	21.50	8.36	23.50
8/9	6875360	2	Bearing-ball	29.92*	---	---	---	---	---	---	29.92	29.92
10	6873824	1	Face gear	2.25	3.00	3.00	11.10	11.10	37.50	37.50	13.35	39.75
11	6875502	1	Bearing-ball	0.49	---	---	---	---	---	---	0.49	0.49
12	6873820	1	Laby. seal	0.55	0.88	0.88	3.26	3.26	11.00	11.00	3.81	11.55
14	6875362	1	Bearing-ball	0.76	---	---	---	---	---	---	0.76	0.76
16	6873815	1	Retainer plate	0.75	0.50	0.50	1.85	1.85	6.25	6.25	2.60	7.00
18	6873816	3	Dual gear	4.50	4.00	12.00	14.80	44.40	50.00	150.00	48.90	154.50
19	6875361	3	Bearing-ball	12.50	---	---	---	---	---	---	37.50	37.50
20	6875359	3	Bearing-roller	25.00	---	---	---	---	---	---	75.00	75.00
21	6875357	1	Nut	0.19	0.62	0.62	2.29	2.29	7.75	7.75	2.48	7.94
22	6875358	1	Lock-spring	0.10	---	---	---	---	---	---	0.10	0.10
23	6873822	1	Oil tube assembly	0.75	0.95	0.95	3.52	3.52	11.88	11.88	4.27	12.63
24	6873825	1	Oil tube	1.25	---	---	---	---	---	---	1.25	1.25
25	6875364	1	Cage	0.34	0.55	0.55	2.04	2.04	6.88	6.88	2.38	7.22
26	6875363	1	Cage	0.15	0.55	0.55	2.04	2.04	6.88	6.88	2.19	7.03
27	6873821	1	Plate-retainer	0.31	0.12	0.12	0.44	0.44	1.50	1.50	0.75	1.81
28		1	Spacer-out	0.40	0.56	0.56	2.07	2.07	7.00	7.00	2.47	7.40
29		1	Spacer-in	0.20	0.36	0.36	1.32	1.32	4.50	4.50	1.52	4.70
	AN122720	3	Dowel	0.09							0.27	0.27
	MS9241-009	2	O-ring	0.06							0.12	0.12
	MS9241-011	2	O-ring	0.06							0.12	0.12
	MS9241-144	1	O-ring	0.12							0.12	0.12
	MS9241-357	1	O-ring	0.41							0.41	0.41
	MS21042-4	19	Locknut-CRES	0.04							0.76	0.76
	MS172238	3	Tab nut	1.34							4.02	4.02
	MS172203	3	Tab washer	0.50							1.68	1.68
	AN960-416	6	Washer	0.01							0.06	0.06
	RR-50S	4	Snapring	0.04							0.16	0.16
	RR-165S	3	Snapring	0.13							0.39	0.39
	MS9321-10	3	Washer	0.11							0.33	0.33
	6875514	13	Stud	0.50							6.50	6.50
	6875515	6	Stud	0.60							3.60	3.60
	6875513	4	Stud	0.75							3.00	3.00
										Totals	308.29	573.20

*One set.

TABLE XIV—ENGINE MOUNT PAD STRESSES.

Right hand installation

Case No.	1	2
	Mount stresses, psi	
Upper mount	15,900	20,300
Limit margin of safety	0.467	0.149
Lower mount	7,800	14,800
Limit margin of safety	1.99	0.576

Bottom installation

Case No.	3	4
	Mount stresses, psi	
Right hand mount	15,150	14,550
Limit margin of safety	0.540	0.604
Left hand mount	9,600	9,550
Limit margin of safety	1.43	1.44

Accessory drive pad

Case No.	1	2	3	4
Membrane stress, psi	1104	950	1450	2750
*Bending stress, psi	1670	4100	1650	800

*For 1-in. lever arm

$$\text{Critical lever arm} = \left[\frac{23,333-950}{4100}\right] = 5.46 \text{ in.}$$

AMS 4215 aluminum casting
Design T_S 35,000 psi min
Design Y_S 28,000 psi min
Limit T_S 23,333 psi min

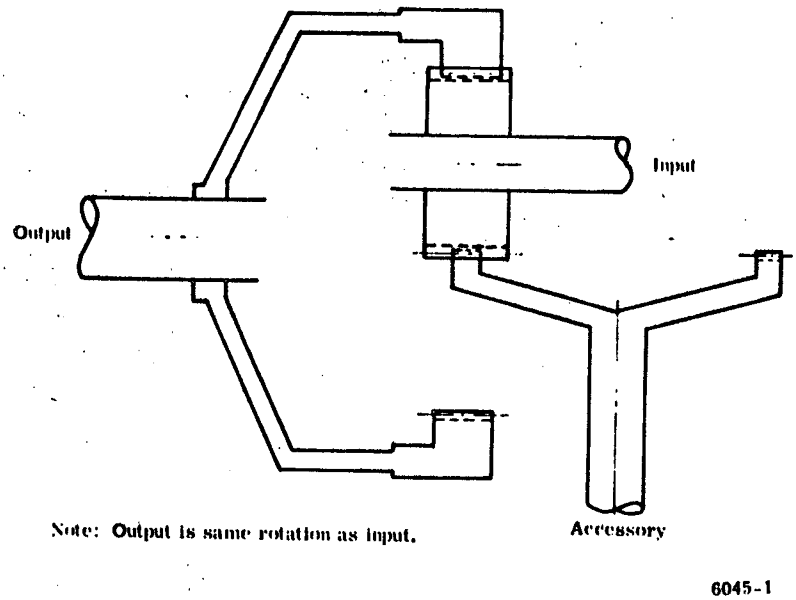


Figure 1. Single-stage external-internal reduction with offset shafting.

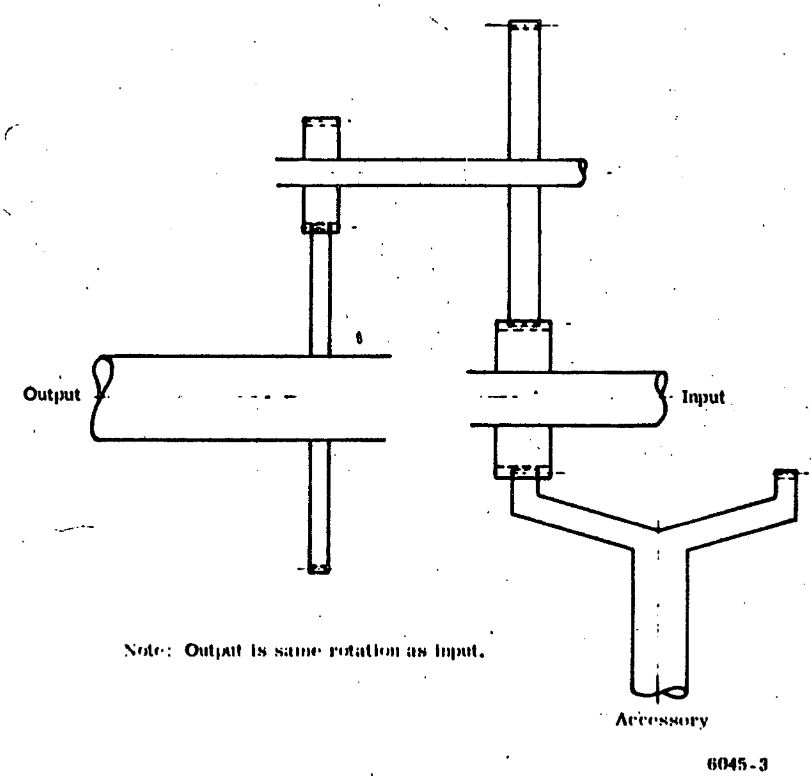


Figure 2. Multiple compound idler with concentric shafting.

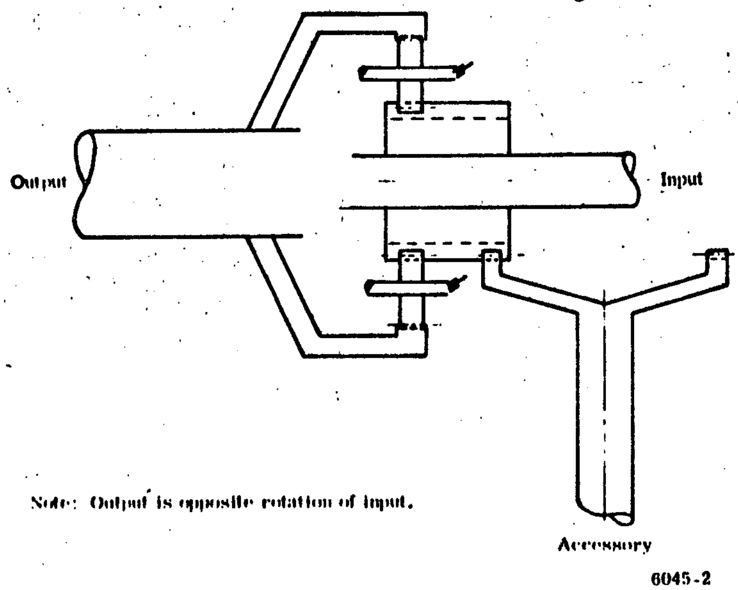


Figure 3. Star gear planetary system with ring gear.

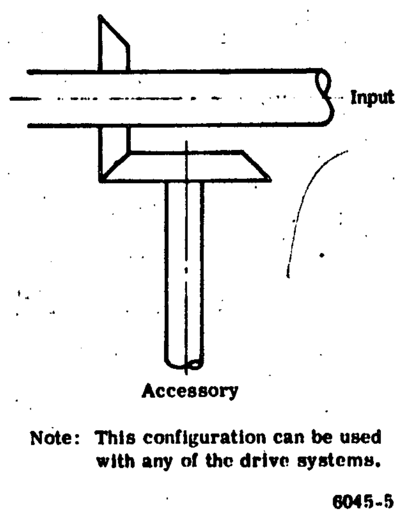


Figure 4. Optional bevel gear accessory drive.

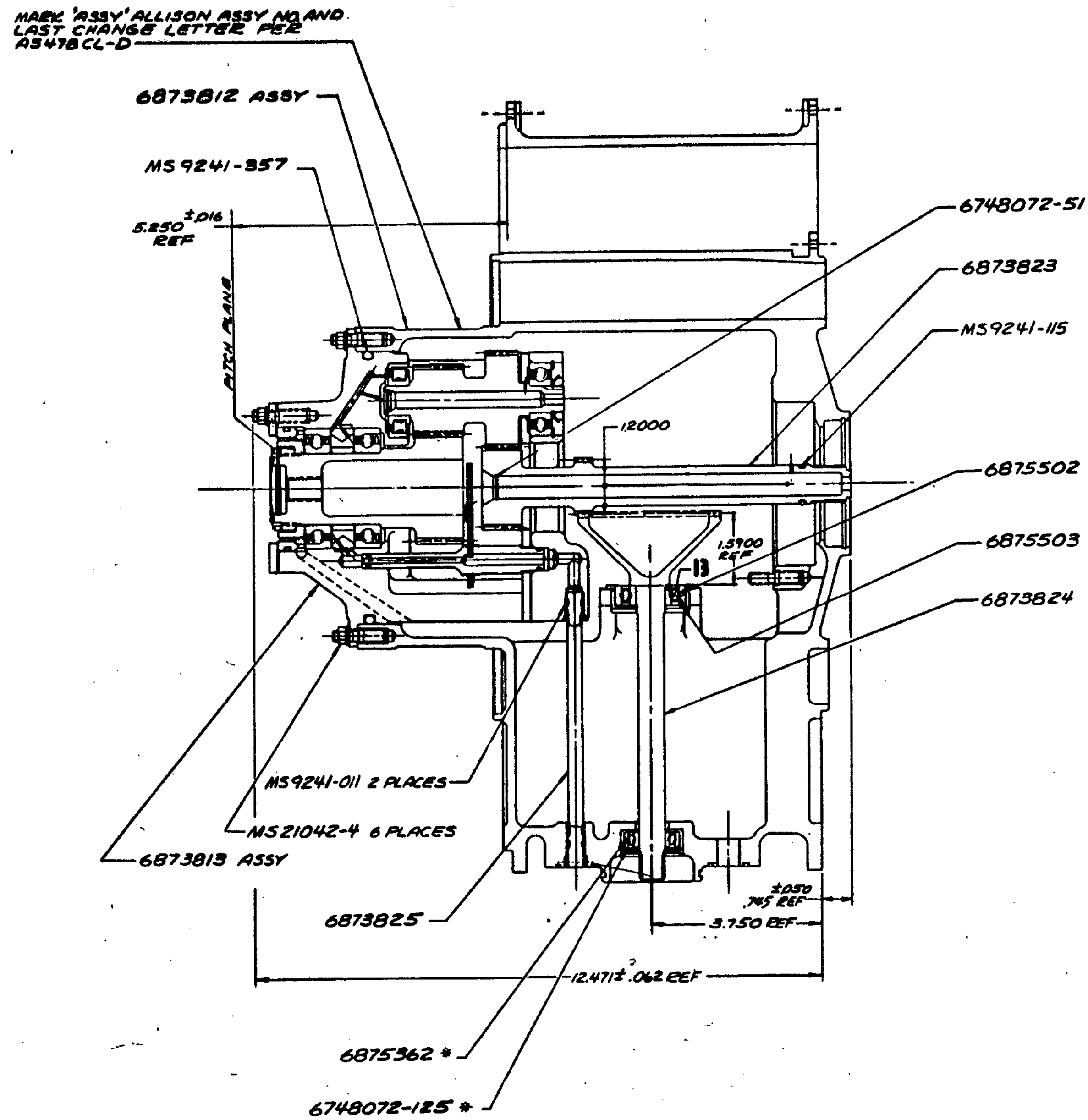


Figure 5. Cross section of layshaft design approved for detailing.

