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USAAVLABS TECHNICAL REPORT 70-3

1100-HP ROLLER GEAR DRIVE

By

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January 1970

U. S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

CONTRACT DA 44-177-AMC-30(T)

TRW MECHANICAL PRODUCTS DIVISION

CLEVELAND, OHIO

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DEPARTMENT OF THE ARMY
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This report presents a part of a continuing research program to investigate new concepts of high-speed gear reducers for use as main transmissions in helicopters. The main effort of this particular program was to conduct endurance tests on two 1100-hp roller gear transmissions.

Although the specific design tested under this program proved to be unsatisfactory, minor design changes in a second-generation model should render a successful test unit.

It is concluded from the tests conducted under this program that the roller gear transmission represents an improvement in the state of the art for helicopter transmissions.

The roller gear concept is being explored further under Contract DAAJ02-69-C-0042 with Sikorsky Aircraft Division of United Aircraft Corporation.

It is recommended that the roller gear system be considered in advanced gearbox and transmission system designs.

Task 1G162203D14414
Contract DA 44-177-AMC-30(T)
USAAVLABS Technical Report 70-3

January 1970

1100-HP ROLLER GEAR DRIVE

Final Report

TRW
Engineering Report 7378

By

Algirdas L. Nasvytis

James H. Hemlein

Prepared by

TRW Mechanical Products Division
Cleveland, Ohio

for

U.S. ARMY AVIATION MATERIEL LABORATORIES
FORT EUSTIS, VIRGINIA

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ABSTRACT

This report presents a review of the dynamic load tests performed on each of two roller gear test units. The primary purpose of this task was to conduct a 200-hour dynamic load test at 100-percent design speed and 100-percent design load.

The test units did not complete the scheduled testing because of test rig malfunction and minor design deficiencies which became apparent during the full-load portion of the test program.

A total of 76.5 hours of testing was logged, of which 44.5 hours were at the full-power condition of 1100 hp with an input shaft speed of 21,000 rpm and an output shaft speed of 325 rpm. Two design modifications were necessary during the testing program.

The testing system employed the principle of back-to-back (four-square loop) coupling of two test units in a closed-power path. An instrumentation system was provided to measure loads, speed, oil flows, pressures, temperatures, and vibration levels. In addition, frequent analysis of the lubrication oil was made to determine the iron content in parts per million to provide a warning of incipient failure.

The testing performed has confirmed that the roller gear drive design is sound and is feasible for helicopter speed reductions. It has further identified critical design parameters which must be considered in future designs.

FOREWORD

The U.S. Aviation Materiel Laboratories entered into Contract DA 44-177-AMC-30(T) (DA Task 1G162203D14414) with TRW for the purpose of conducting the analysis, design, fabrication, and test of an 1100-hp roller gear drive. This report is a review of the full-load testing.

The work under this contract was conducted in the Accessories and Mechanical Products Divisions of TRW.

The liaison work with USAAVLABS has been conducted with Mr. Wayne Hudgins and Mr. Nelson Daniel.

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A	sun gear
a	pitch radius of the sun gear
C	ring gear
c	pitch radius of the ring gear
X_1	large gear of the first-row planet gears
x_1	pitch radius of the large gear of the first-row planet gears
X_2	second-row gear
x_2	pitch radius of the second-row gear
Y_1	smaller gear of the first-row planet gear
y_1	pitch radius of the smaller gear in the first-row planet gears
z	distance from the center of the sun gear to the center of the second-row planet
α	toggle angle is the angle between the lines connecting the centers of the sun gear to the first-row planet and the first-row planet to the second-row planet minus ninety degrees
γ	complement of the toggle angle is the angle between the lines connecting the centers of the sun gear to the second-row planet and the second-row planet to the first-row planet
θ	half angle between the second-row planets

INTRODUCTION

This report presents the results of the testing performed on an 1100-hp roller gear power transmission under Contract DA 44-177-AMC-30(T). The objective of this task was to conduct a 200-hour dynamic load test on each of two roller gear transmission test units at 1100 hp with an input shaft speed of 21,000 rpm and an output shaft speed of 325 rpm.

A description of the roller gear drive, the test method, and the test results is included in this report. Load stress calculations are included in the appendix.

TEST HARDWARE

DESIGN CONCEPT

The basic hardware design specification for the 1100-hp drive requires a speed reduction from 21,000 rpm input to 325 rpm output including the ring-angle bevel gear pair. The total drive reduction ratio is 64.62:1. The standard bevel pair reduction ratio is approximately 2:1; the remaining ratio (32:1) is obtained in the roller gear drive. Based on parametric studies of roller reduction drives (reference USAAVLABS Technical Report 64-29), the 32:1 ratio is obtained by using two rows of planets. It is preferable to use fewer rows of planets because fewer parts result in a higher efficiency due to the smaller number of power transfer contacts. Usually in a two-row drive, the output is the rotating ring gear and the planets are fixed. The two-row drive with rotating planets has the output rotating in a reversed direction and at a reduced ratio. The reversed output requires larger bearings due to higher reaction torques. Therefore, a rotating ring output was selected for the design to be tested.

Exploratory layouts and calculations show that four-planet drives will be heavier and will require a larger OD ring gear at 1100 hp. The final decision was to use six planets in a row with staggered first-row planets. This type of drive for the required ratio can be designed without a step in the second-row planets. Simple second-row planets save twelve gear elements and simplify output ring design. A weight saving results from fewer gear elements and a narrow output ring gear. The axial length of the drive is also reduced as a result of the narrow output ring gear.

Tooth Number Selection

It is desirable to use all roller gear elements in a single row with identical indexing. Therefore, there must be even numbers of teeth on the gears between input and output contacts. The sun gear must have a tooth number divisible by six. The first-row planet in contact with the sun gear does not have a tooth number requirement. To simplify the tooth number requirement in the second row input-output and in the ring gear, the selected toggle angle is $\alpha = 30^\circ$. The roller gear cluster geometry with the symbols used in the following discussion is shown in Figure 1.

From the roller gear cluster geometry (Figure 1), two equations can be written:

$$(a + x_1) \sin \theta = (x_2 + y_1) \sin \gamma \qquad (1)$$

$$(a + x_1) \cos \theta + (x_2 + y_1) \cos \gamma + x_2 = c \qquad (2)$$

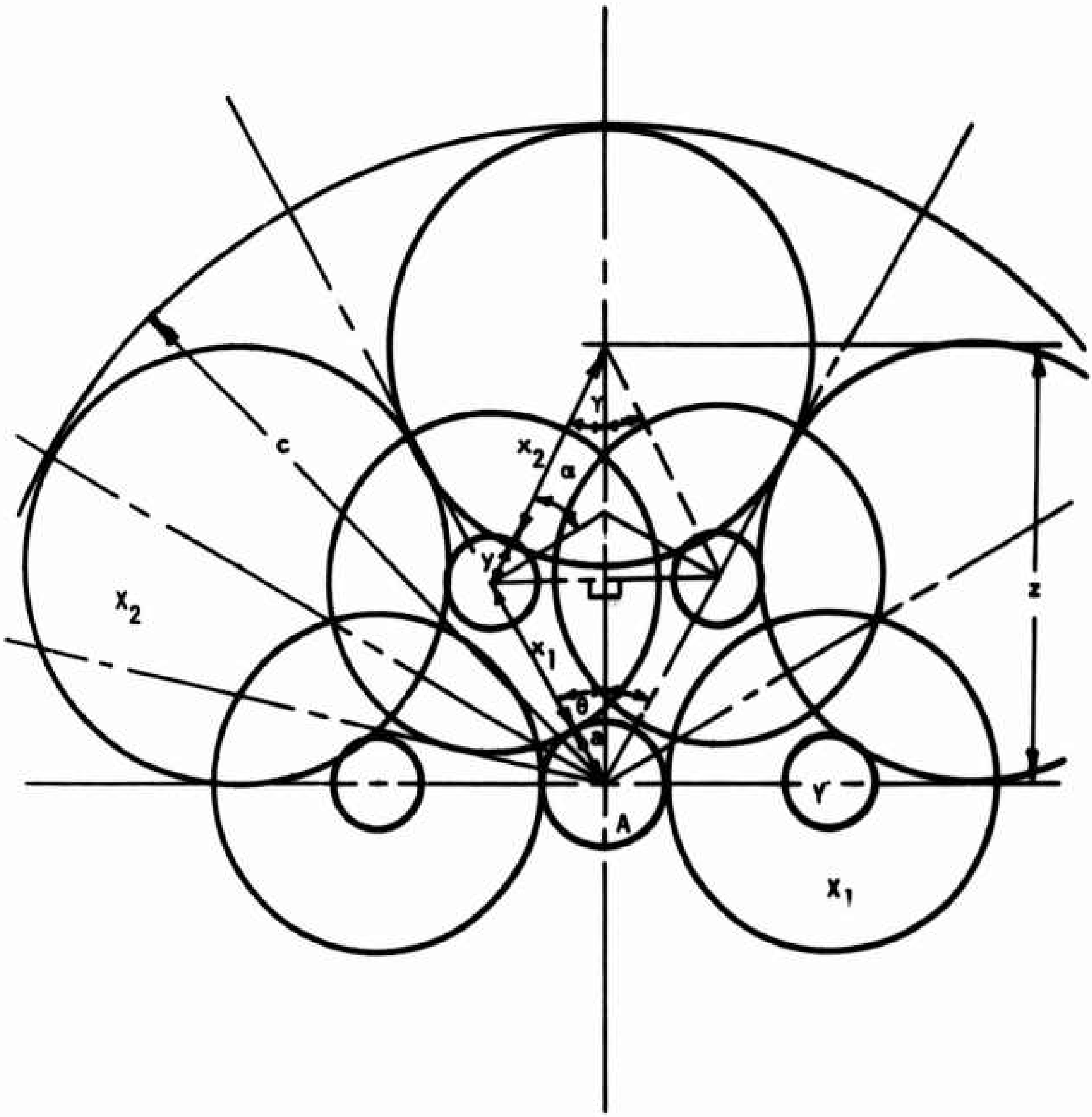


Figure 1. Roller Gear Cluster Geometry.

Since $(a + x_1) = (x_2 + y_1)$, the linear distances c and z can be expressed by the following equations:

$$2\,(x_2 + y_1)\cos 30^\circ + x_2 = c$$

$$2\,(x_2 + y_1)\cos 30^\circ = z$$

From the same geometry, gear tooth assembly conditions with even numbers of teeth are

for Y_1 $\dfrac{2\,(90-\alpha)}{360}$ x number of teeth in Y_1 = a whole number

since $\alpha = 30^\circ$

$$\frac{Y_1}{3} = \text{a whole number}$$

for X_2 $\dfrac{2\gamma}{360}$ x number of teeth in X_2 = a whole number

since $\gamma = 30^\circ$

$$\frac{X_2}{6} = \text{a whole number}$$

The minimum number of teeth for Y_1 is either 15 or 18. Eighteen was selected.

From the layout study, the number of teeth for X_2 could be 66, 72, 78, or 84. The X_2 planet without steps requires that the ring gear have the same diametral pitch as X_2 and Y_1 and that the number of teeth be divisible by six. Since the number of teeth in the ring gear is a function of the number of teeth in the first and second-row planets, a search must be made for as close as possible a solution for the number of teeth in the ring gear. (An exact solution is not possible with a toggle angle of 30° since the cosine of this angle is not an exact number.

In Equation (2) the linear dimension can be replaced by the number of teeth. The trial calculations are:

Y_1	X_2	C
18	66	211.490
18	72	227.885
18	78	243.270
18	84	260.670

The combination 18-72 results in a number of teeth in the ring gear close to a number divisible by six with an error of 228/227.88 = 1.0005.

The size of the sun gear is based on the maximum torsional windup condition and the requirement not to exceed a .0002-in. radial twist between the two most remote gear edges.

From the layout study, the maximum ratio of $x_1/a = 2.75$ with $a = 1.200$ in. and $x_1 = 3.300$ in.

The torsional windup calculation indicated a satisfactory condition for a sun gear having a P.D. of 2.400 in. The selected number of teeth for the sun gear was 24.

Using $x_1/a = 2.75$, the number of teeth for $X_1 = 24 \times 2.75 = 66$.

Summaries of the roller gear and bevel gear data are presented in Tables I and II, respectively.

Bearing Cam Preload Arrangement

The reaction torque in the selected design is absorbed through the second-row planet bearings. The selected gearbox material was a magnesium alloy. The approximate thermal expansion coefficient difference between the gearbox material and the steel roller gears is 4.9×10^{-6} in./in./degree. It is assumed that the maximum temperature difference between ambient and rotating parts in the box is 180°F. The distance of the second-row planets from the center of the gearbox is 7.794 in. Therefore, the maximum relative growth between the steel parts and the housing is:
$\Delta y = 4.9 \times 10^{-6} \times 180 \times 7.794 = .0068$ in. For proper load distribution in the roller gear drive, it is essential that all rollers stay in proper geometric relation. The motion of the center of the second-row planets, approximately .06 in., will distort the geometry of the roller gear cluster. To eliminate this factor, either a special steel suspension inside the drive is necessary or a special preload cam arrangement for the bearings must be used.

One bearing reaction force from the output reaction torque is equal to 4430 lb. The roller gear planets are subjected to reaction forces through the rollers. The roller cluster force diagram shows that the bearing load is equal to 4700 lb. with an outward radial component. The geometric sum of the two separation forces from the first to the second planet contact is smaller than the inward force from the second planet contact with the ring gear; additional forces are originated by roller contacts.

TABLE I
ROLLER GEAR SUMMARY DATA

A-X_1 6 Contacts Y_1-X_2 12 Contacts X_2-C 6 Contacts	Gear A Sun Gear	Gear X_1 1st Row	Gear Y_1 1st Row	Gear X_2 2nd Row	Gear C Ring Gear
Diametral Pitch	10	10	10	10	10
Pressure Angle	25°	25°	25°	25°	25°
Pitch Dia., In.	2.400	6.600	1.800	7.200	22.800
Number of Teeth	24	66	18	72	228
RPM	11,333	4,121	4,121	1.030	325
Face Width, In.	.872	.872	2.0	2.0	2.0
Torque, In.-Lb	6,115	2,803	2,803	11,210	213,000
Tooth Load, Lb	849.3	849.3	1,557	1,557 & 3,117	3,117
Pitch Line Velocity, fpm	7,120	7,120	1,940	1,940	1,940
Bending Stress, psi*	33,400	33,400	27,500	27,500 48,500	26,300
"K" Factor	482	482	542	542 & 148	148
Hertz Stress, psi	134,750	134,750	133,350	133,350 70,000	70,000
Unit Load $\frac{Wt \times DP}{FW}$	9,740	9,740	7,785	7,785 & 15,570	15,570
Backlash, In.	.006 - .008		.004 - .006		.004 - .008

*Based on the Lewis Formula

TABLE II

BEVEL GEAR SUMMARY DATA

Characteristics	Pinion	Gear
Diametral Pitch	8	8
Pressure Angle	20°	20°
Pitch Dia., In.	4.250	7.875
Number of Teeth	34	63
RPM	21,000	11,333
Spiral Angle	30°	30°
Pitch Angle	28°-21'	61°-39'
Shaft Angle	90°	90°
Hand of Spiral	LH	RH
Rotation	CW	CCW
Face Width, In.	1.125	1.125
Torque, In.-Lb	3,300	6,115
Bending Stress, psi*	26,800	26,800
Hertz Stress, psi	176,000	176,000
Pitch Line Velocity, fpm	23,400	23,400
Backlash, In.	.003 - .005	

*Based on the Gleason Formula

To keep all rollers in contact, the bearing must have slight inward preloading to offset tooth separation forces. The bearing preload can be eliminated with a cam arrangement. The cam mechanism was selected. It consisted of a bearing mounted on an eccentric collar which, through a needle bearing, was fastened to the stationary stud. The eccentric collar was situated so that radial forces on the stud and the reaction force on the bearing, located eccentrically in relation to the stud, created a moment which forced the bearing collar to rotate. This forced the bearing and the roller gear planet to move radially inward until the rollers of the first-row planets generated the reaction force by elastic preload, which produced a reaction moment equal to a moment created by the cam eccentricity. The cam preload was proportional to the transmitted load, and the selected eccentricity was large enough to guarantee a non-separation condition on all the rollers. The cam mechanism eliminated the differential thermal expansion problem. The spherical roller bearings eliminated any reaction to the roller cluster assembly which can result due to distortion of the gearbox housing.

HARDWARE DESCRIPTION

As shown in Figure 2, the drive employed a multiplanetary arrangement of gears and rollers. The attached input right-angle drive consisted of a pair of straddle-mounted spiral bevel gears which reduced the input speed from 21,000 rpm to 11, 333 rpm input to the roller gear drive sun gear. The speed reduction of the roller gear drive was accomplished by a first row of six staggered inner planet gears (three narrow and three wide), driven by the sun gear, and a second row of six outer planet gears, driven by the inner planets (which, in turn, drove the integral ring gear and output shaft at a speed of 325 rpm). Automatic centering or preloading of the roller gear elements was accomplished by cam action on the six outer planets to compensate for tolerance stackups, differential thermal expansion, and operational deflections. Preload automatically increases as a function of output torque.

Roller Gear Drive Components

The sun roller gear shown in Figure 3 is a hollow, cylindrical, multiple-piece element with an internal, centrally located input spline to minimize effects of torsional windup across the face of the gear teeth. The inner gears, which meshed with the three narrow first-row planets, are an integral part of the sun gear shaft. The outer gears, which meshed with the three wide first-row planets, were splined to each shaft end. The teeth of the four gears, A, were integrally cut and finish-ground on both ends of the cylinder.

The outer gears were then removed from the splined shaft ends, and the roller elements were press-fitted to the shoulders on either side of the two outer gears. The gear/roller assemblies were then reinstalled on the shaft splined ends, and the rolling surfaces were finish-ground and lapped

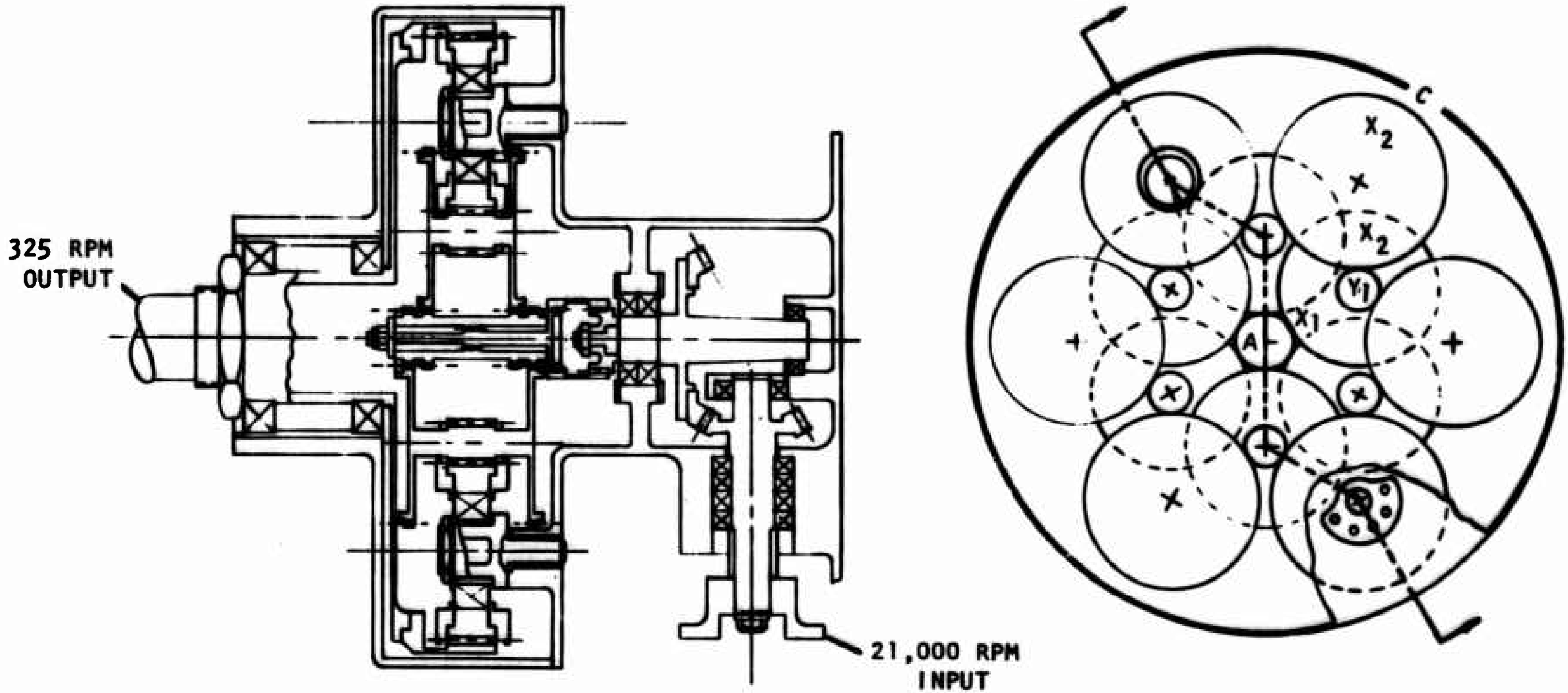


Figure 2. 1100-hp Roller Gear Drive.

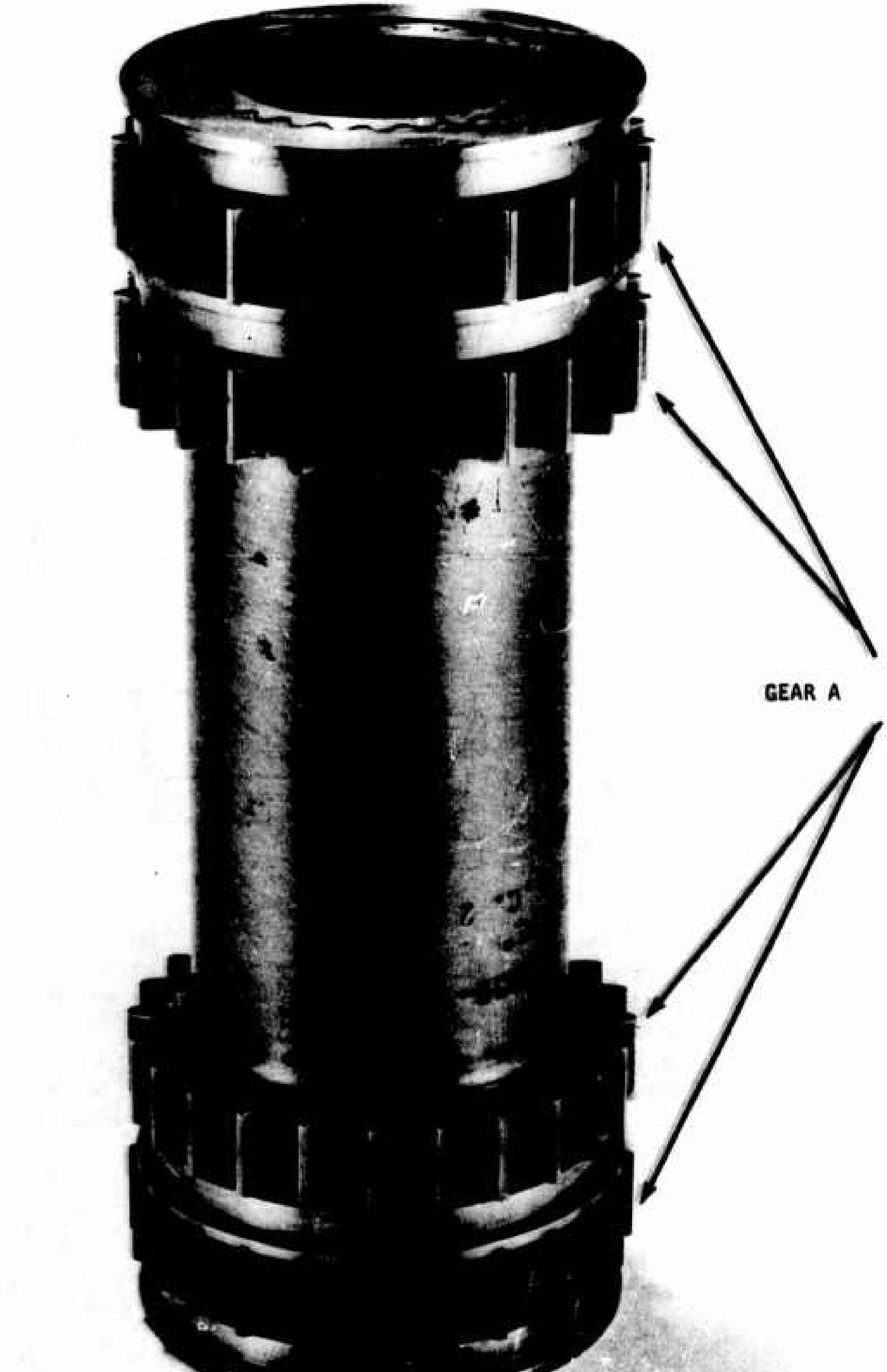


Figure 3. Sun Roller Gear.

to conform with and be concentric with the pitch diameter of the sun gears. Flanges were provided on the four rollers to provide axial retention of the sun roller gear element.