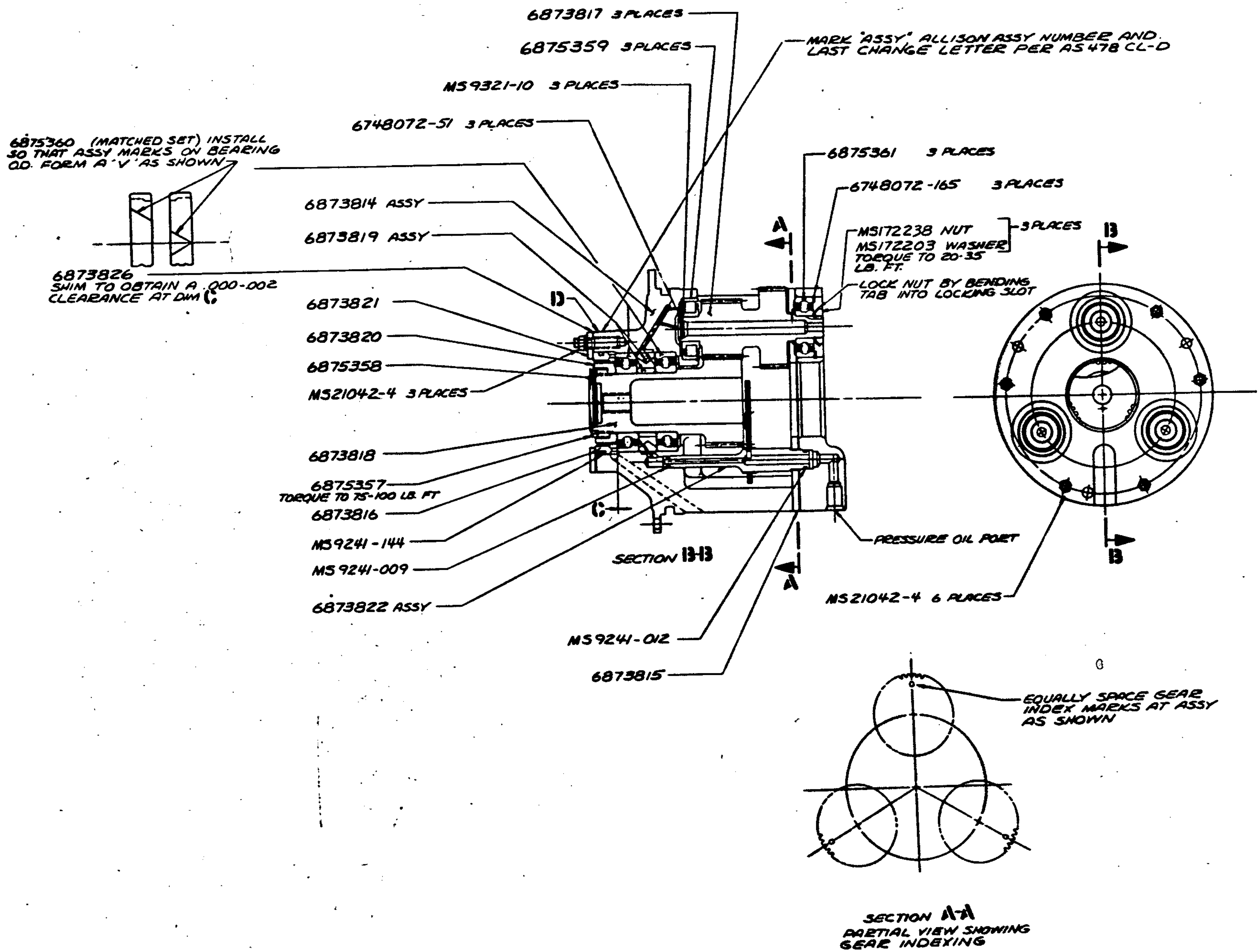


Figure 6. Cross section of layshaft gear and bearing support housing assembly.

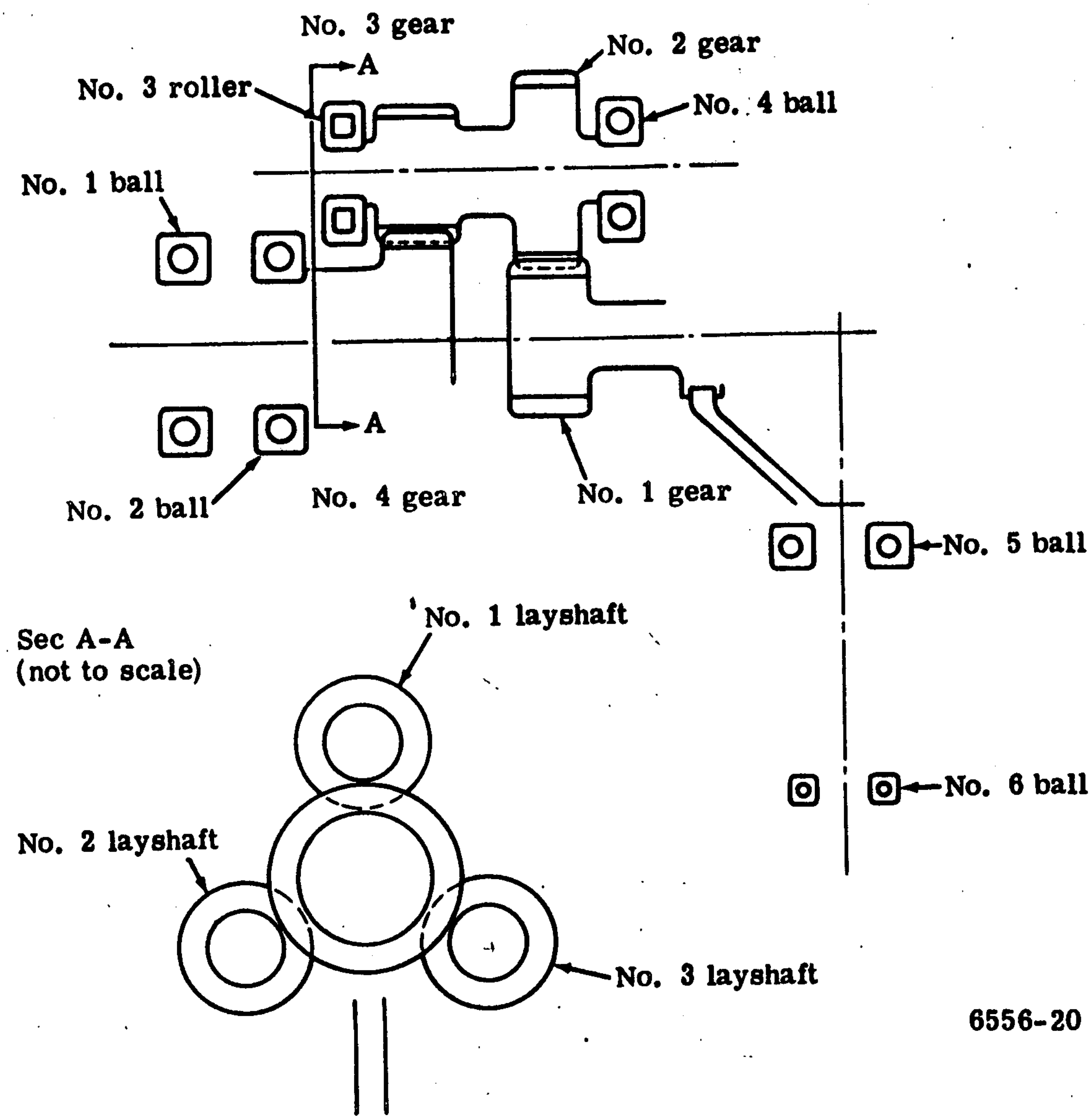


Figure 7. Bearing, gear, and layshaft positions.

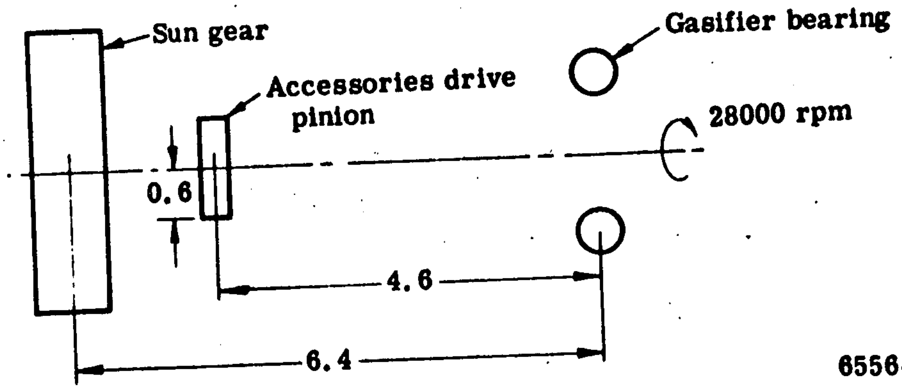
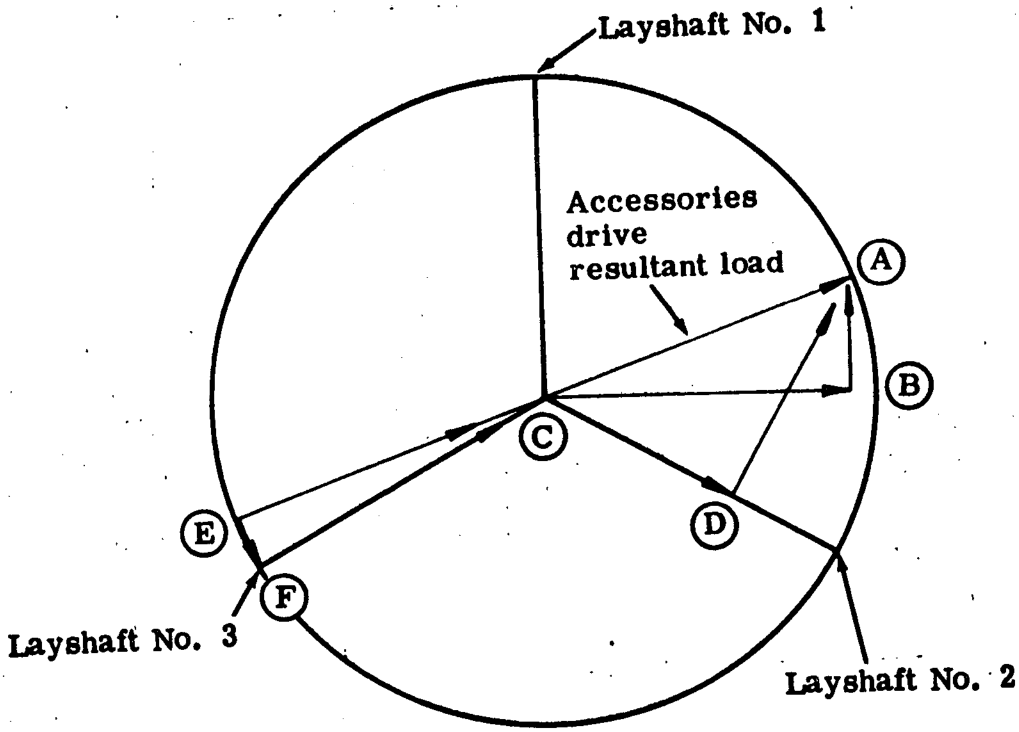


Figure 8. Power input gear, and accessories drive pinion gear axial dimensions.



AB = 13.7396
AD = 30.7735
CF = 39.5617

BC = 37.7493
CD = 25.8220
EF = 6.9758

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Figure 9. Vector loads.

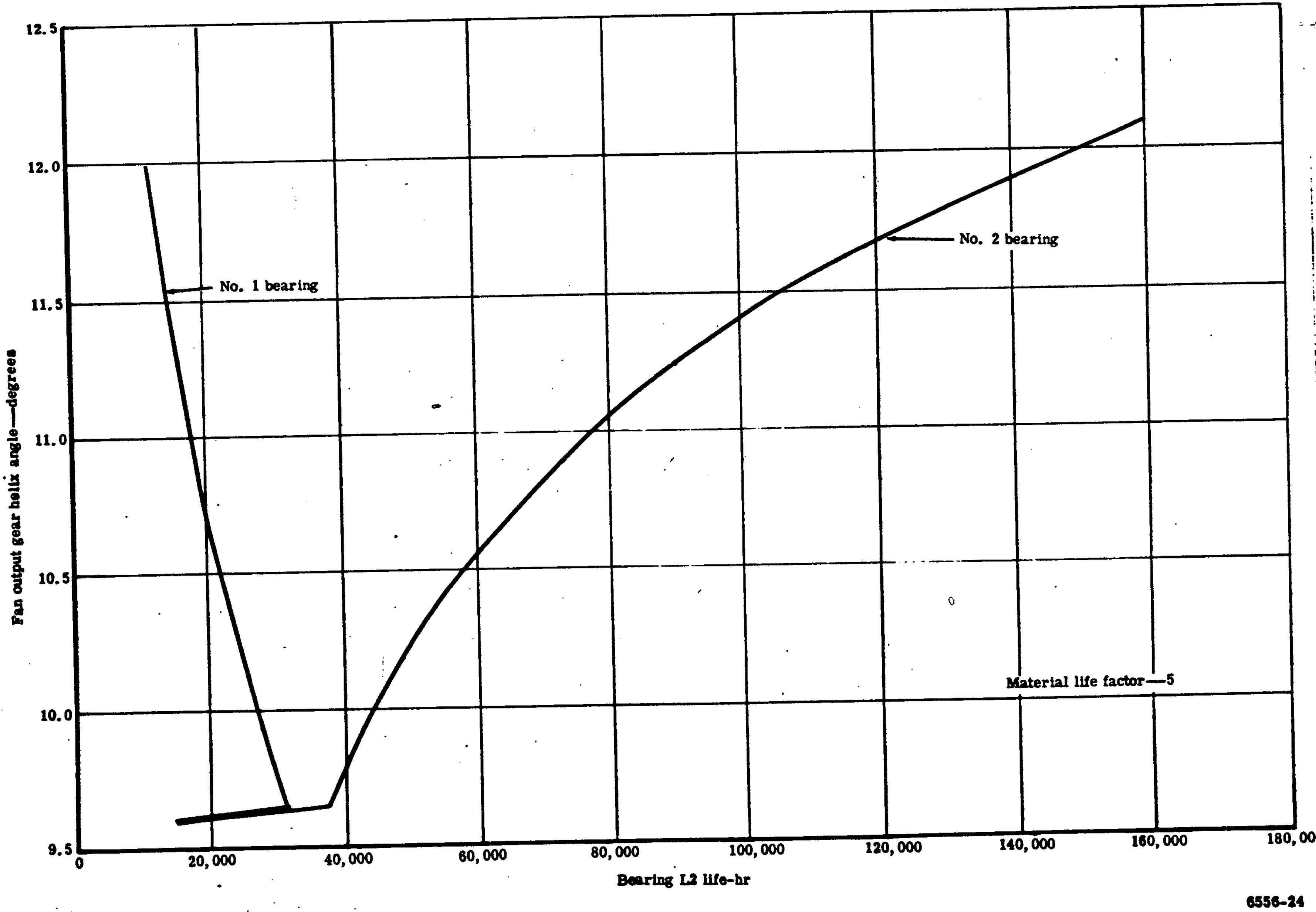

Figure 10. Fanshaft bearing life over a range of fan output gear helix angles.