There are six staggered, equally spaced planet roller gears surrounding the input sun roller gear in the first row. Each of these first-row roller gears consists of three gear elements: two larger gears, X_1, at each end and a smaller gear, Y, at the center.

Initially the smaller gear, Y_1, was ground to final size, after which the two larger gears were pressed and electron-beam welded to the shaft extensions of the smaller gear, Y_1. The larger gears, X_1, were then ground to maintain a tooth index relationship with the Y_1 gear of plus or minus .0001 in. At this time the two roller surfaces on either side of the Y_1 gear were finish-ground and lapped to conform with and be concentric with the pitch diameter of the Y_1 gear. Finally, the two larger rollers were pressed and then locked in place on the ends of the two larger gears, X_1, and were ground and lapped to conform with and be concentric with the pitch diameter of the X_1 gears.

This assembly constitutes the X_1/Y_1 roller gear element shown in Figures 4 and 5. The two large gears, X_1, along with their corresponding matching rolling surfaces, are in contact with the rolling and gear surfaces of the sun roller, A, as shown in Figures 6 and 7, providing six equally spaced line contacts.

Surrounding the first-row roller gears are six equally spaced planet roller gears, X_2, comprising the second row (Figure 8). The second-row planet gears turn on spherical roller bearings which are mounted on eccentric needle bearing sleeves, one end of the sleeves being flanged to retain the roller gear inner race and the other being threaded to accept the bearing lock nuts. Two rows of needle bearings are located between the eccentric sleeve inside diameters and the stationary shafts, which are piloted and bolted to the roller gear drive housing. The planet gear rollers are piloted and bolted to each side of the gear body to allow removal and subsequent assembly or disassembly of the roller drive gears. The roller surfaces were finish-ground and lapped to conform with and be concentric with the pitch diameter of the X_2 gears. Flanges were provided on the two rollers to furnish axial retention of the planet gears (Figure 9). A retainer plate, which retains the needle bearings and eccentric sleeve, was fastened to the end of the stationary shaft by four flat-head screws.

The eccentric sleeve, or cam, in each of the second-row outer planet gears provides a .060-in. offset or eccentricity of the gear with respect to the stationary shaft. This cam action provided automatic centering or preloading of the gears with torque application to compensate for tolerance stackup, differential thermal expansion, and operational deflections.

The second-row outer planet roller gears, X_2, transmit torque to the ring gear C, at six equally spaced contacts. The ring gear is open at the drive input end, and the output end has a piloted mounting flange which is doweled

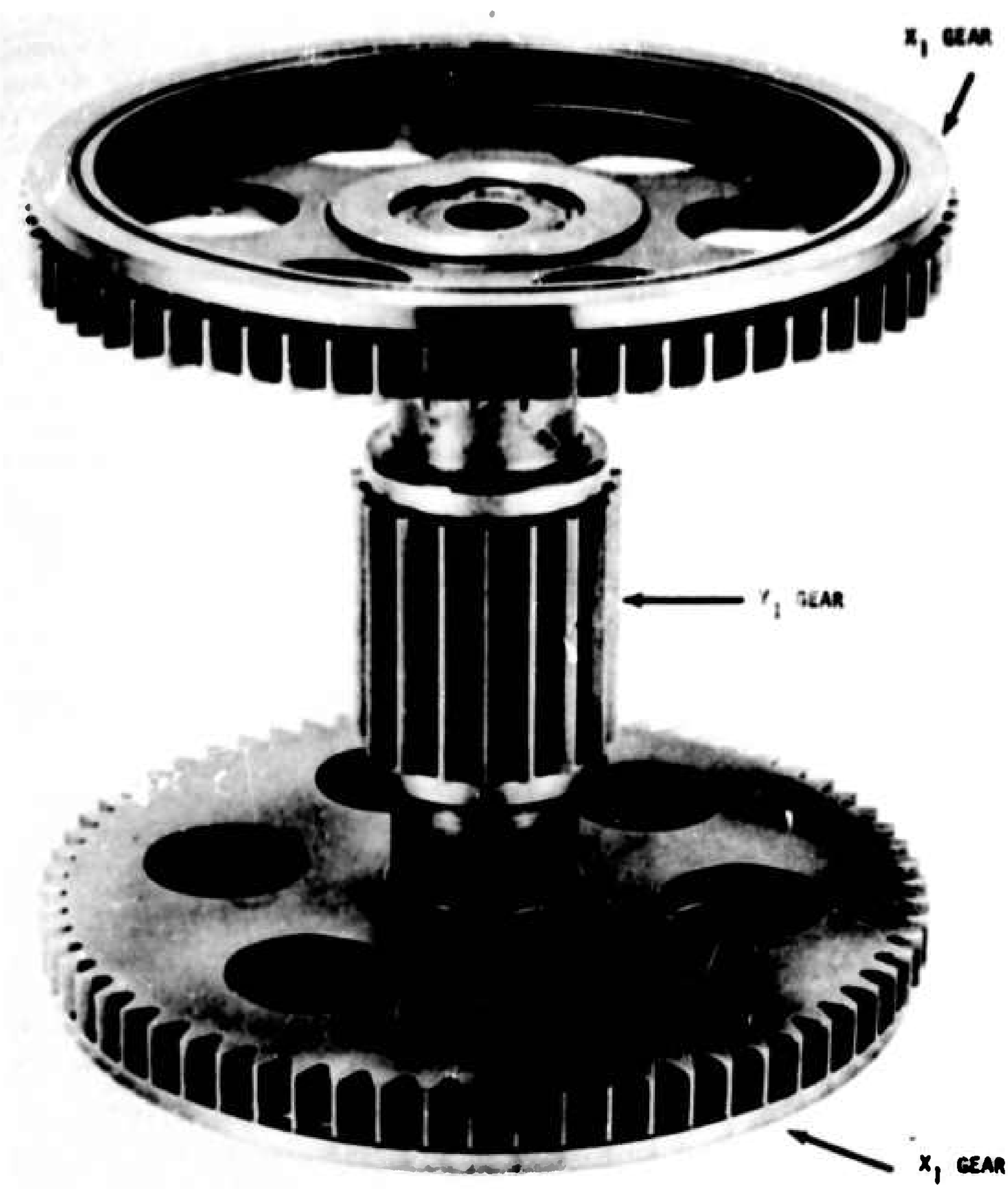


Figure 4. First-Row Planet Roller Gear (Wide).

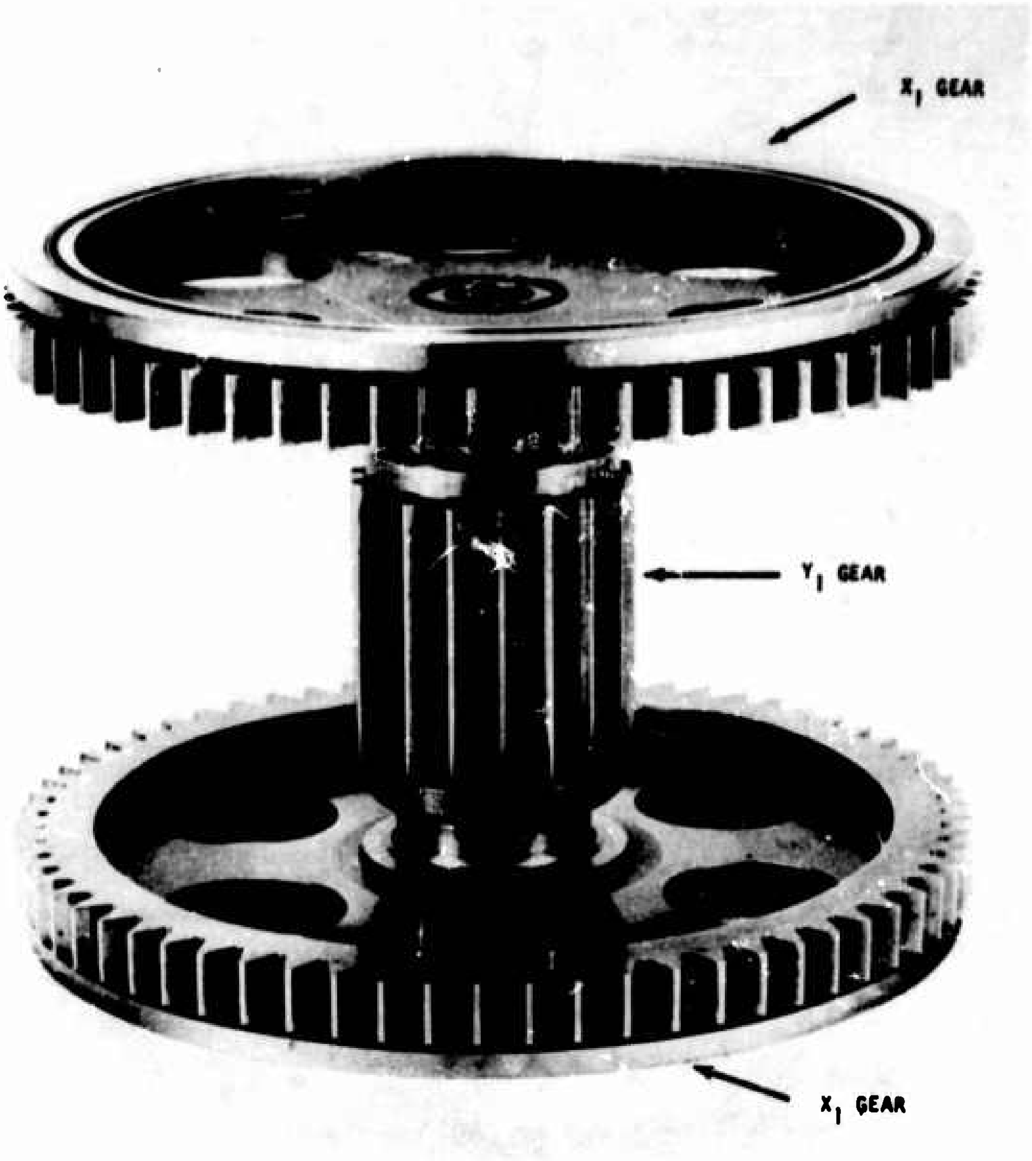


Figure 5. First-Row Planet Roller Gear (Narrow).

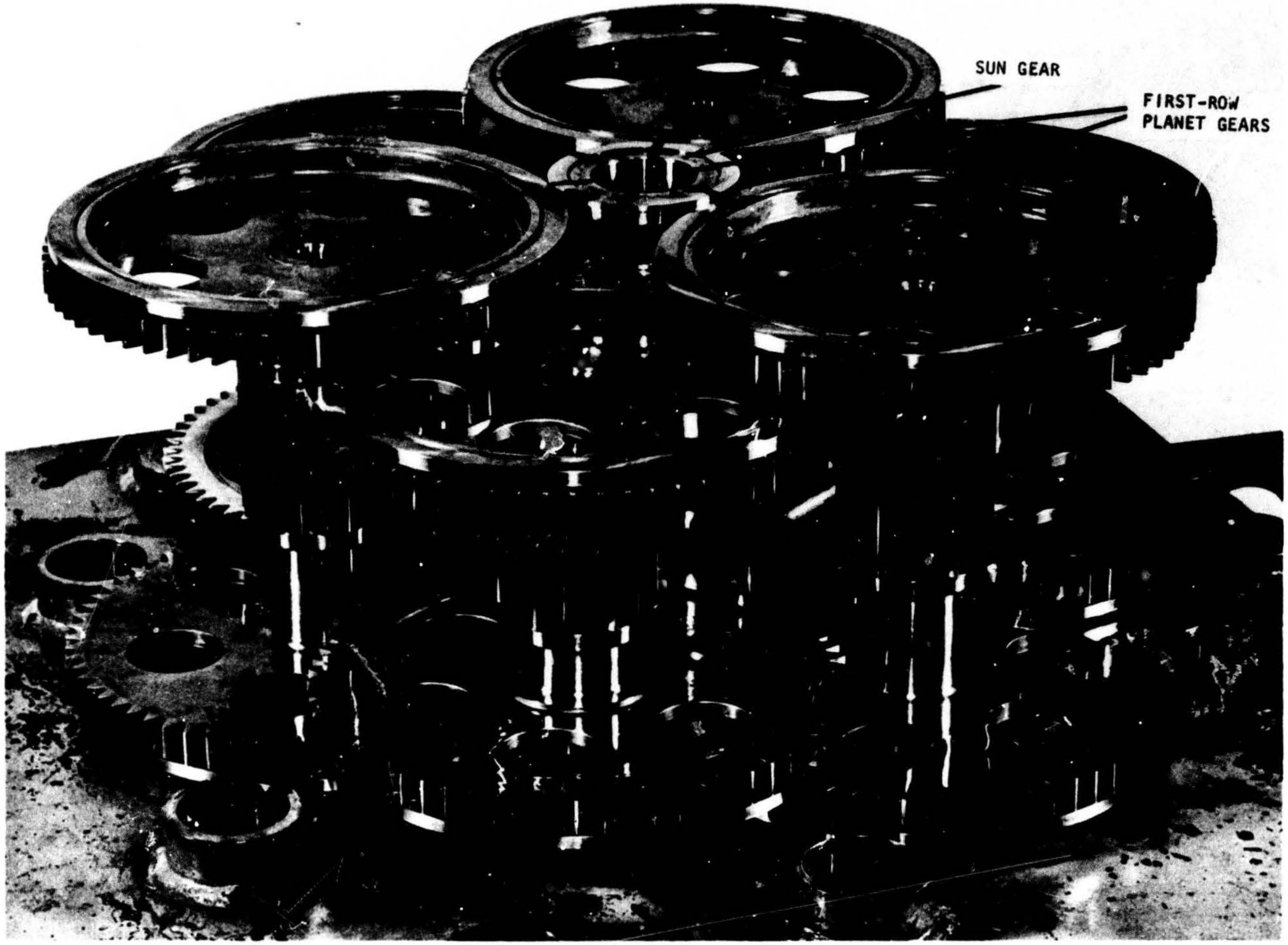


Figure 6. Sun To First-Row Planet Gearing.

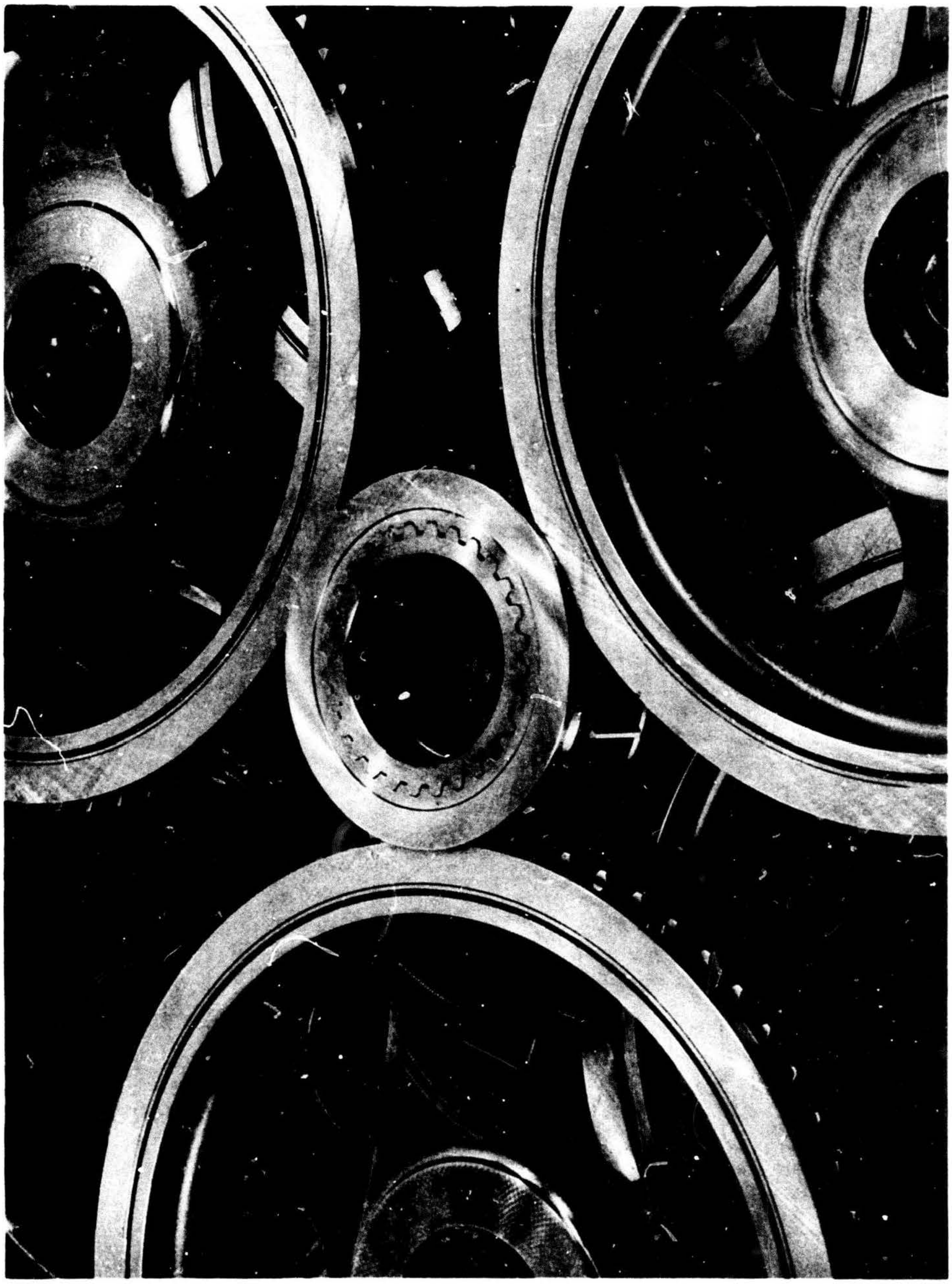


Figure 7. Sun Roller Gear Contact.

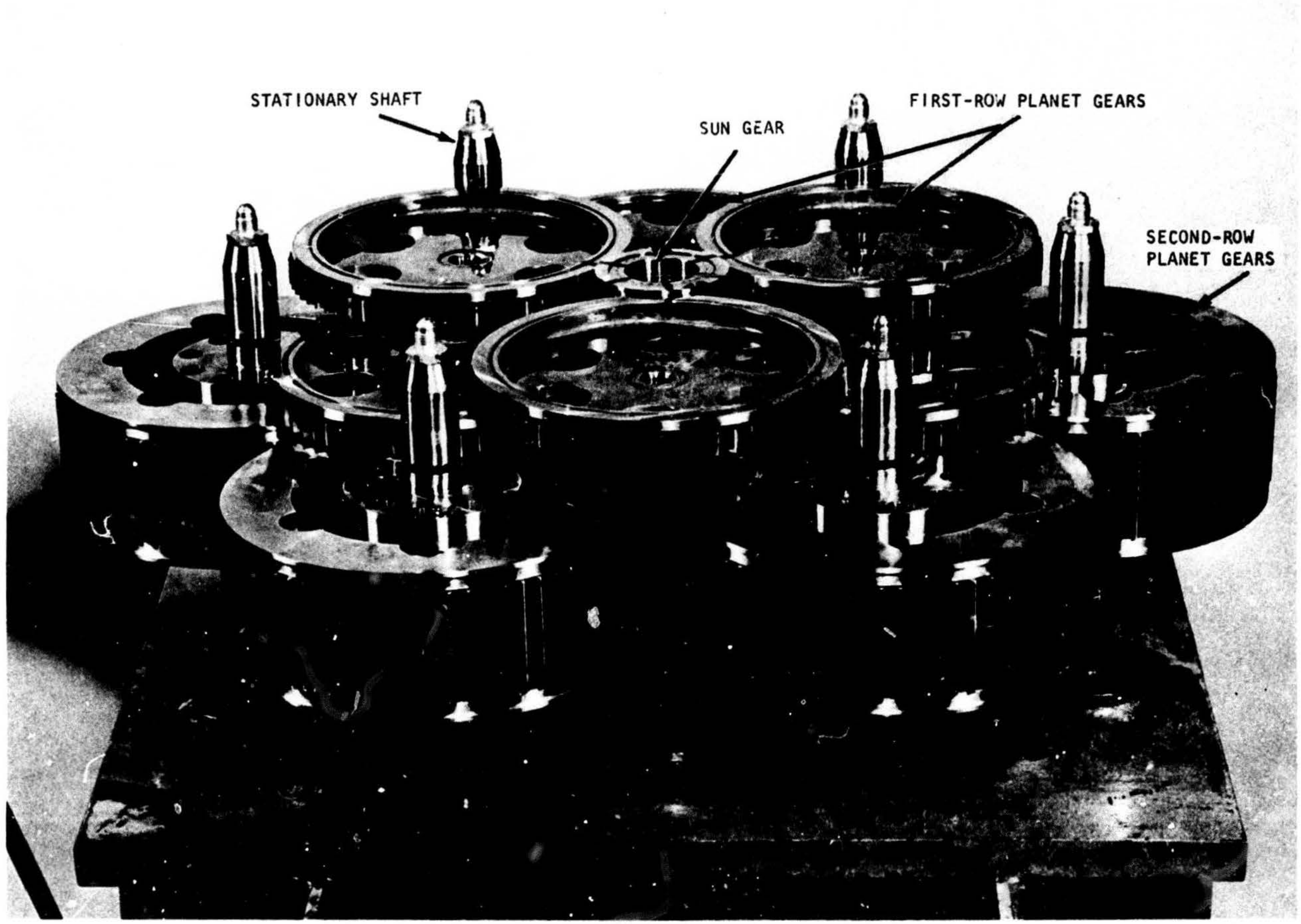


Figure 8. Second-Row Planet Gearing.