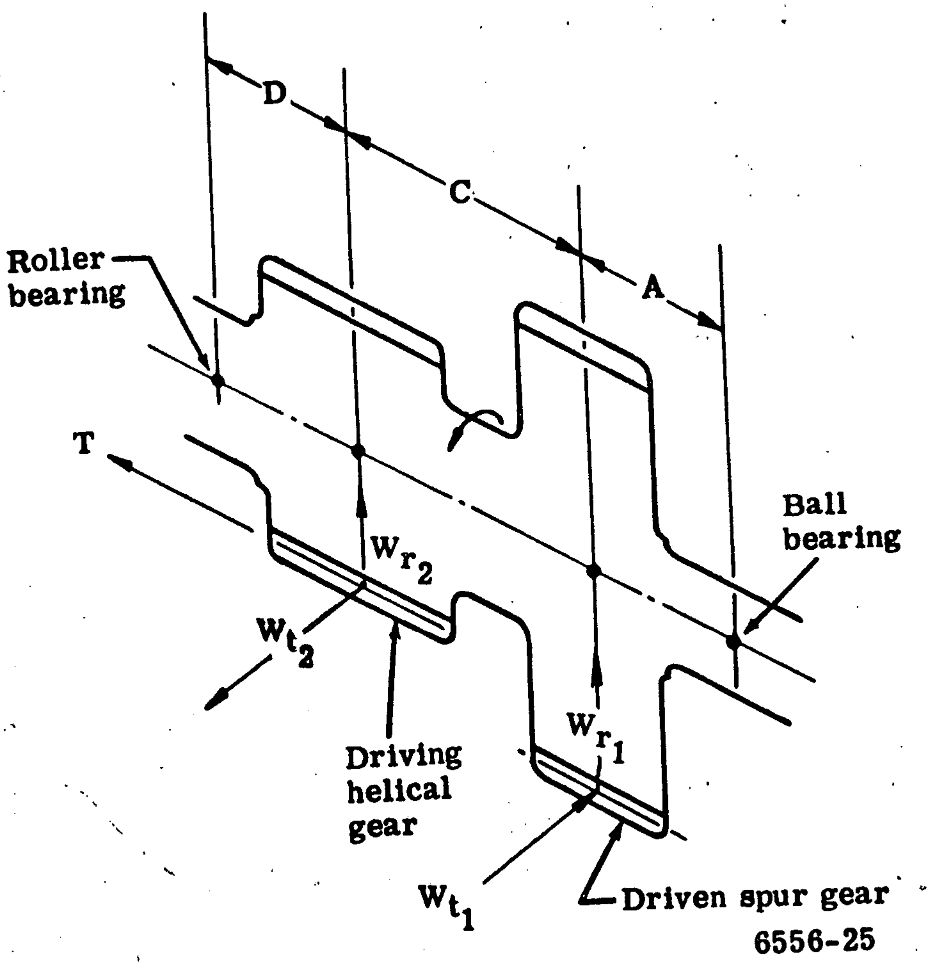

Figure 11. Layshaft cluster gear loads.

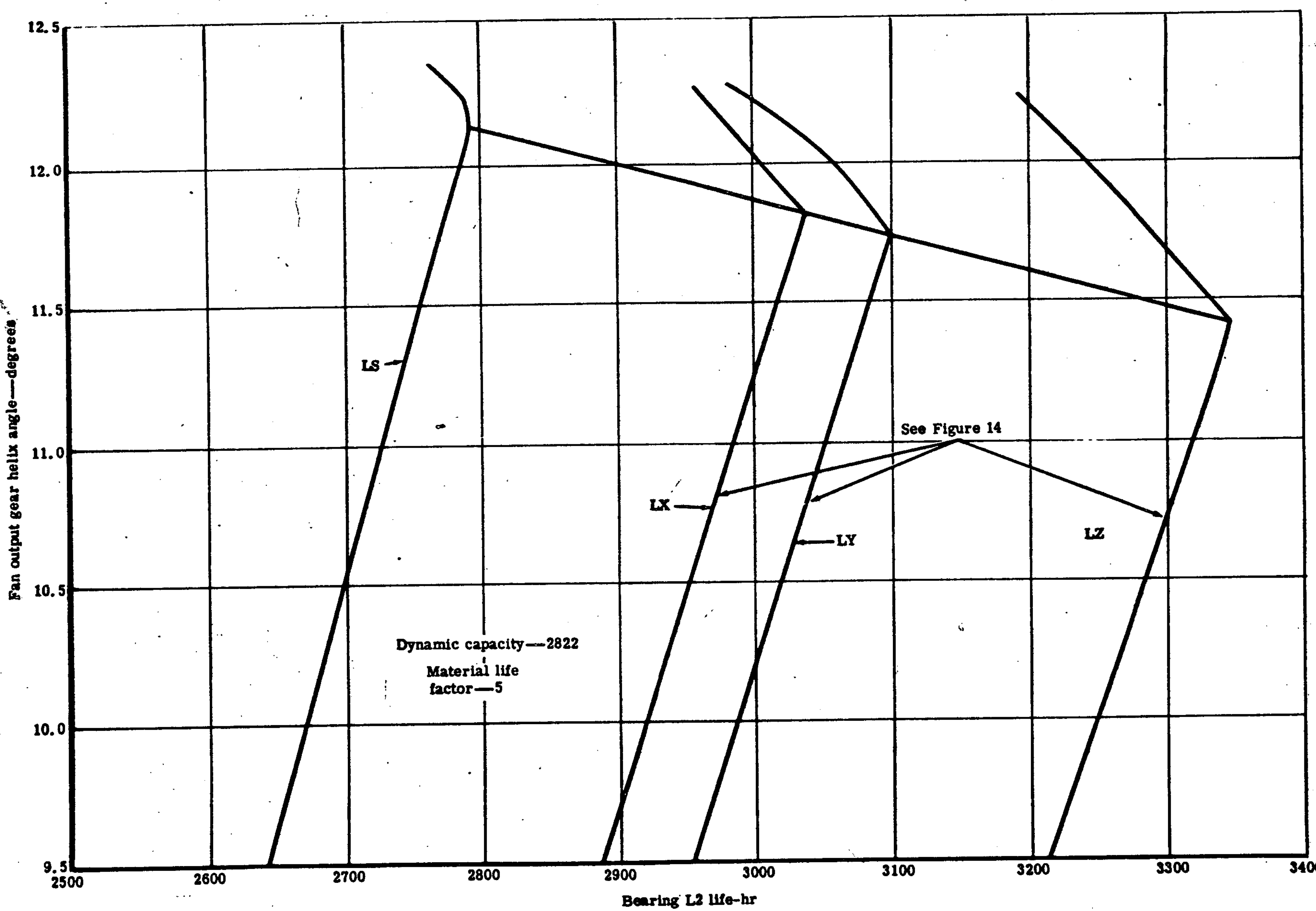

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Figure 12. Layshaft No. 1 ball bearing life.

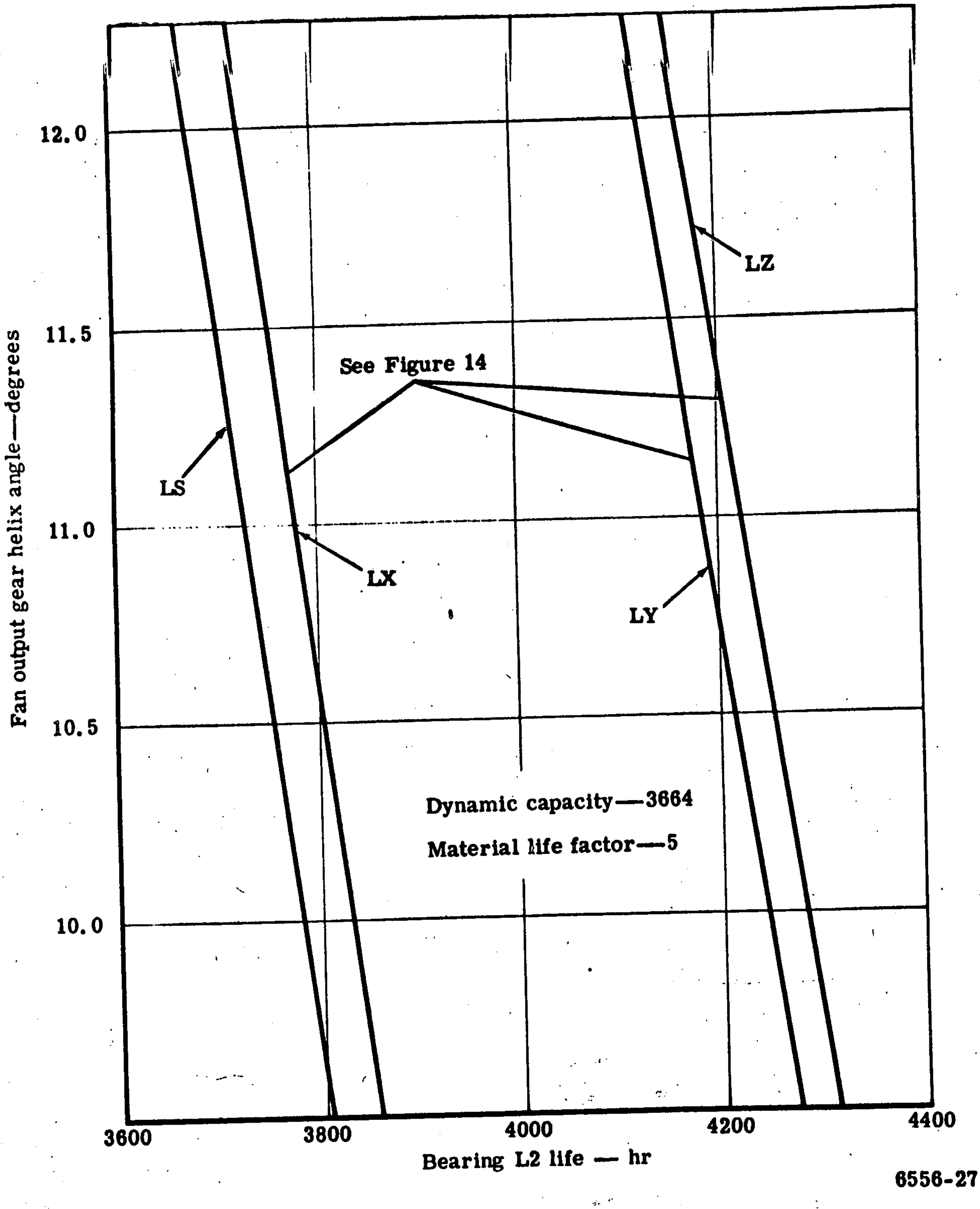


Figure 13. Layshaft No. 1 roller bearing life.

$A = 0.8596$
$A_1 = 0.9596$
$A_2 = 1.0596$

$B = 0.8374$
$B_1 = 0.9374$

$C = 1.4125$

See Figures 12 and 13

$LS = A + B + C$ (as detailed)
$LX = A_1 + B + C$
$LY = A_1 + B_1 + C$
$LZ = A_2 + B_1 + C$

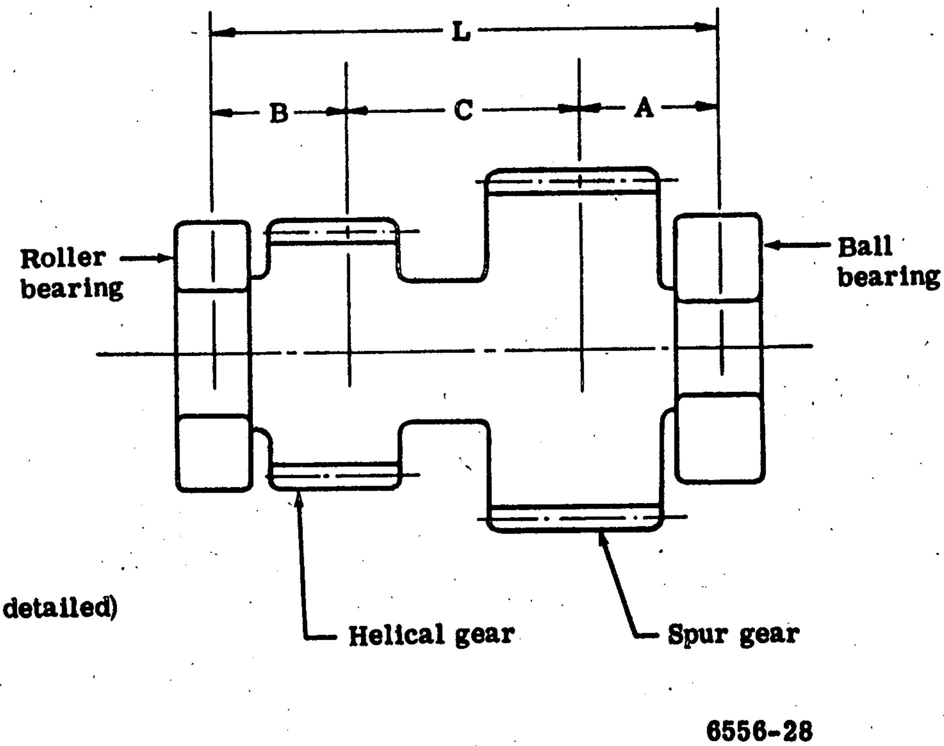


Figure 14. Layshaft span dimensions.

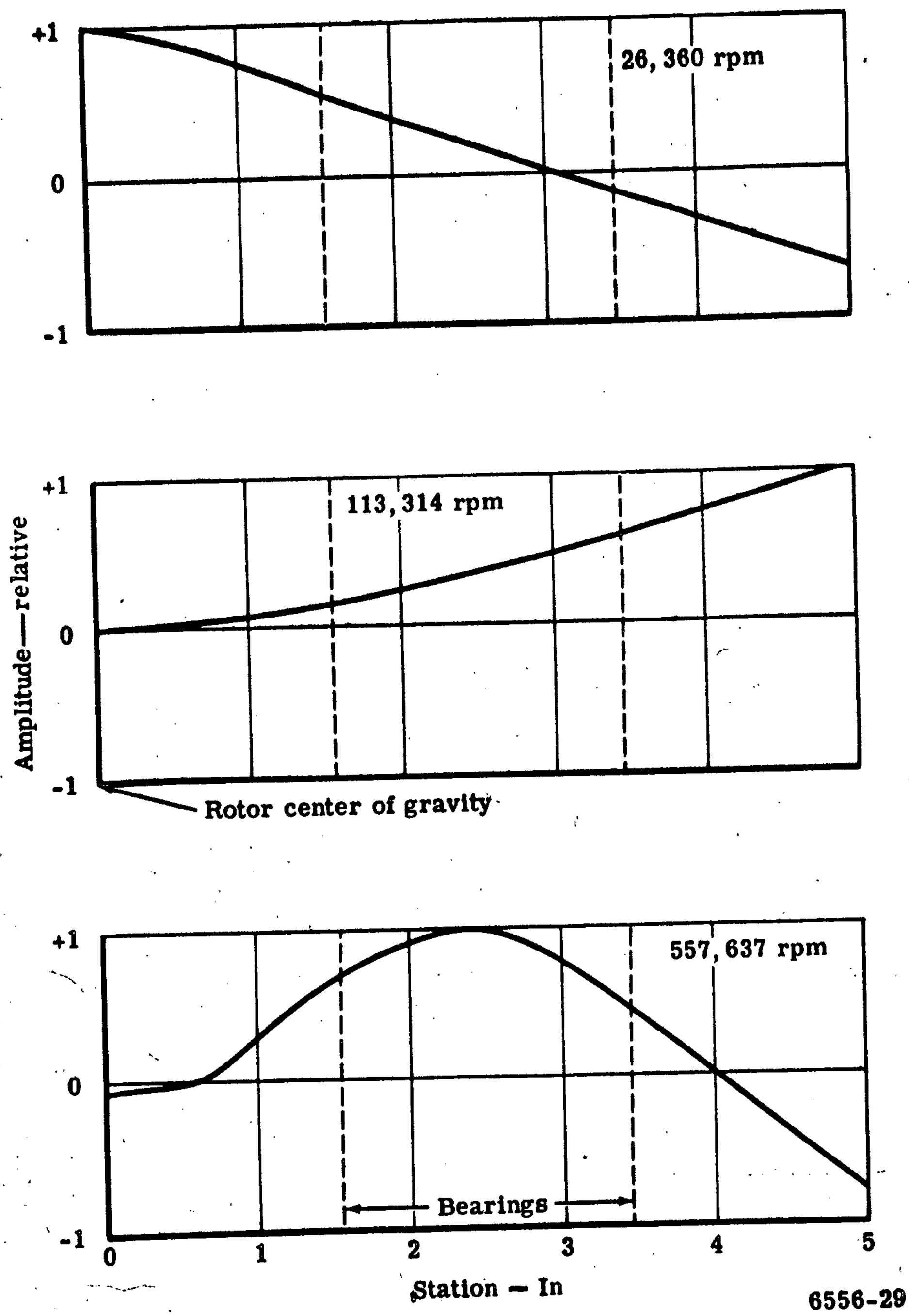


Figure 15. Fanshaft and rotor critical speeds.

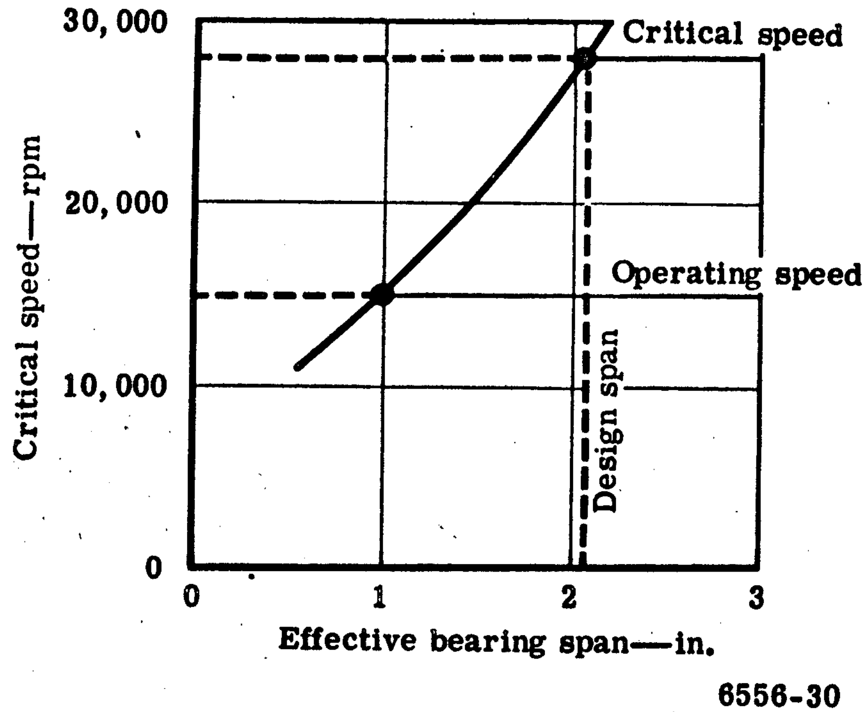


Figure 16. Rotor assembly critical speed versus effective bearing span.

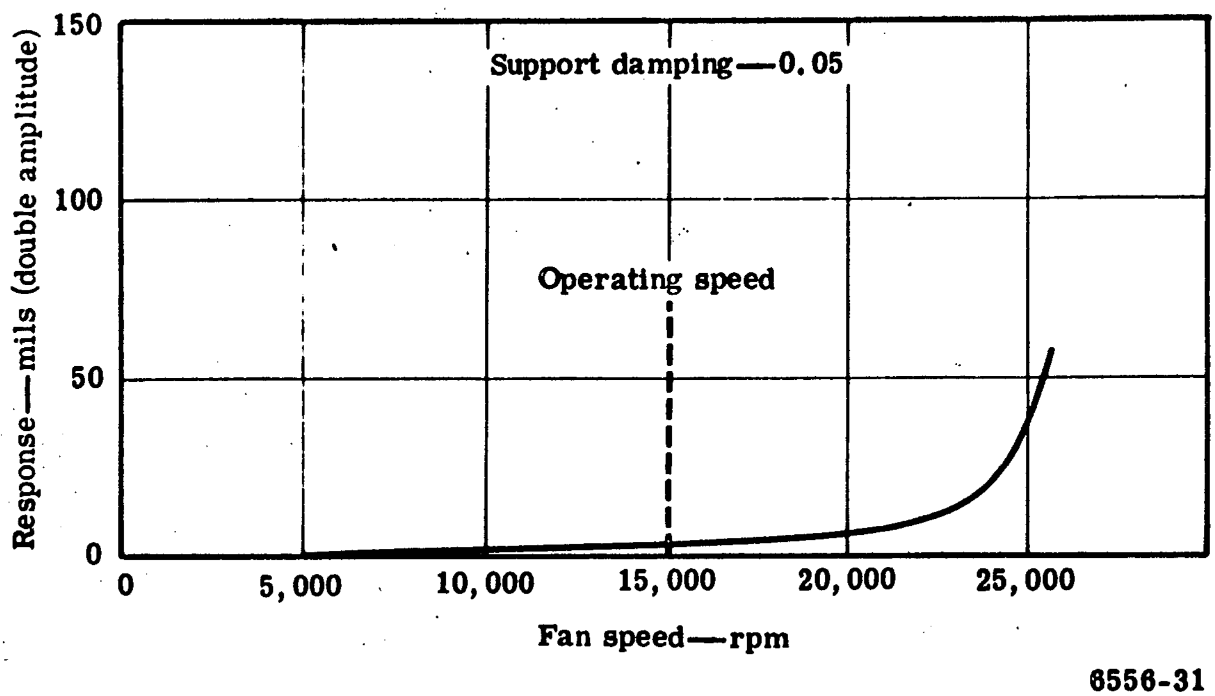


Figure 17. Fanshaft and rotor respondence to 0.02 lb-in. unbalance.

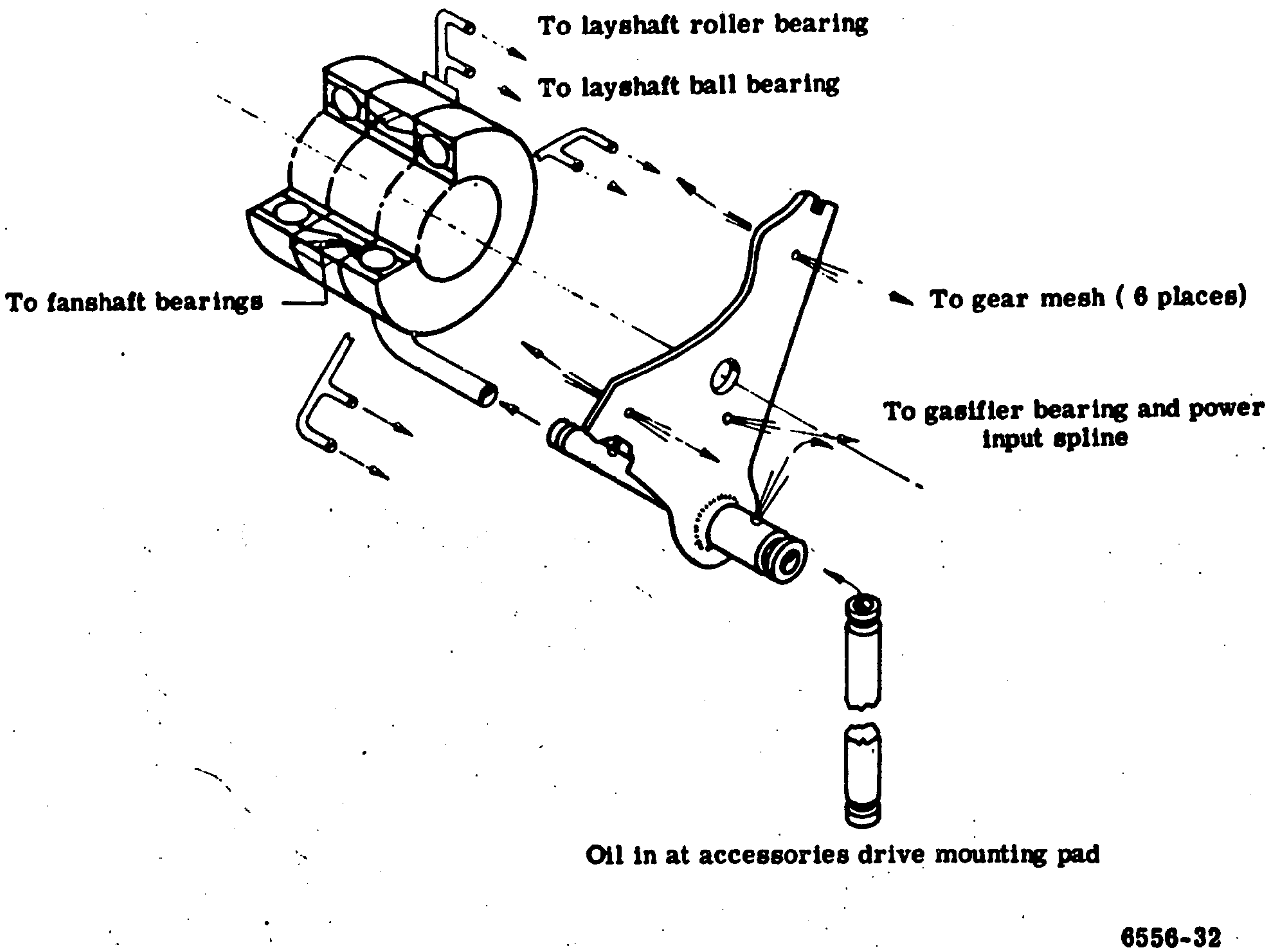
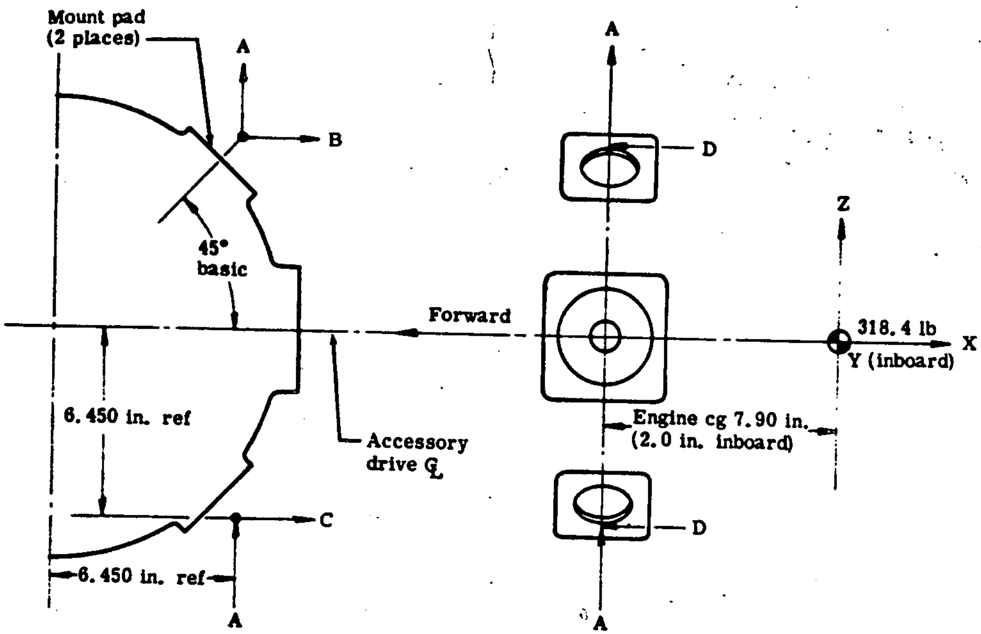


Figure 18. Oil system schematic.



Critical engine reactions at forward mount

Condition	Limit load (lb) A	B	C	D
$G_x = +10$ $G_y = -2$ $G_z = +3$ $\alpha_y = -14$ $\alpha_z = +6$ THR = 1300	+367	-312	-999	-2237
$G_x = +2$ $G_y = -2$ $G_z = +10$ $\alpha_y = -14$ $\alpha_z = +6$ THR = 1300	+1158	+707	-1597	-963
$G_x = +10$ $G_y = -3$ $G_z = +2$ $\alpha_y = -6$ $\alpha_z = +14$ THR = 1300	+214	-556	-1020	-2254
$G_x = +2$ $G_y = -10$ $G_z = +2$ $\alpha_y = -6$ $\alpha_z = +14$ THR = 1300	+214	-1136	-1600	-980

Loads based on MIL-E-5007C: α = rad/sec; thrust = lb fwd
Coordinate system fixed to engine
First two conditions for right-side mounted engine
Last two conditions for bottom-mounted engine

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Figure 19. Engine mount and accessory gearbox installation dimensions.

APPENDIX A

PRELIMINARY STUDIES—ALTERNATE LAYSHAFT ARRANGEMENT

INTRODUCTION

In addition to the layshaft design selected by the NASA Project Manager for design detailing, a number of other arrangements were studied; however, only one was completely layed out for detail approval. The layout is shown in Figure 20.

DESIGN FEATURES

The arrangement is similar to that detailed except:

- Incorporates helical gears (both gear reductions)
- Has fixed input sun gear
- Floating input quill shaft
- Helical gear thrust of first reduction layshaft gears reacted at input sun locating bearing
- Layshafts mounted in roller bearings
- Output sun floating (spline coupled to fan shaft)
- Fan thrust reacted by second reduction helical gears

This configuration was not acceptable to NASA because of expense of additional bearings required at the input sun. These bearings were eliminated in the design detailed.

DESIGN DATA

Figure 21 shows the gear and bearing positions. Tables XV through XVIII show the gear design data, gear loads, bearing loads, and bearing lives, respectively.

COST STUDIES

Value engineering cost studies based on 2000 units per year at a labor rate of $12.50 per hour are summarized as follows:

Raw material	$ 50.25
Purchased parts—Miscellaneous	36.00
—Bearings	323.00
Estimated standard hours (44.1)	551.25
Total	$960.50

TABLE XV—ALTERNATE LAYSHAFT GEAR DATA.