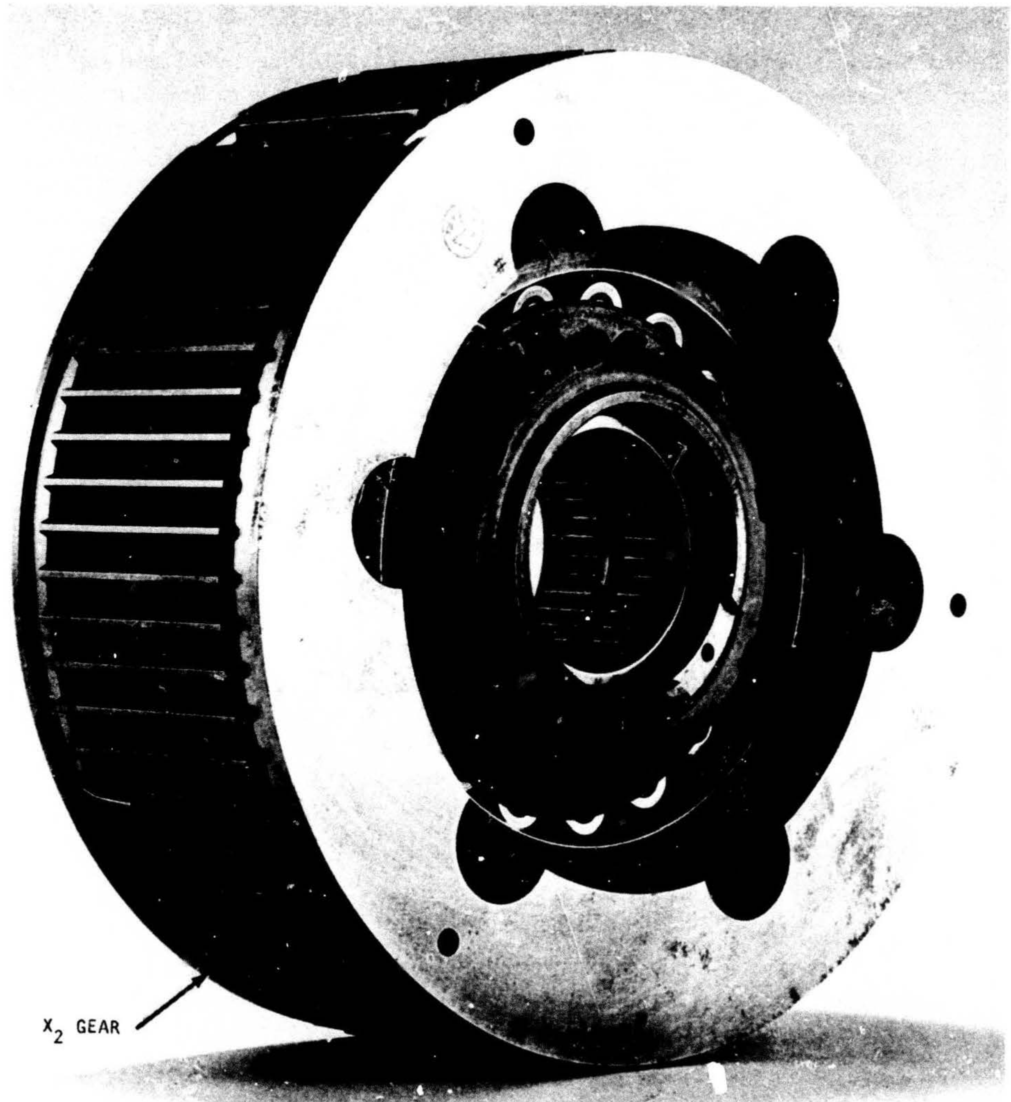


Figure 9. Secoı d-Row Planet Roller Gear.

and bolted to the output drive shaft flange (Figures 10 and 11). The output shaft/ring gear assembly was mounted on two angular contact ball bearings (mounted back to back and spaced 4.5 in. between ball centers) at the output shaft end. The output shaft has an internal spline to accept the output torque coupling adapter.

The second-row planets are retained axially by the needle bearing and eccentric sleeve retainer washer, which is bolted to the stationary shaft, which, in turn, is bolted to the drive housing. The second-row roller outer flanges retain the first-row planet gears at the roller contacts on either side of the small Y_1 gear. The large gear rollers at either end of the first-row planet gears retain the sun gear axial location through positioning of the sun gear roller flanges on either side of the first-row large rollers. Thus, axial locations for the first-row gear elements and sun gear were provided by the flanges of the roller contacts. These contacts maintain the roller gear elements in precise alignment, resulting in excellent gear contact. No high-speed, anti-friction bearings with their resulting looseness and location tolerances are required.

The roller gear drive housing assembly consists of the flanged housing or spider, which supports the six second-row roller gears, X_2; stationary shafts and subsequently all roller gear elements; and the bell housing or cover, which houses the ring gear, C, and output drive shaft. The two halves of the housing assembly are piloted, pinned two places 180° apart, and bolted together at eighteen places on a 25.750-in.-diameter bolt circle (Figure 12).

The roller gear support housing was machined from a magnesium casting. Six equally spaced bosses were provided for support of the second-row roller gear shafts. Stainless steel liners were pressed into 1.177/1.197-in.-diameter through-holes in each of the bosses. When assembled with the cover, these liners were bored and reamed to 1.0000/1.0007-in.-diameter to slip-fit with the roller gear shafts. At this time, five .457/.472-in.-diameter through-holes were drilled around each of the six liners for attachment bolts for installing the roller gear shafts (Figure 13). Six equally spaced 2.000-in.-diameter inspection holes were provided for inspection of the second-row roller gears, X_2, and ring gear, C. These were sealed at assembly by installing flat disk plates with annular O-ring seals. The disks were secured in the holes by installing snap rings on either side of the disk plates.

This support housing contains the mounting flange with twelve bolt holes for attaching the bevel gear duplex bearing support and the bevel gear housing. Three through-holes and two threaded bosses were provided for installing the roller gear lube system manifold. The housing pilot (24.870/24.866-in.-diameter) contains an annular O-ring seal when assembled with the cover.

The roller gear cover was also machined from a magnesium casting. The ribbed housing contains a stainless steel bearing liner in the output

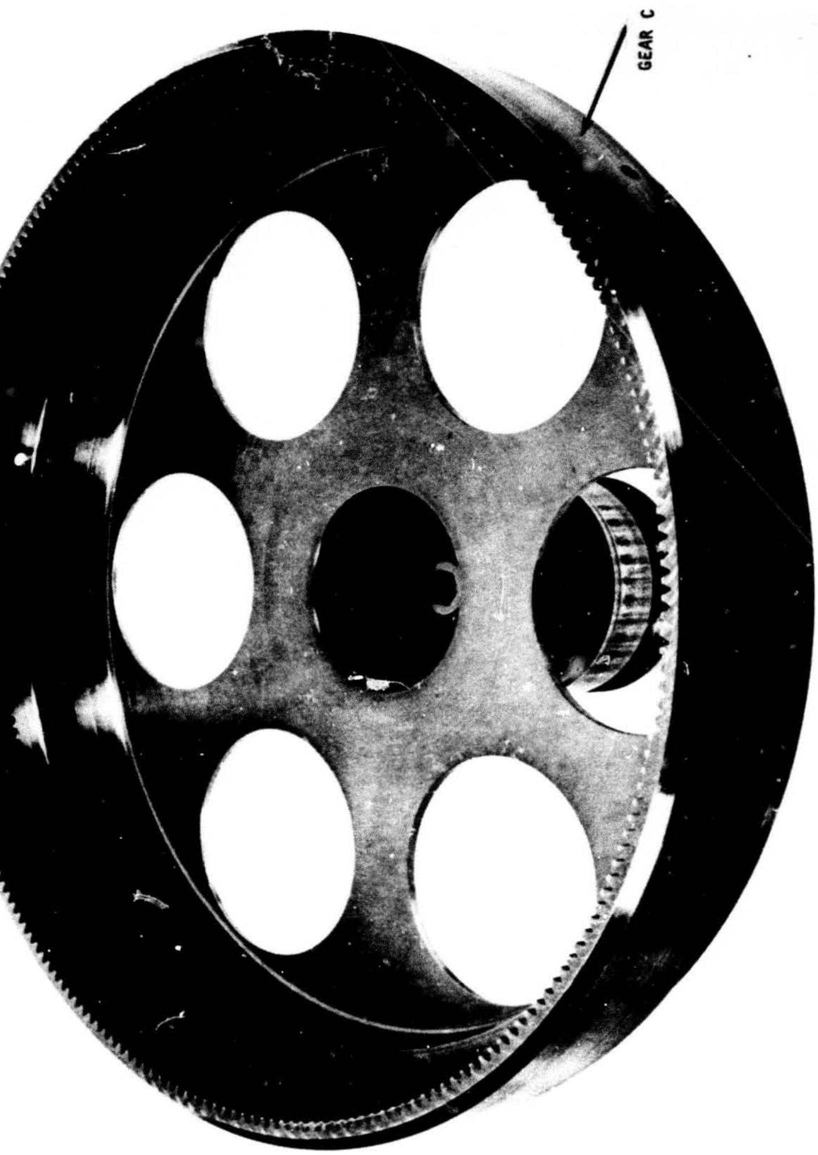


Figure 10. Ring Gear.

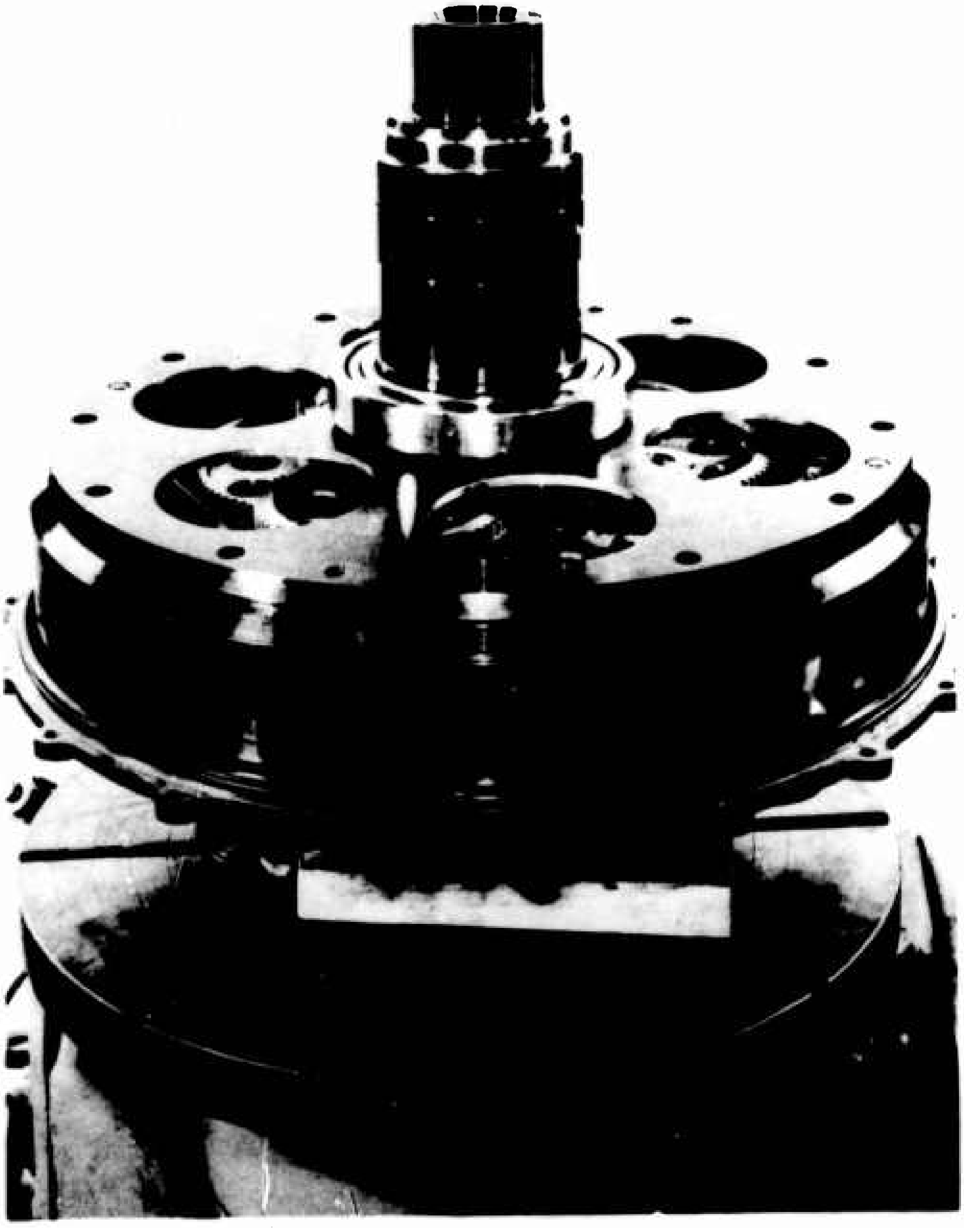


Figure 11. Assembled Ring Gear.