Position No.	No. teeth	Diametral pitch: Plane of rotation	Diametral pitch: Normal	Helix: Angle (degrees)	Helix: Hand	Pitch diameter (in.)	Face width: Minimum (in.)	Face width: Maximum (in.)	Pitch line velocity (ft/min)	Profile contact ratio: Minimum	Profile contact ratio: Maximum	Face contact ratio: Minimum	Face contact ratio: Maximum	*Bending stress (psi)	*Crushing stress (psi)	Pressure angle (degrees)
1	36	20	20.7683	15.6333	RHCW	1.8	0.647	0.652	13,195	1.39	1.62	1.063	1.072	19,248	137,084	20
2	42	20	20.7683	15.6833	LHCCW	2.1	0.597	0.602	13,195	1.39	1.62	1.063	1.072	19,925	137,084	20
3	30	20	20.4468	12	LHCCW	1.5	0.927	0.932	9,425	1.42	1.187	1.187	1.193	20,576	138,662	20
4	48	20	20.4468	12	RHCW	2.4	0.877	0.882	9,425	1.42	1.187	1.187	1.193	18,515	138,662	20

*Duty cycle stresses for single-tooth contact
Bending stresses are for beginning of single-tooth contact
Crushing stresses are for operation at pitch point

TABLE XVI—ALTERNATE LAYSHAFT GEAR LOADS.

Gear No.	Type load (lb)	Gear loads		Gear No.	Type load (lb)	Gear loads	
		Takeoff	Cruise			Takeoff	Cruise
3 and 4	Tangential	764.47	400.325	1 and 2	Tangential	551	288.45
	Separating	278.25	145.70		Separating	200.56	105
	*Thrust	162.5	85.09		*Thrust	154.2	80.72

*Thrust on gears 1 and 4 is 3 times the values indicated.

TABLE XVII—ALTERNATE LAYSHAFT BEARING LOADS.

Bearing position	No. 1		No. 2		No. 3		No. 4	
Power	Takeoff	Cruise	Takeoff	Cruise	Takeoff	Cruise	Takeoff	Cruise
Radial, lb	31.7	31.7	19.4	19.4	467.4	245	319	167
Thrust, lb	237.5	205.3	62.9	94.7	---	---	---	---
Equivalent load, lb	219.6	191.6	62.68	90.34	467.4	245	467.4	245
Duty cycle load, lb	196.33		87.22		303		207	

Bearing position	No. 5		No. 6		No. 7		No. 8	
Power	Takeoff	Cruise	Takeoff	Cruise	Takeoff	Cruise	Takeoff	Cruise
Radial, lb	40	40	89	89	73	73	16.8	16.8
Thrust, lb	---	---	463	242	4.6	4.6	---	---
Equivalent load, lb	40	40	439	247	---	---	---	---
Duty cycle load, lb	40		295		73		16.8	

TABLE XVIII—ALTERNATE LAYSHAFT BEARING LIFE.

Bearing position No. →	No. 1	No. 2	No. 3	No. 4	No. 5	No. 6	No. 7	No. 8
Capacity, lb	3620	3620	3680	3680	1050	5420	1650	965
Load, lb	196.33	87.22	303	207	40	295	73	16.8
L2 life, hr	11,495	131,100	4,576	16,297	51,208	5,908	25,660	421,232

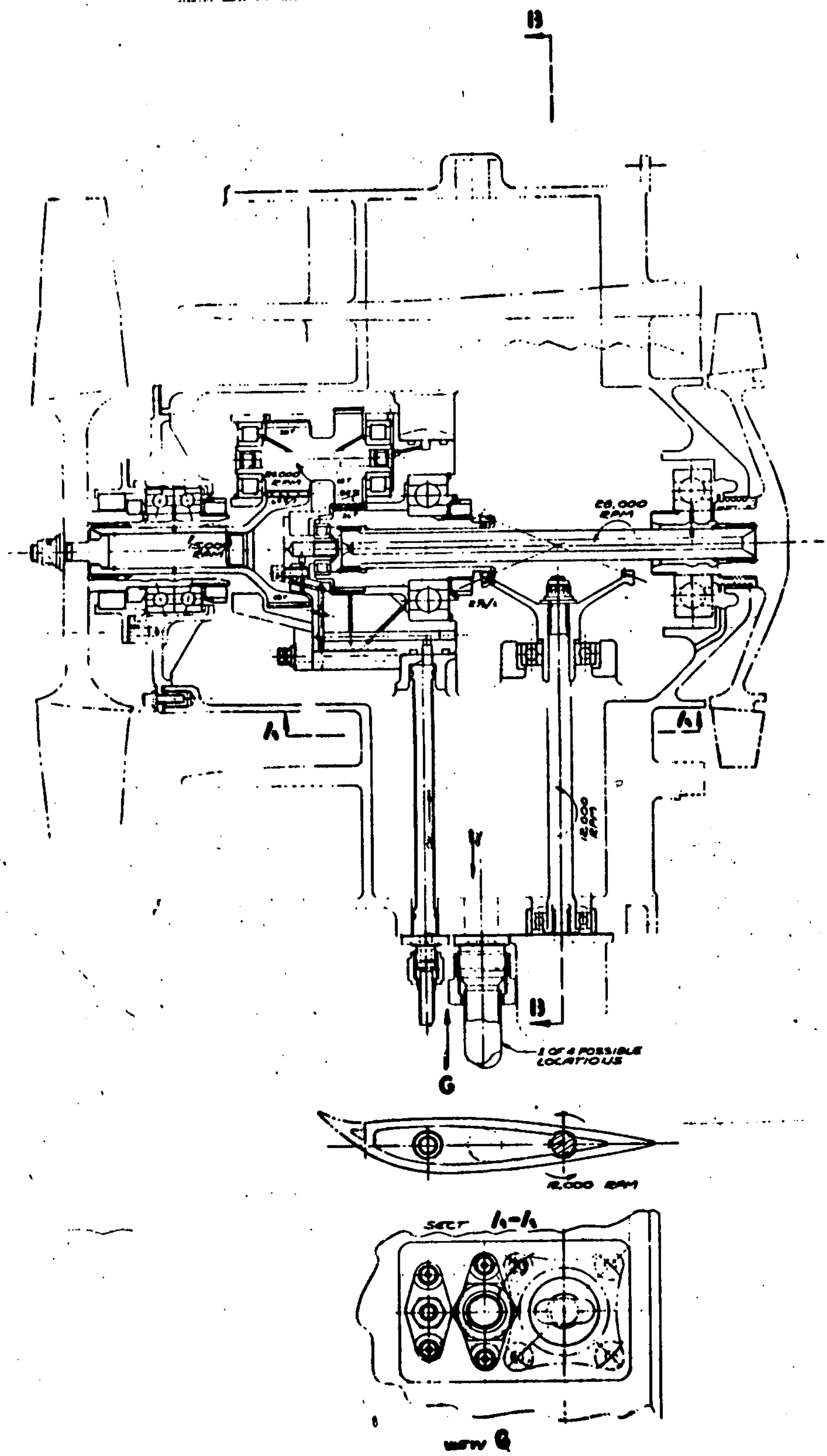

Figure 20. Cross section of alternate layshaft.

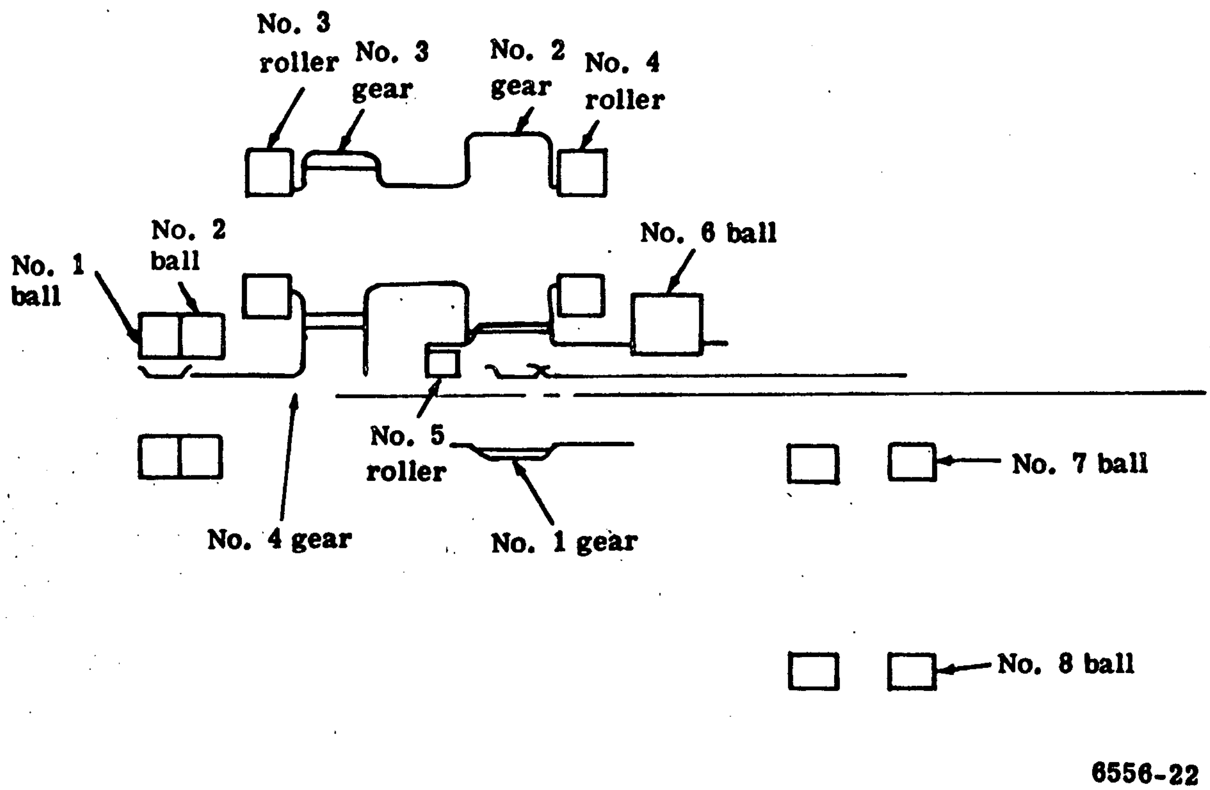

Figure 21. Alternate layshaft gear and bearing positions.

APPENDIX B

PRELIMINARY STUDIES—STAR-PLANET ARRANGEMENTS

INTRODUCTION

Three preliminary star-planet arrangements were studied. The radial space envelope (defined by NASA drawing CR651297 and the gear ratio range specified in NASA Contract NAS3-12417) eliminated the use of rolling element bearings for planet mounting and prevented optimization of a compact star-planet arrangement. NASA letter dated 29 July 1969 from Harold Gold, NASA Project Manager, to R. A. Hirsch suspended all further work effort on designs using hydrodynamic bearings.

ARRANGEMENT STUDIES

Figure 22 shows the general arrangement of the first star-planet concept studied. Torque is transmitted from an internal spline in the gas generator rotor shaft to an internal spline in the gear train power input sun gear by means of a quill shaft. The sun gear is straddle mounted in sleeve bearings, therefore the reduction train imposes only torsional loads on the driving quill shaft. The accessories drive pinion is integral with the sun gear and accessories drive power is extracted through a mating face gear. The planets are also straddle mounted in sleeve bearings.

The planets mesh with a flexible internal gear which is connected to the output shaft through a splined coupling ring. The floating flexible coupling ring provides near equal load sharing between the planet gears. The output shaft is mounted in angular contact bearings and the fan rotor is bolted to it.

The reduction unit consists of a 48 tooth sun gear, four 21 teeth planets, and a 91 tooth ring gear. Speeds of the input, planet, output, and accessories drive are 28,000, 65,333, 15,077, and 12,000 rpm, respectively. Rotation of the fan is opposite to that of the gasifier.

Figure 23 shows an alternate method of extracting the power from the planets to the fan rotor. Figure 24 shows the second arrangement studied. This design is similar to the first with the following exceptions:

- Longer more flexible driving quill shaft
- Planets are centrally supported in sleeve bearings
- Increased span between fan shaft bearings

The third configuration studied shown by Figure 25 is a derivative of the first two with the following exceptions:

- Helical sun, planets, and ring gear

- Sun gear mounted in angular contact ball bearings
- Helical sun gear thrust loads reacted by thrust from spiral bevel accessories drive gear
- Planet thrust reacted by fan shaft bearings

COST STUDIES

Value engineering cost studies based on 2000 units per year at a labor rate of $12.50 per hour are summarized below for the configuration shown on Figure 24.

Raw material	$ 42.75
Purchased parts—Miscellaneous	68.50
—Bearings	124.00
Estimated standard hours (38.2)	477.50
Total	$712.75

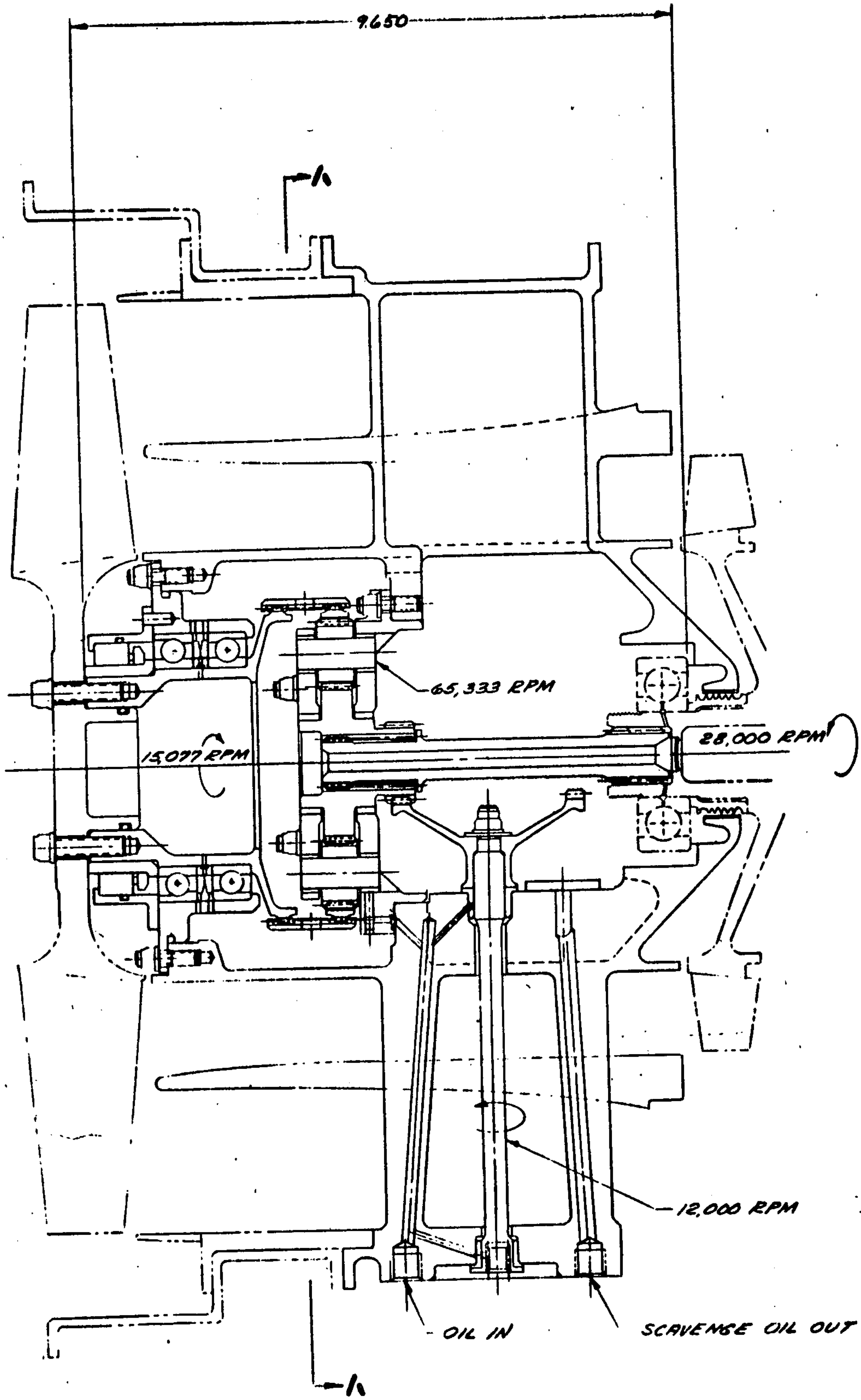

Figure 22. Starplanet arrangement.