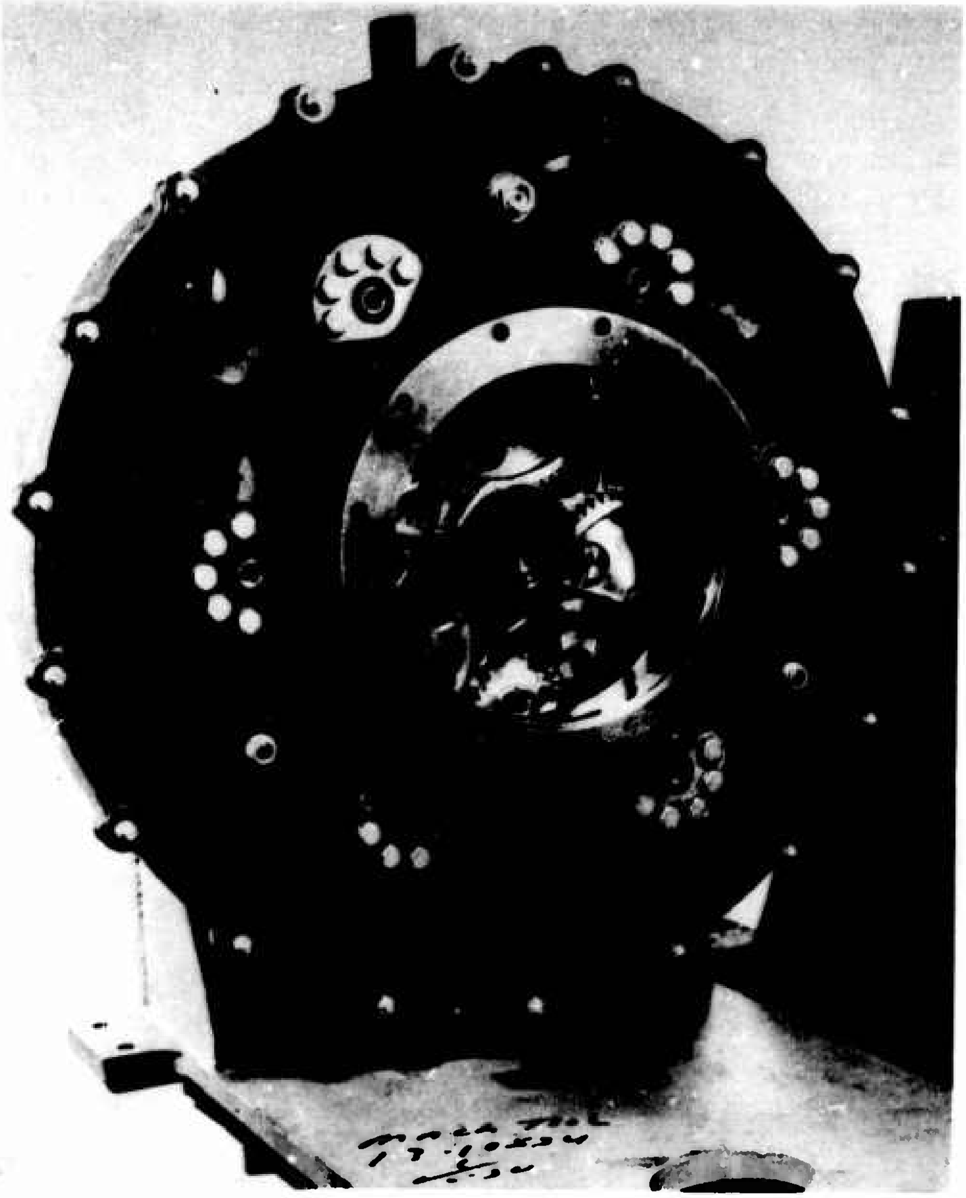


Figure 12. Roller Gear Support Housing.

Figure 13. Roller Gear Housing Cover.

shaft bearing cartridge bore. The inner end of the liner is flanged and attached to the housing inner wall with four flat-head screws which are staked in place. The bottom of the cover contains a cast boss with a 2.5-in. tapped hole to accept an oil drain plug with chip detector. At the top front of the cover is a tapped hole for a gear-case vent plug. The output shaft end of the cover contains six studs for attachment of the output shaft seal cartridge. The open end of the cover has a 24.879/24.875-in. ID pilot and eighteen studs for attachment with the roller gear housing (Figure 13). Tapped bosses were provided for lifting eyes in two planes.

At assembly of the roller gear housing and cover, two holes were drilled and reamed 180° apart at the outer bolting flange to accept locating pins which are press-fit in the cover and slip-fit in the housing. At this time the six shaft liners in the roller gear housing were machined to final size and within .001-in. true position of the axis established by the cover output shaft bearing liner bore diameters. The pilot diameter at the interface with the bevel gear support was also machined concentric within .002 in. total indicator reading with this same axis.

All gears in the roller gear drive, with the exception of the ring gear, were made from standard AMS-6260 gear steel, R_c 60 hardness, and finish-ground for a surface roughness of 32 μ in. The ring gear was made from 4340 steel heat-treated to R_c 33-38 hardness. The gear teeth were shaped with a surface roughness of 63 μ in. All roller elements, with the exception of the first-row small gear, Y_1, rollers, were made from standard 52100 bearing steel, R_c 61-65 hardness, and finish-ground and lapped for a surface roughness of 8 μ in. The first-row small gear rollers are part of the outer large gears and were made from the AMS 6260 gear steel. The small rollers were carburized and hardened to R_c 60 with finished effective case depth of .050-.065 in., and finish-ground and lapped for a surface roughness of 8 μ in. The output shaft was made from 4340 steel heat-treated to R_c 33-38 hardness. The internal spline teeth were shaped with a surface roughness of 63 μ in.

The drive shaft, which transmits torque from the driven bevel gear to the roller gear drive sun gear, A, through a coupling and coupling adapter, engages at the sun gear internal spline. The drive shaft, which is reduced to 1/2-in. diameter beyond the internal spline engagement, extends through the sun gear ID and is threaded on the end. A washer with pilot shoulder which slip-fits into the sun gear ID was placed on the end of the drive shaft threaded end, and a 1/2-in. lock nut was installed on the shaft. This draws the sun gear stackup together and places the sun gear end thrust collars in bearing on the drive shaft shoulder at the input end and on the washer flange at the opposite end. The drive shaft input or large spline is identical to the coupling adapter spline, which was internally splined and locked to the bevel gear output shaft with a flat washer and lock nut (Figure 14). The two splines were engaged with an internally splined drive coupling. The coupling is axially retained by two internal snap rings, which limit axial movement at either spline end.

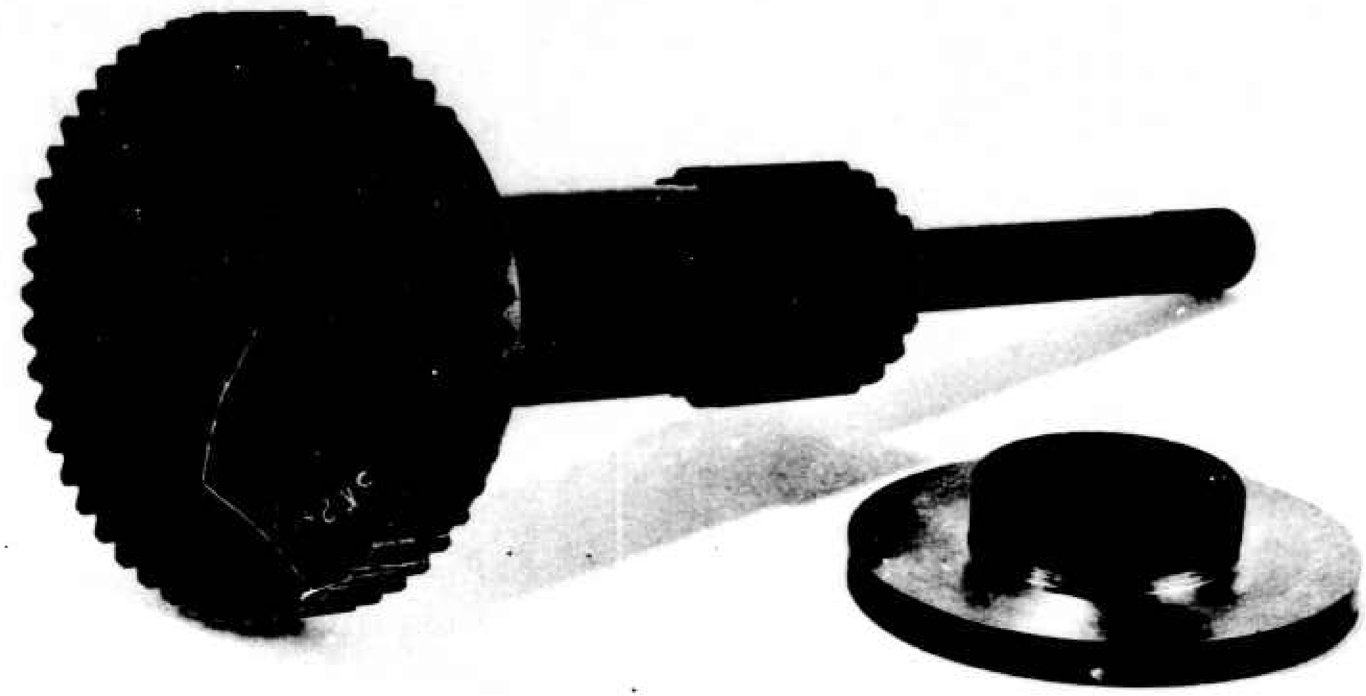


Figure 14. Drive Shaft.

The drive shaft, coupling, and coupling adapter were made from 4340 steel, heat-treated to R_c 33-38 hardness. All spline teeth were shaped with a surface roughness of 63μ in.

The total reduction of the roller gear drive (minus the bevel gear right-angle reduction) is:

$$\frac{x_1}{A} \times \frac{x_2}{y_1} \times \frac{C}{x_2}$$

A = 24 teeth

x_1 = 66 teeth

y_1 = 18 teeth

x_2 = 72 teeth

C = 228 teeth

$$\text{Reduction ratio} = \frac{66}{24} \times \frac{72}{18} \times \frac{228}{72} = 34.83$$

The approximate dimensions of the roller gear drive are 25.75 in. diameter x 18 in. length. Figures 15 and 16 portray the assembly of the drive and all basic elements. Refer to Table I for gear data.

Bevel Drive Components

The right-angle bevel gearboxes, one mounted on each of the two roller gear drives, enable the roller gear drives to be mounted in a back-to-back (four-square loop) test stand. The high-speed bevel gear input shafts of each gearbox face each other. Thus, the system with a high-speed torque coupling and low-speed closing gearbox can be preloaded to the desired torque. The right-angle bevel gear drive reduction is 1.85; thus, input speed of 21,000 rpm is reduced to 11,333 rpm into the roller gear drive.

The spiral bevel pinion is straddle-mounted. The input end of the gear shaft was mounted on a stack of three angular contact bearings and a roller bearing. Two of the angular contact bearings nearest the gear were mounted in tandem to carry gear thrust loads. The third bearing was mounted back to back with the second to carry reversing thrust loads. The outer roller bearing carried shaft radial loads. The opposite end or gear shaft stub end was mounted on a single roller bearing which carried the highest gear radial loads (Figure 17).