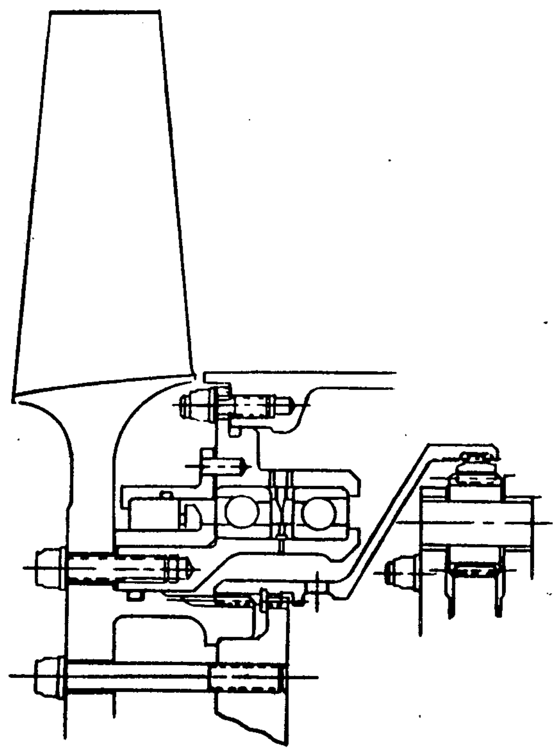

Figure 23. Alternate output construction.

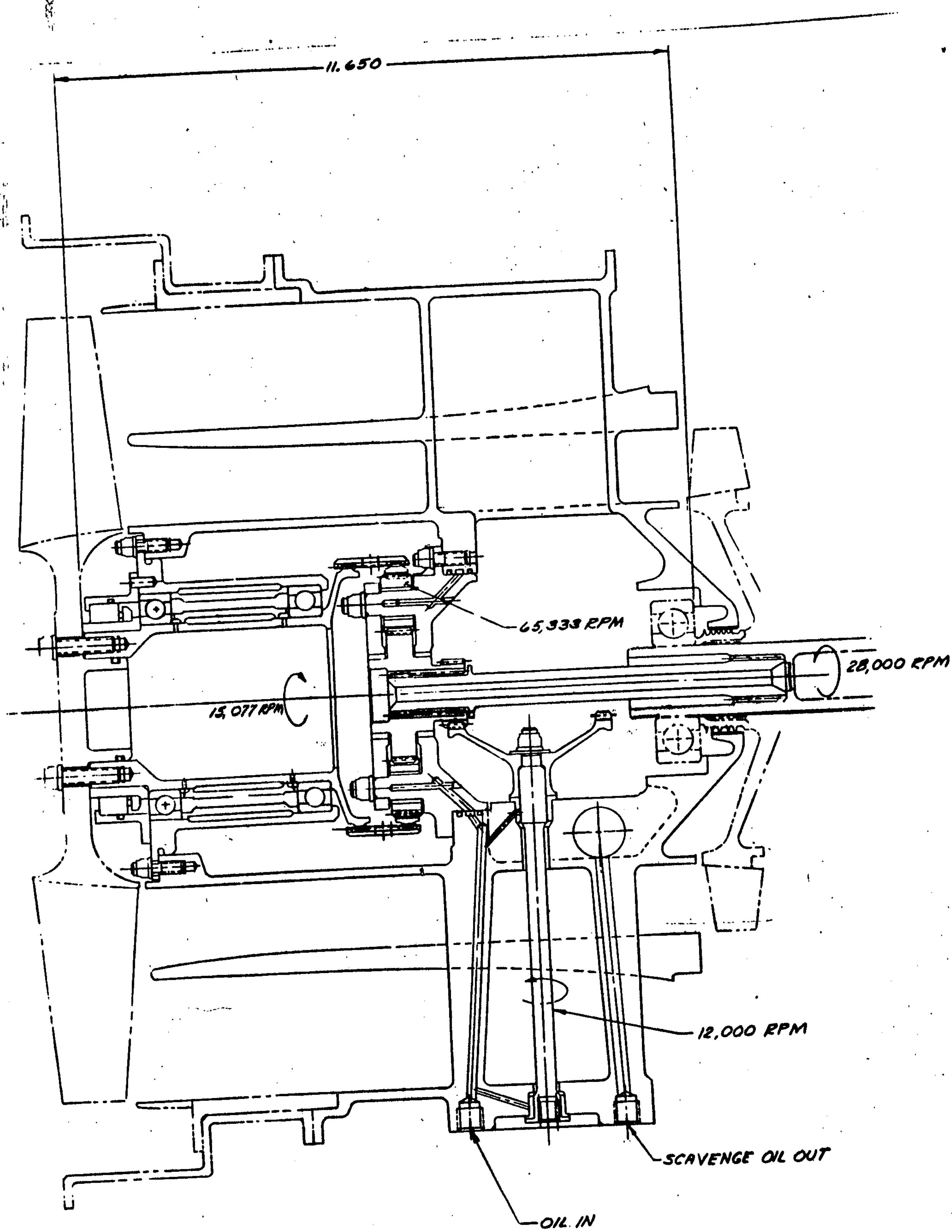

Figure 24. Starplanet arrangement.

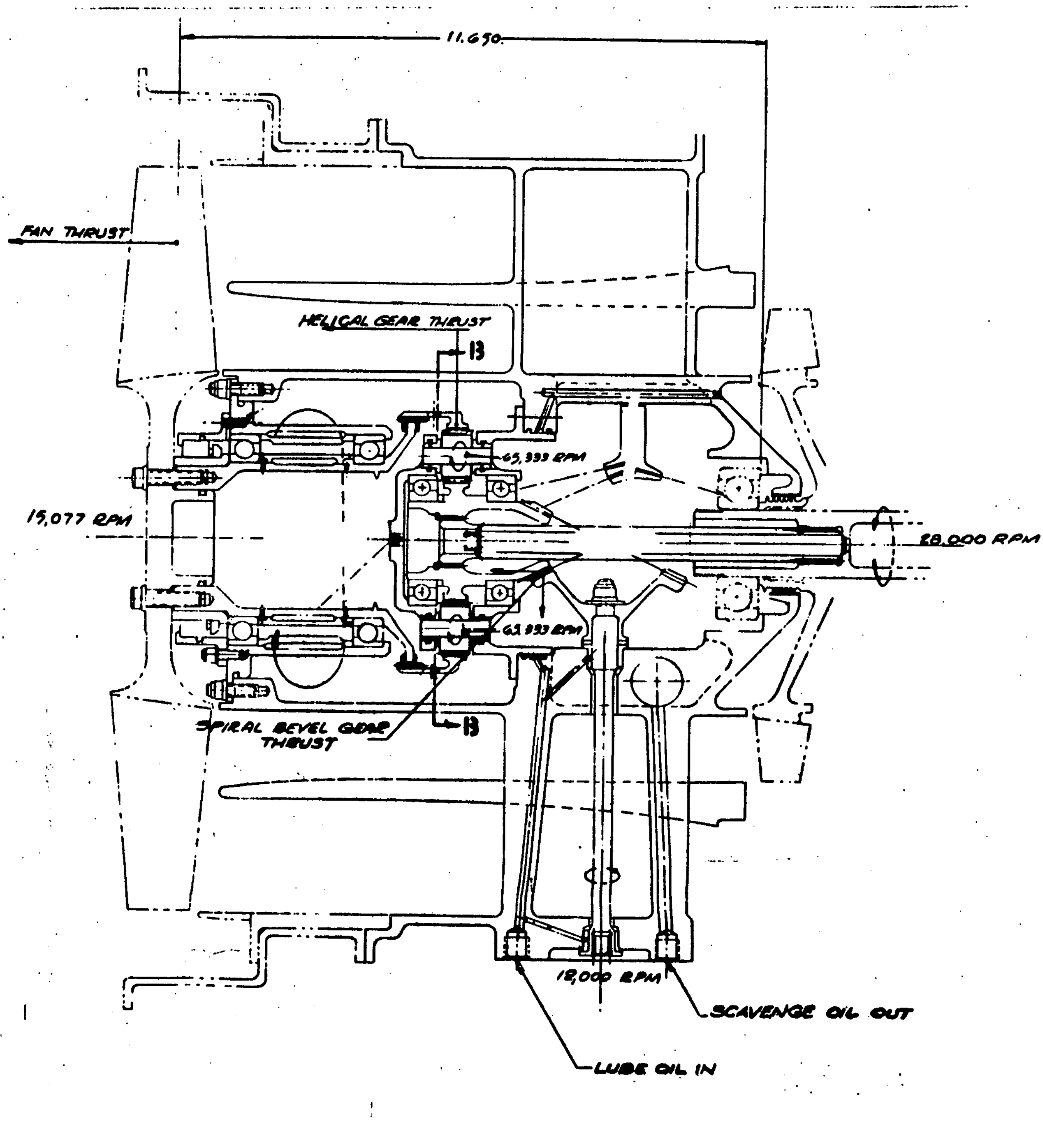

Figure 25. Helical gear starplanet arrangement.

APPENDIX C

PRELIMINARY STUDIES—INTERNAL—EXTERNAL ARRANGEMENTS

INTRODUCTION

Four gear arrangements, six concept sketches, and a number of alternate bearing studies were made. The accessories drive was taken off of the low speed shaft on the early versions, but the high speed shaft arrangement appears to fit more appropriately into the space envelope.

In the interest of low cost, sleeve bearings were considered in all but the last design. All work on sleeve bearing studies was stopped per NASA letter dated 29 July 1969.

Initially the internal-external concept appeared to be most attractive due to simplicity and minimum number of parts; however, it requires offset shafting and ducting and has poor growth potential due to extremely high bearing loads.

ARRANGEMENT STUDIES

Figure 26 shows the general arrangement of the first concept studied. The sun gear is integral with the input drive and is mounted in a sleeve bearing. The internal gear and fan shaft is supported by an angular contact ball bearing at the front and by a sleeve bearing at the rear. The accessories drive is taken off of the low speed shaft.

Figure 27 is similar to Figure 26 except for a change in gear diametral pitch, fanshaft bearings, and location of mounting sleeve bearing yoke.

Figure 28 shows the accessories drive taken off of the high speed shaft. The sun gear, accessories drive pinion and input shaft are integral and the sun gear is mounted in a sleeve bearing. The integral internal gear and fanshaft is mounted in antifriction bearings. Figure 29 is a value engineered version of Figure 28 and Figure 30 shows the input sun gear straddle mounted in antifriction bearings. Figure 31 is similar to Figure 27 except for a change in fanshaft bearings and sleeve bearing yoke modification.

Figure 32 shows the last arrangement studied and is similar to Figure 28 except for straddle mounting of sun gear in antifriction bearings.

DESIGN DATA

Table XIX shows the gear data for the configurations studied and Tables XX through XXIII show the bearing load and bearing life calculations for Figures 28 and 32.

COST STUDIES

Value engineering cost studies based on 2000 units per year at a labor rate of $12.50 per hour are summarized below for the configuration shown on Figure 32.

Raw material	$ 45.25
Purchased parts—Miscellaneous	93.75
—Bearings	208.00
Estimated standard hours (24.8)	310.00
Total	$657.00

TABLE XIX—INTERNAL-EXTERNAL GEAR DATA.

Study No.	Type gear	No. teeth	Shaft offset (in.)	RPM	Pressure angle (degrees)	Pitch diameter (in.)	Diametral pitch		Helix angle (degrees)	Face width (in.)	Gear loads (psi)		Tooth stress (psi)	
							Plane of rotation	Normal plane			Takeoff	Cruise	Bending	Crushing
1	Pinion gear Internal gear	23 43	0.760	28,025 15,000	25	1.7483 3.2685	13.1557	14	20	0.760	1923	1044	30,000	130,000
1A	Pinion gear Internal gear	30 56	0.8125	28,000 15,000	25	1.875 3.500	16.00	---	0	0.750	---	---	25,000	140,000
2	Pinion gear Internal gear	30 56	1.083	28,000 15,000	25	2.500 4.666	12.0	---	0	---	1347	731	---	---
3	Pinion gear Internal gear	30 56	1.083	28,000 15,000	20	2.500 4.666	12.0	---	0	---	1299	705	---	---

TABLE XX—BEARING LOAD INTERNAL-EXTERNAL. SEE FIGURE 28.

Gear No.	Type load	Gear loads Takeoff (lb)	Gear loads Cruise (lb)	No. 1 bearing Takeoff (lb)	No. 1 bearing Cruise (lb)	No. 2 bearing Takeoff (lb)	No. 2 bearing Cruise (lb)	No. 3 bearing Takeoff (lb)
2	Tangential	1221	663	720.33	390.98	1941.2	1053.65	1221
	Separating	569.3	309	355.89	182.31	905.19	491.3	569.3
	Thrust			400	200			
	Radial			794.8	431.4	2141.9	1162.6	
	Equivalent load			1143	647.6			
	Duty cycle			(769)		(1411)		
3	Tangential	52.5						43.54
	Separating	19.11						15.85
	Radial							(1394)

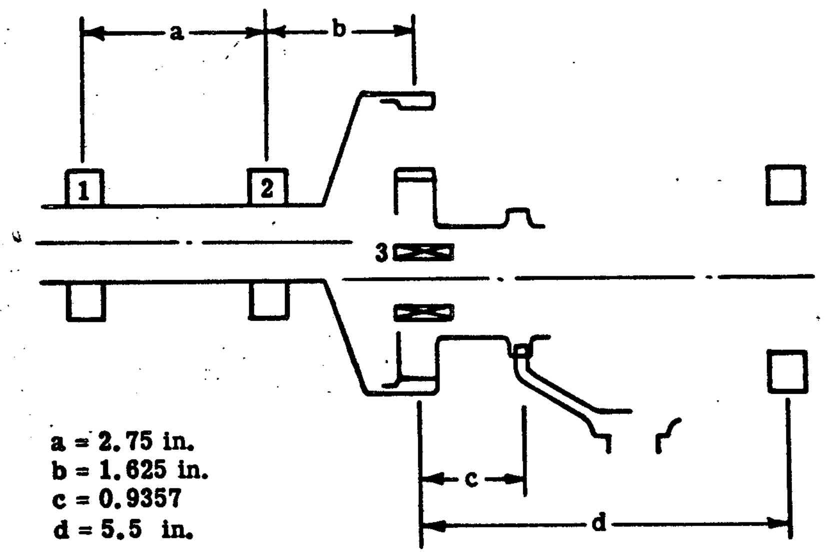

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TABLE XXI—BEARING LIFE DATA FOR FIGURE 28.