The three angular contact bearings and roller bearing were retained as a cartridge by a stainless steel flanged liner, which in turn was pressed

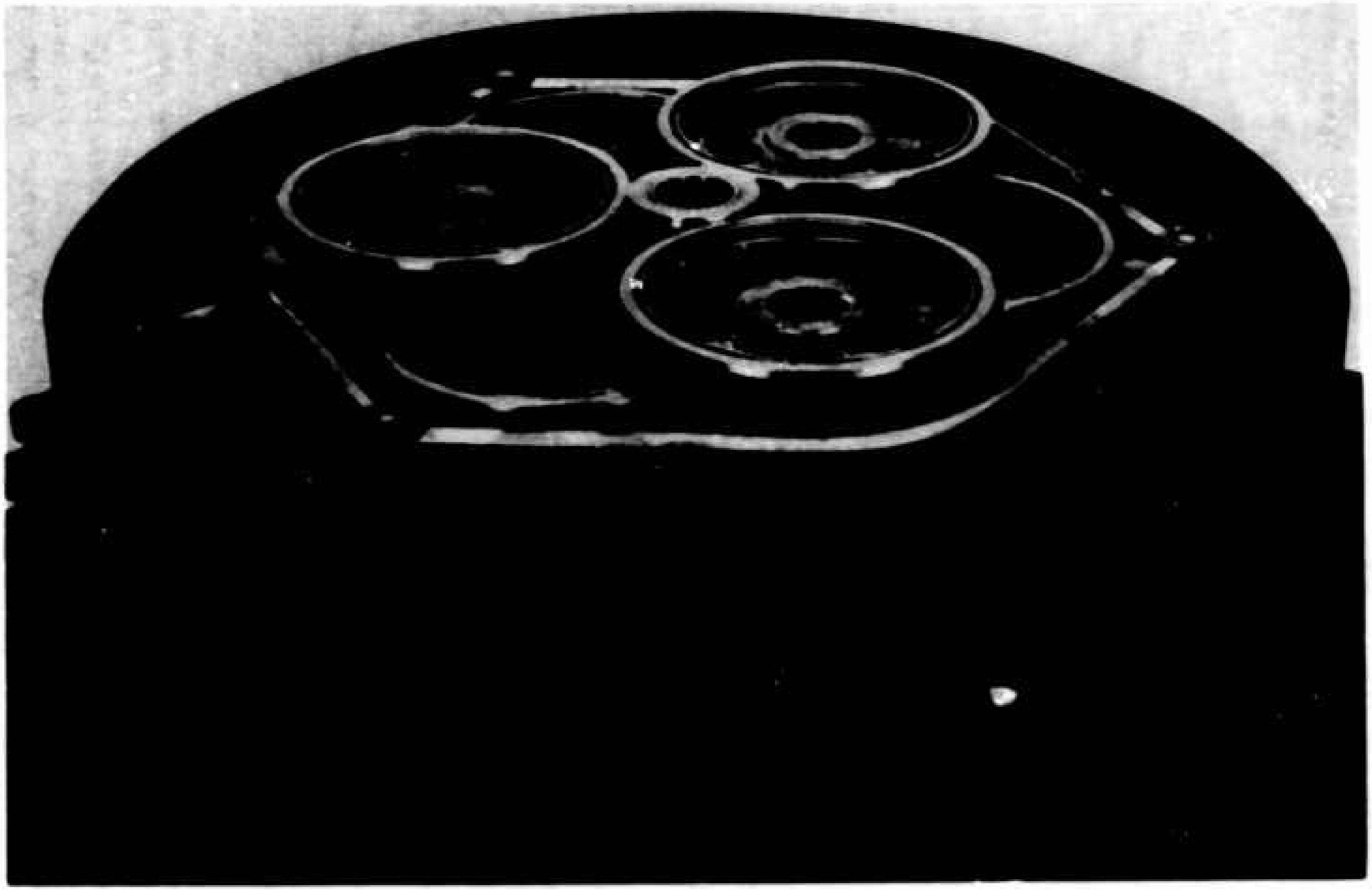


Figure 15. Roller Gear Drive Basic Assembly.

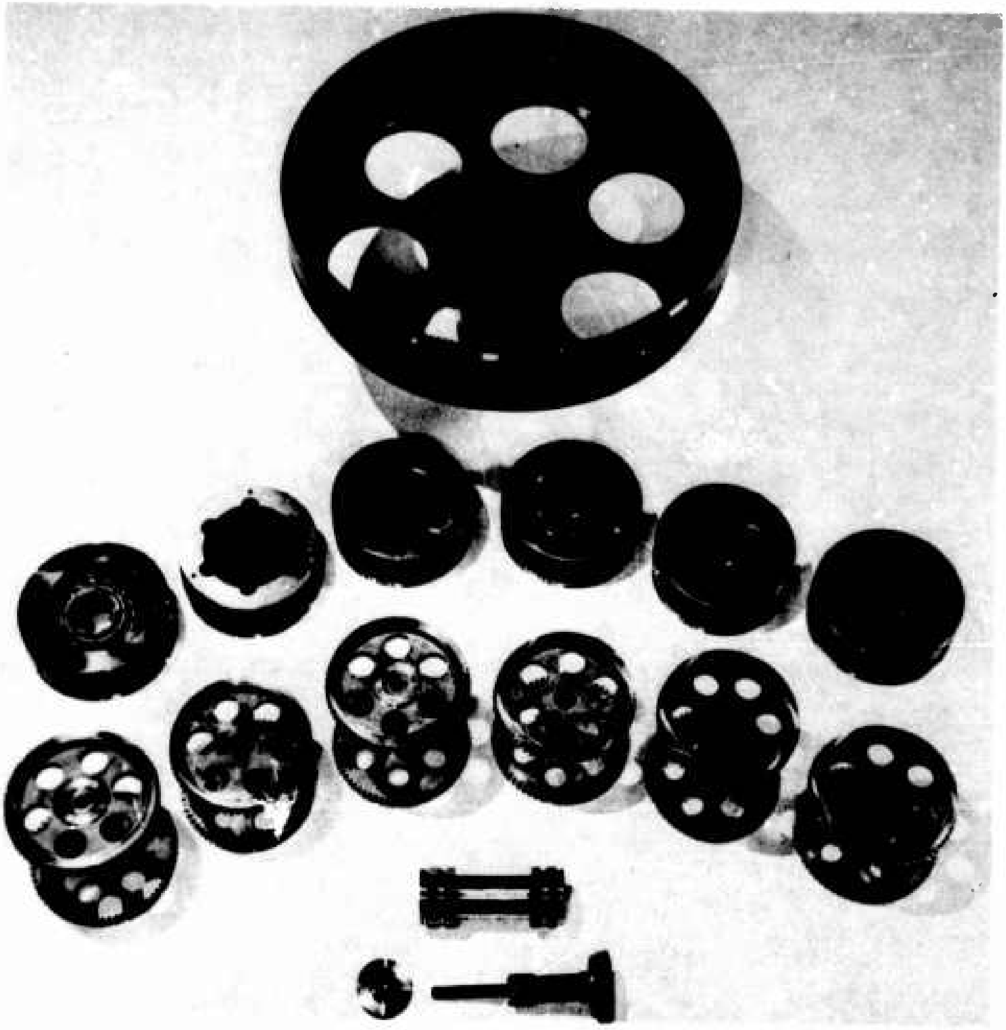


Figure 16. Roller Gear Drive Basic Components.

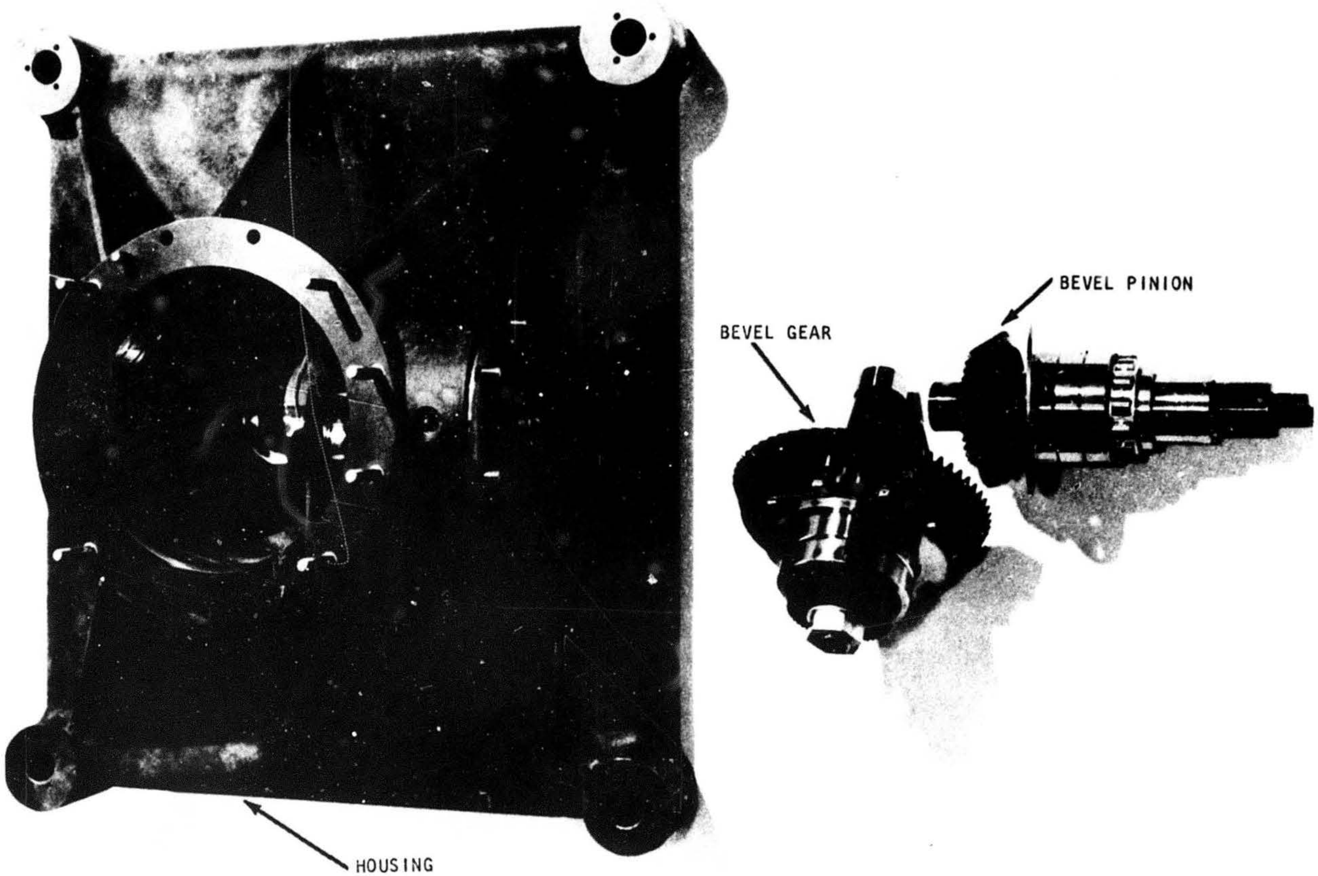


Figure 17. Bevel Drive Components.

into a magnesium bearing housing. A steel bearing retainer plate, through-bolted four places to the end of the bearing housing, retained the bearing stack and assured correct bearing preload. The bearing housing outer end was flanged with six equally spaced through-holes for attachment to the bevel gear housing with its six attachment studs (Figure 18).

The inner roller bearing is retained by a stainless steel liner which has an inner flange on one end and a snap ring on the other to retain the bearing. This liner was pressed into a bore concentric with the other bearing cartridge bore in the bevel gear housing (Figure 19).

The three angular contact bearings, roller bearing, bearing liner and retainer plate, bearing housing, seal gasket face seal rotating ring, face seal and seal housing, shaft seal spacer, and coupling flange which is keyed to the shaft were stacked on the spiral pinion shaft as a subassembly and retained by a flat washer and lock nut on the threaded input end of the pinion shaft. The pinion subassembly was inserted into the bevel gear housing bore as a cartridge and positioned for correct gear backlash (.003 - .005 in.) by inserting a machined steel shim between the bearing housing flange and bevel gear housing mounting face.

The bevel pinion/shaft was made from standard AMS 6260 gear steel. The gear teeth and shaft stub end roller bearing journal surfaces were carburized and hardened to R_c 60. The gear teeth were finish-ground for a surface roughness of 25 μ in. The bearing journal surface was ground to a 8 μ in. finish.

The spiral bevel-driven gear is straddle-mounted. The output end of the gear shaft was mounted on a pair of angular contact bearings (mounted face to face). These bearings carry the gear thrust loads plus most of the radial load. The bearings, along with the output coupling adapter, were retained by a flat washer and lock nut on the threaded output end of the gear shaft. The opposite or inner end of the gear shaft was mounted on a single roller bearing (Figures 20 and 21).