Bearing position No.	1	2
Size, mm	65 × 120 × 23	70 × 110 × 20
Type	Ball	Roller
Capacity, lb	9900	15,700
Load, lb	769	1,411
*L2 life, hr	3794	5,468

*Material life factor = 5

TABLE XXII—BEARING LOAD DATA FOR INTERNAL-EXTERNAL DESIGN. SEE FIGURE 32.

Gear No.	Type load	Gear loads		No. 1 bearing		No. 2 bearing		No. 3 bearing		No. 4 bearing	
		Takeoff (lb)	Cruise (lb)	Takeoff (lb)	Cruise (lb)	Takeoff (lb)	Cruise (lb)	Takeoff (lb)	Cruise (lb)	Takeoff (lb)	Cruise (lb)
2	Tangential	1221	663	505.2	274.2	1755.6	952.87				
	Separating	444.4	241.2	183.88	99.8	628.25	341				
	Thrust			400	200						
	Radial	1299.2	705.2	537.63	291.8	1868	1014	649.6	352.6	649.6	352.6
	Equivalent load			982.27	560						
	Duty cycle load			(663)		(1231)					
3	Tangential	52.5	52.5								
	Separating	19.11	19.11								
	Radial							-28.0	-28.0	+83.8	+83.8
	Σ radial							621.6	324.6	733.4	436.4
	Duty cycle load							(402.28)		(506.34)	

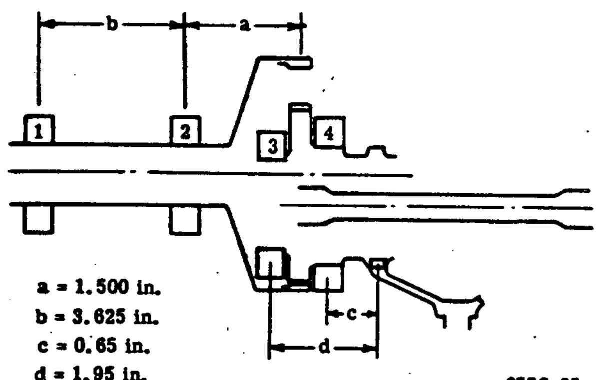

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TABLE XXIII—BEARING LIFE DATA FOR FIGURE 32.

Bearing position No.	1	2	3	4
Size, mm	90 × 125 × 18	90 × 125 × 18	25 × 52 × 15	35 × 62 × 14
Type	Ball	Roller	Roller	Roller
Capacity, lb	8240	15, 500	5420	6690
Load, lb	663	1, 231	402. 28	506. 34
*L2 life, hr	3413	9, 620	5544	5194

*Material life factor = 5

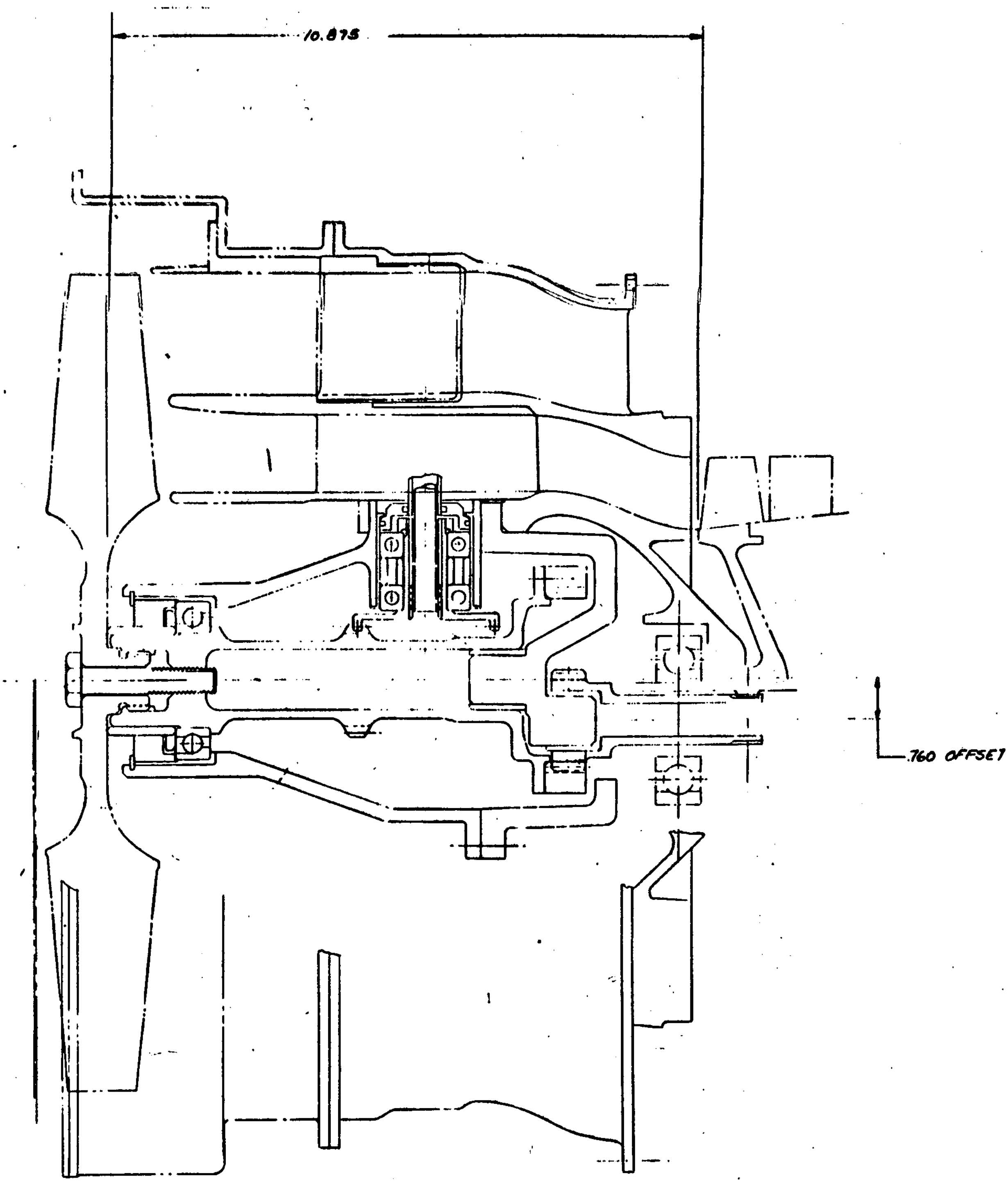

Figure 26. Internal-external arrangement—helical gears.

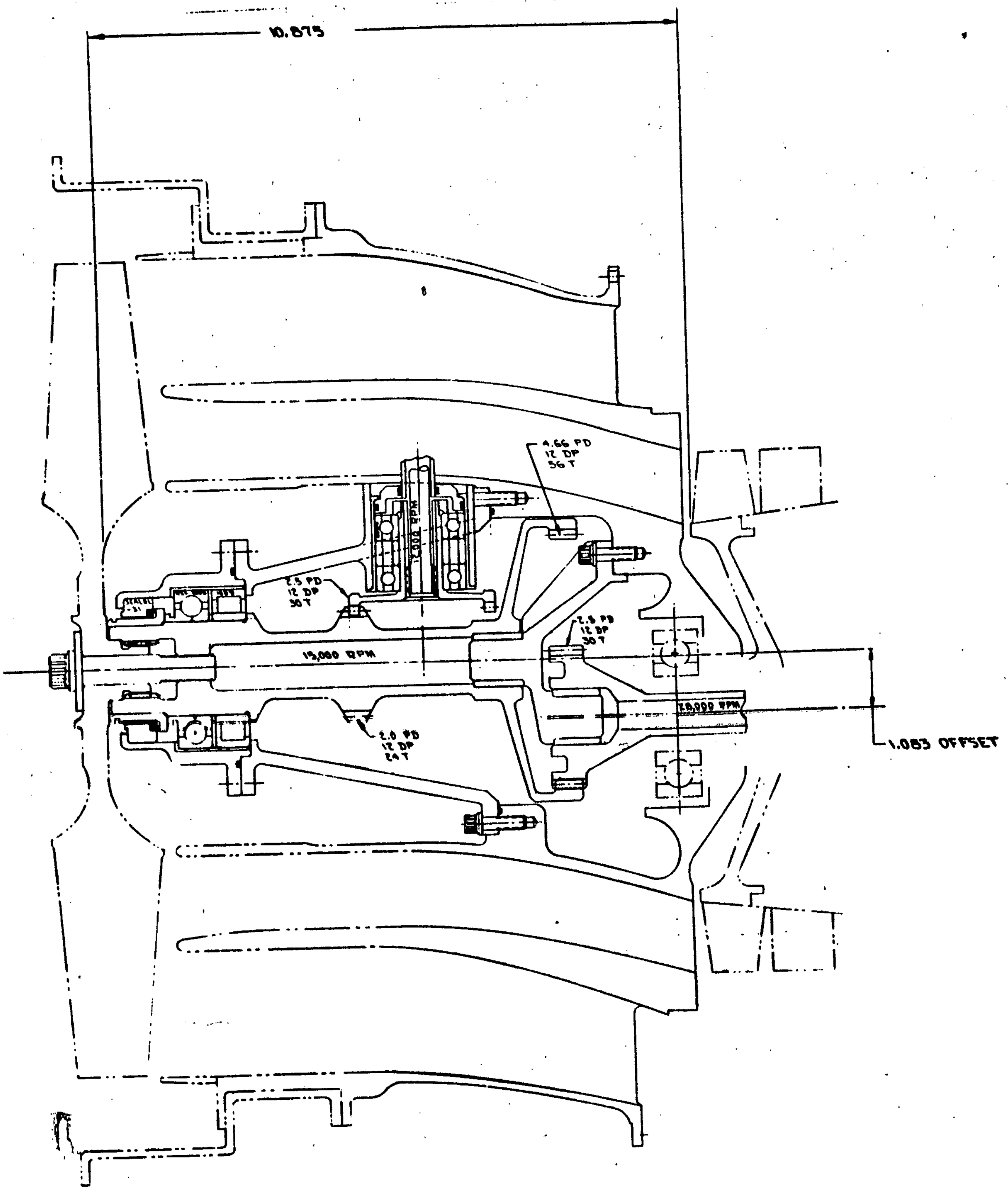

Figure 27. Internal-external arrangement—spur gears.

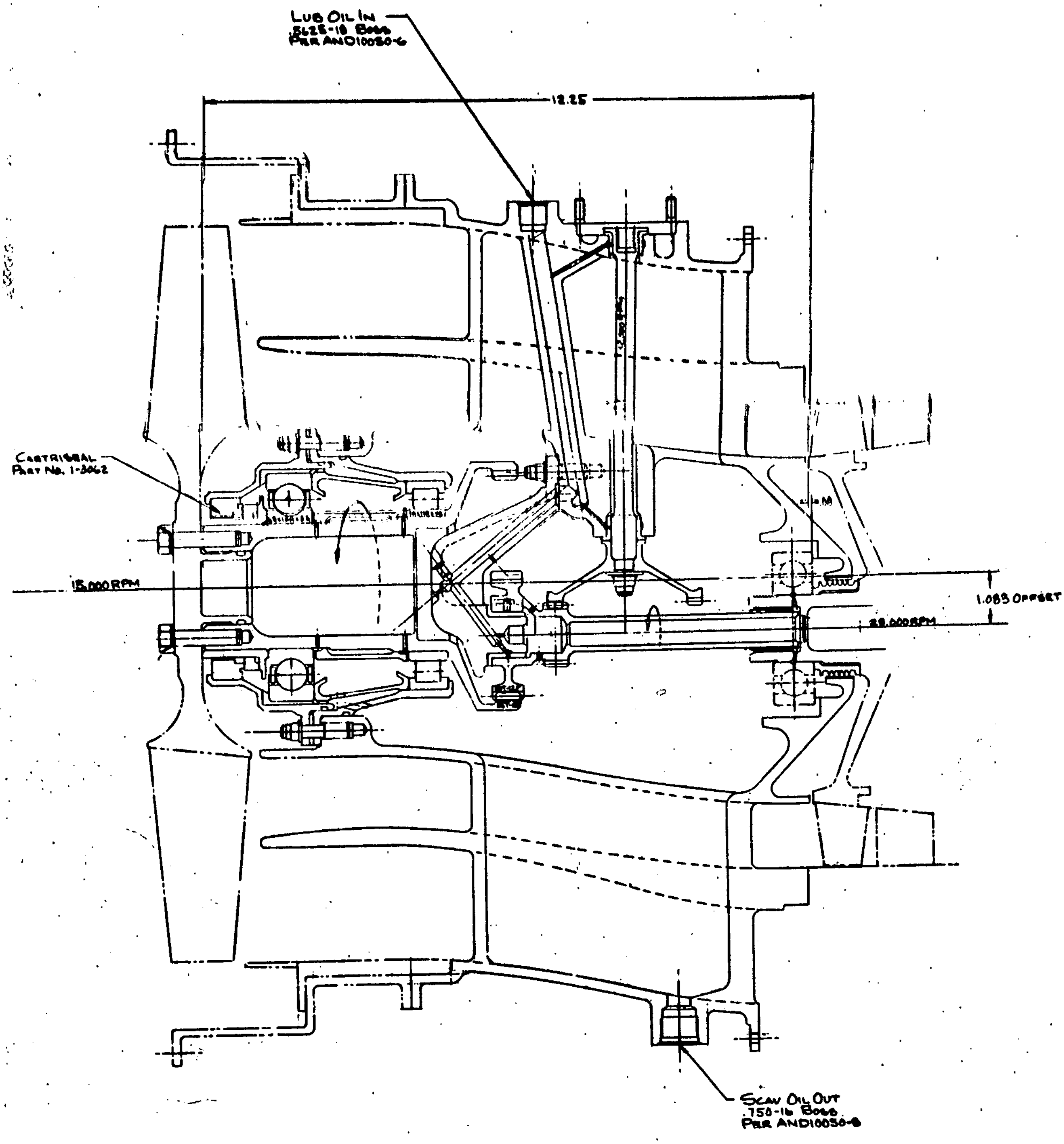

Figure 28. Internal-external arrangement—spur gears (accessories drive from high speed shaft).

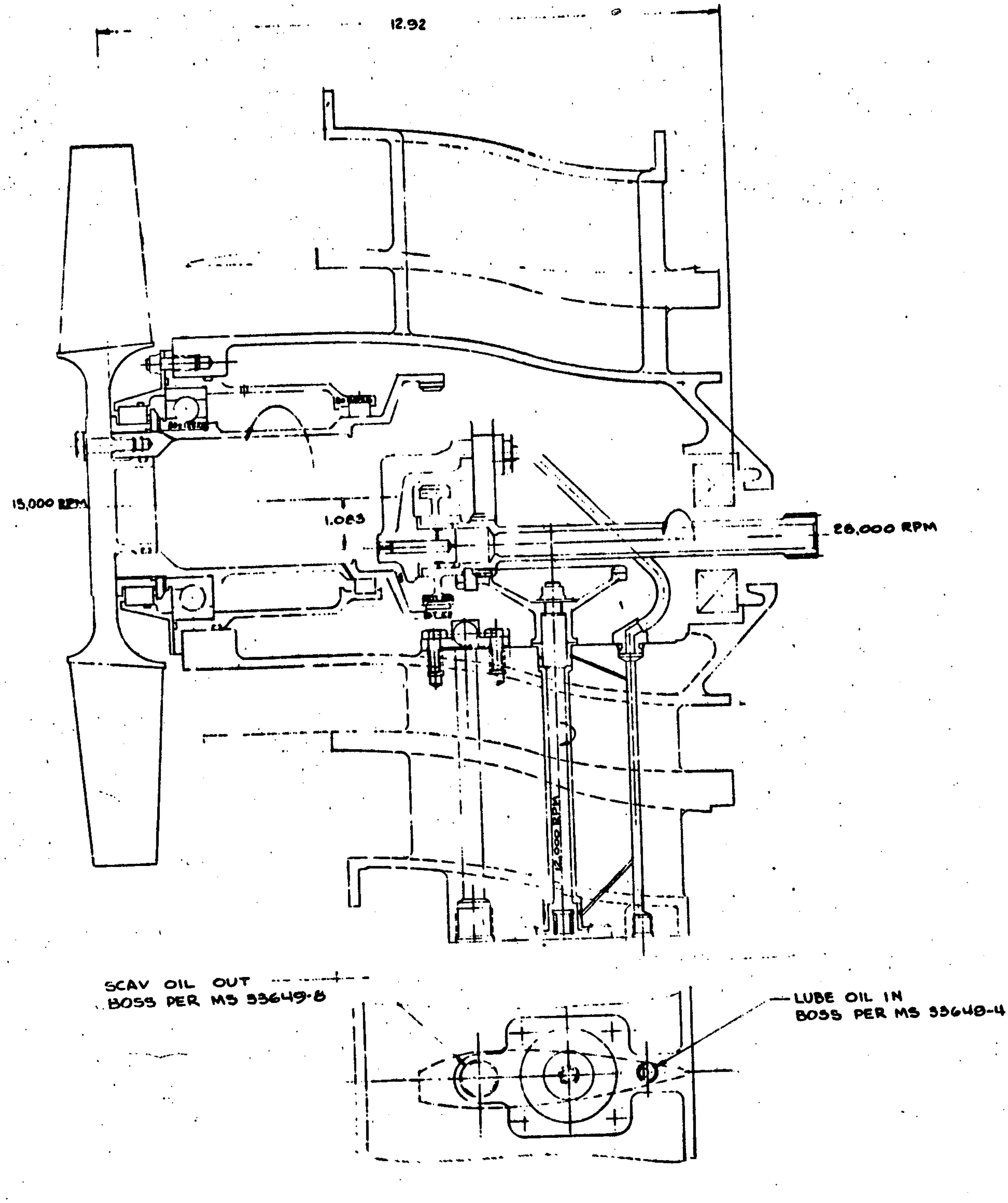

Figure 29. Valve Engineering version of Figure 28 showing internal/external arrangement—spur gears.

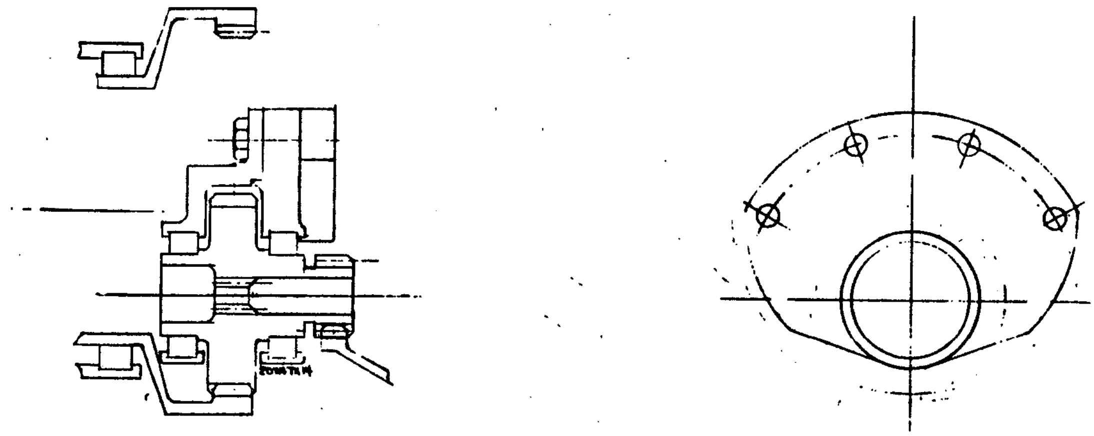

Figure 30. Alternate design (sun gear mounted in antifriction bearings).

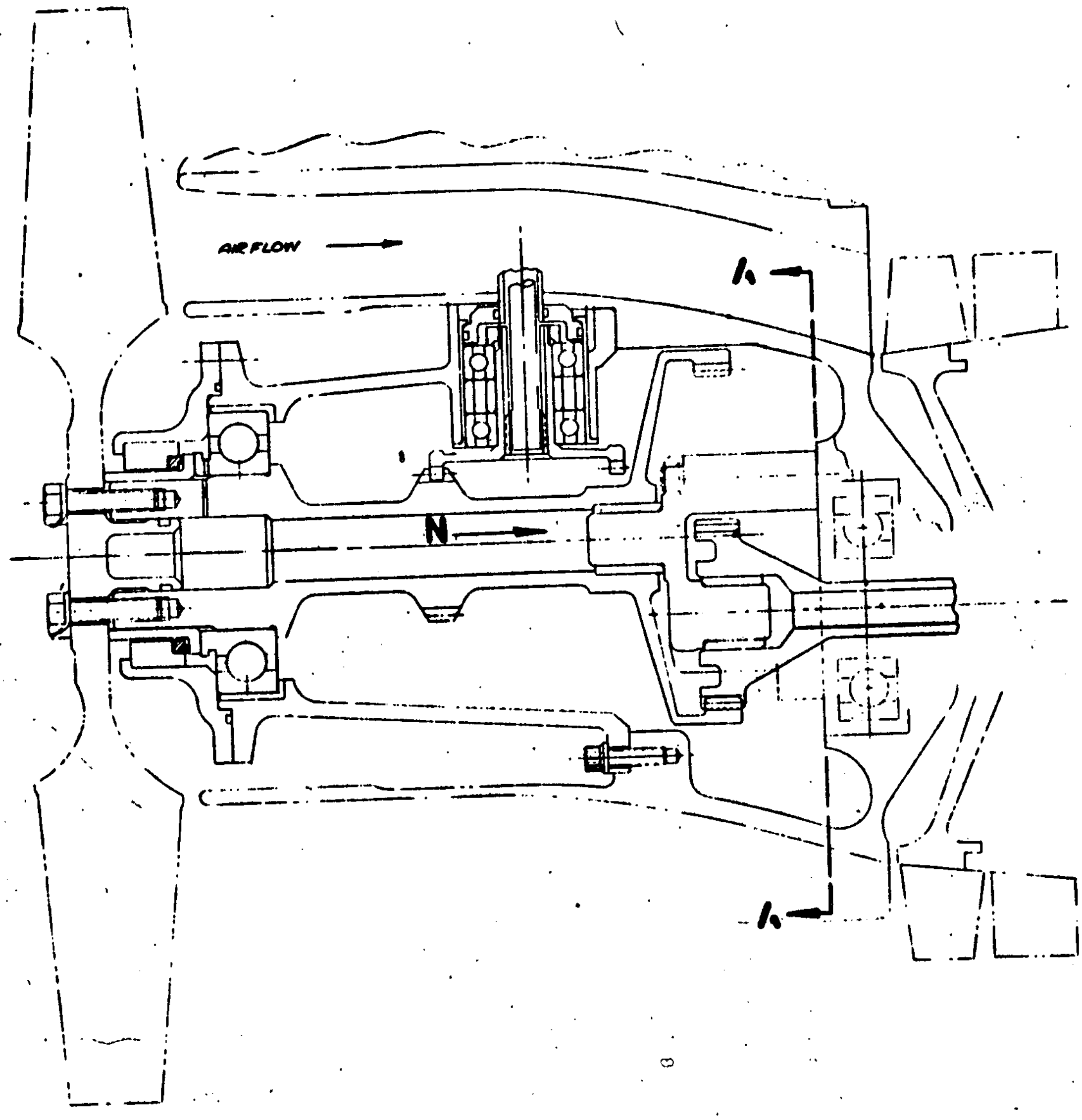

Figure 31. Alternate version of Figure 27 showing internal/external arrangement—spur gears.

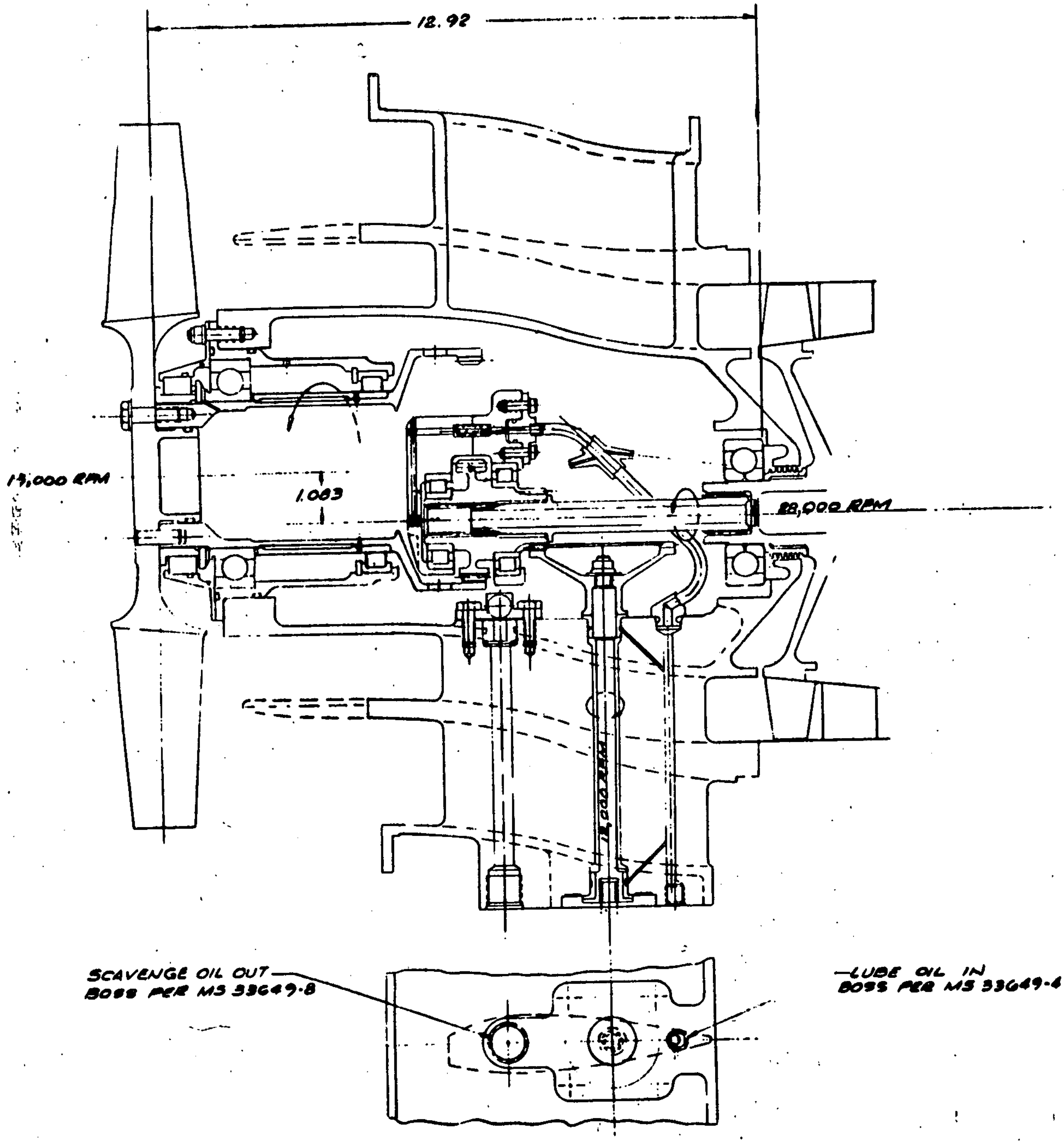

Figure 32. All antifriction bearings except accessories shaft bearings.

APPENDIX D

SYMBOLS

B_s	Gear tooth bending stress, psi
C	Bearing basic load rating for 1 million revolutions, lb
C_o	Bearing static capacity, lb
CF	Centrifugal force, lb
CG	Center of gravity
C_s	Gear tooth crushing stress, psi
D	Bearing ball diameter, in.
D_i	Inside diameter, in.
D_o	Outside diameter, in.
d_m	Bearing pitch diameter, in.
e	Bearing factor
e	Effective bearing span, in.
F_a	Bearing axial load, lb
F_r	Bearing radial load, lb
f_c	Bearing factor
g	Acceleration of gravity, ft/sec^2
hp	Horsepower
I	Moment of inertia, slug ft^2
i	Number of rows of balls
L	Spline length, in.
mm	Bearing dimensions, milimeter
P	Bearing load, lb

PD	Pitch diameter, in.
$\bar{R}$	Distance to centroid, ft
S_c	Spline crushing stress, psi
S_s	Torsional shear stress, psi
T	Maneuver load, lb-ft
T	Gear thrust, lb
T	Shaft and gear torque, lb-in.
V	Bearing rotation factor
W	Weight, lb
W_r	Gear radial load, lb
W_t	Gear tangential load, lb
X	Bearing radial factor
Y	Bearing thrust factor
Z	Number of balls
α	Bearing contact angle
Ω	Radian per second maneuver
ω	Radians/sec