The two angular contact bearings were slip-fit into a stainless steel liner, which in turn, was pressed into the magnesium bearing support. The bearings were retained axially by retainer plates at either end which were bolted to the bearing support (Figure 20).

The inner roller bearing is retained by a stainless steel liner which has an inner flange on one end and a snap ring on the other to retain the bearing. This liner was pressed into a bore concentric with the bearing retainer pilot diameter in the bevel gear housing (Figure 19).

The driven gear was positioned axially for the correct mounting distance by a shim plate that was inserted between the bearing retainer and housing at the mounting flange interface (Figure 21).

Gear teeth of the gear ring were finish-ground for a surface roughness of 25 μ in. prior to the gear ring's being bolted to the shaft mounting flange. The gear ring was made from standard AMS 6260 gear steel; the gear teeth surfaces were carburized and hardened to R_c 60. The gear shaft was made

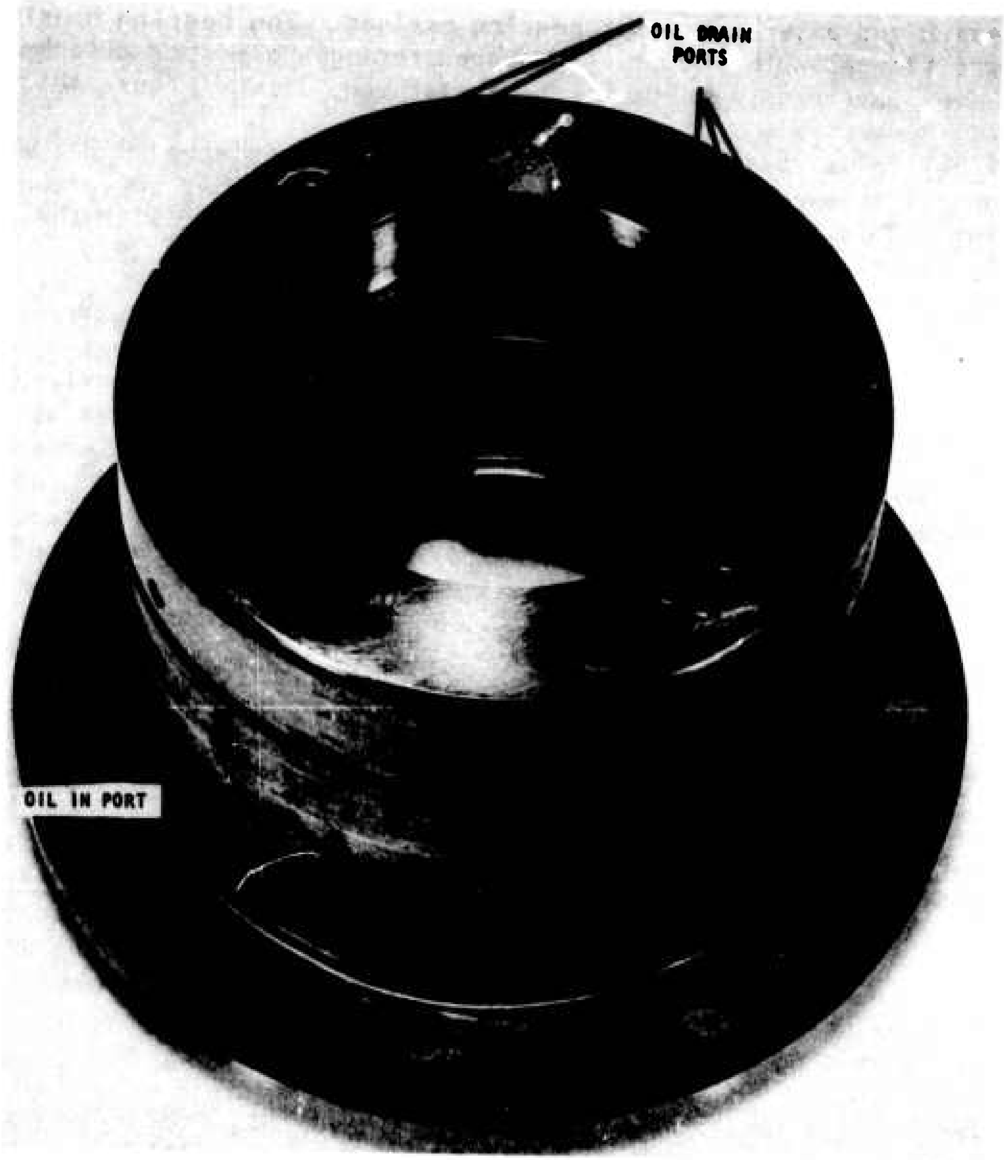


Figure 18. Bevel Pinion Bearing Housing.

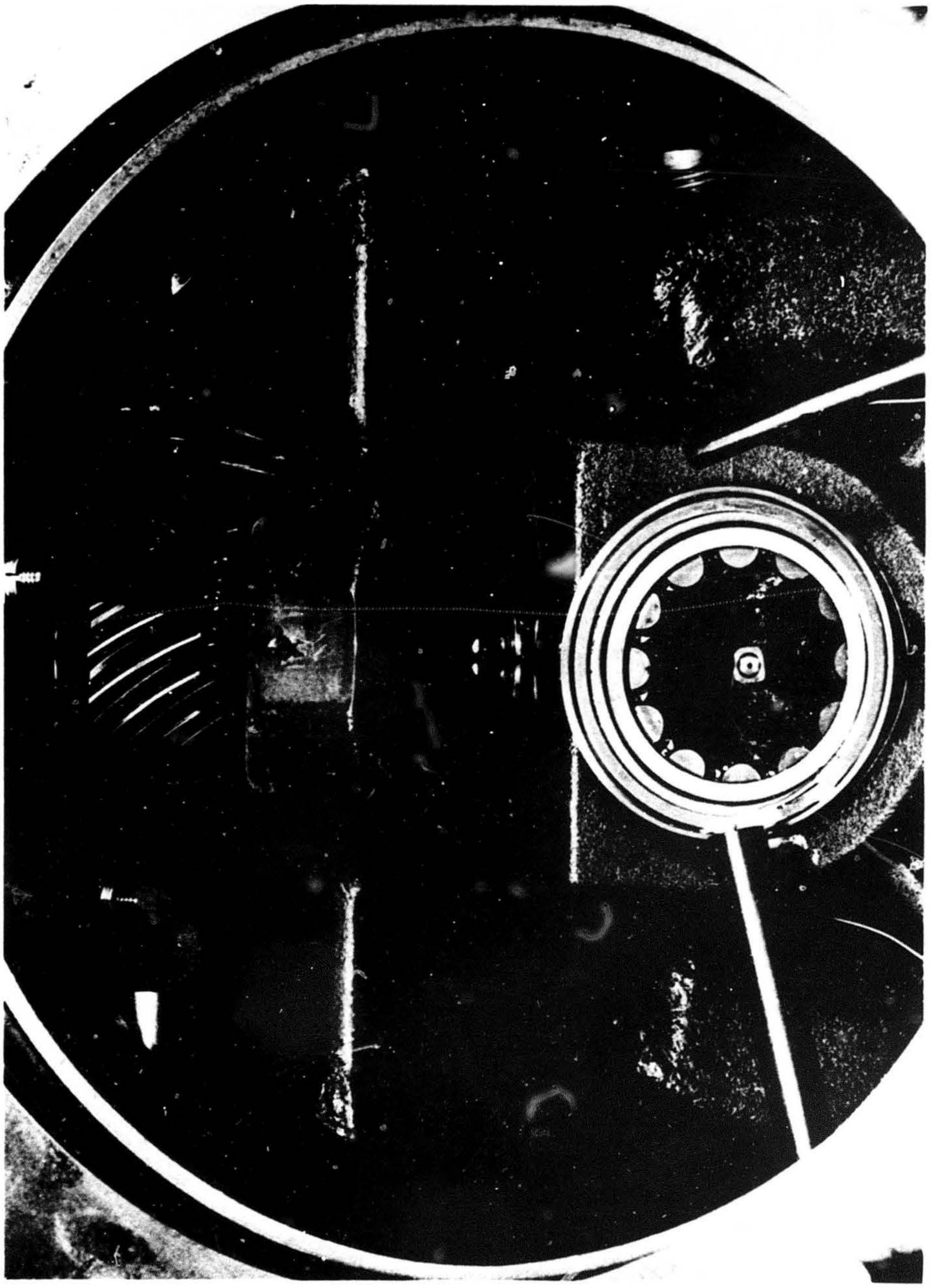


Figure 19. Bevel Gear and Pinion Inner Roller Bearings.

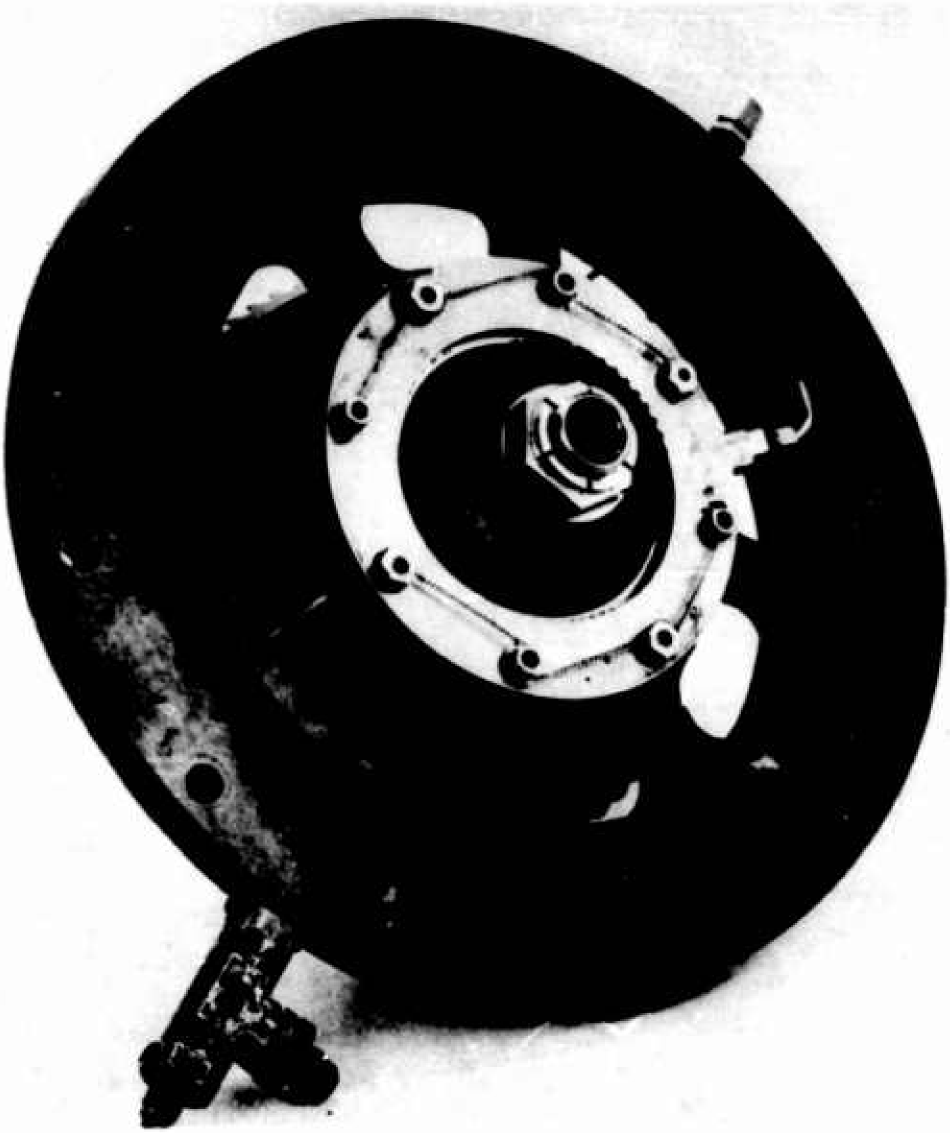


Figure 20. Bevel Gear/Shaft and Bearing Support Assembly.

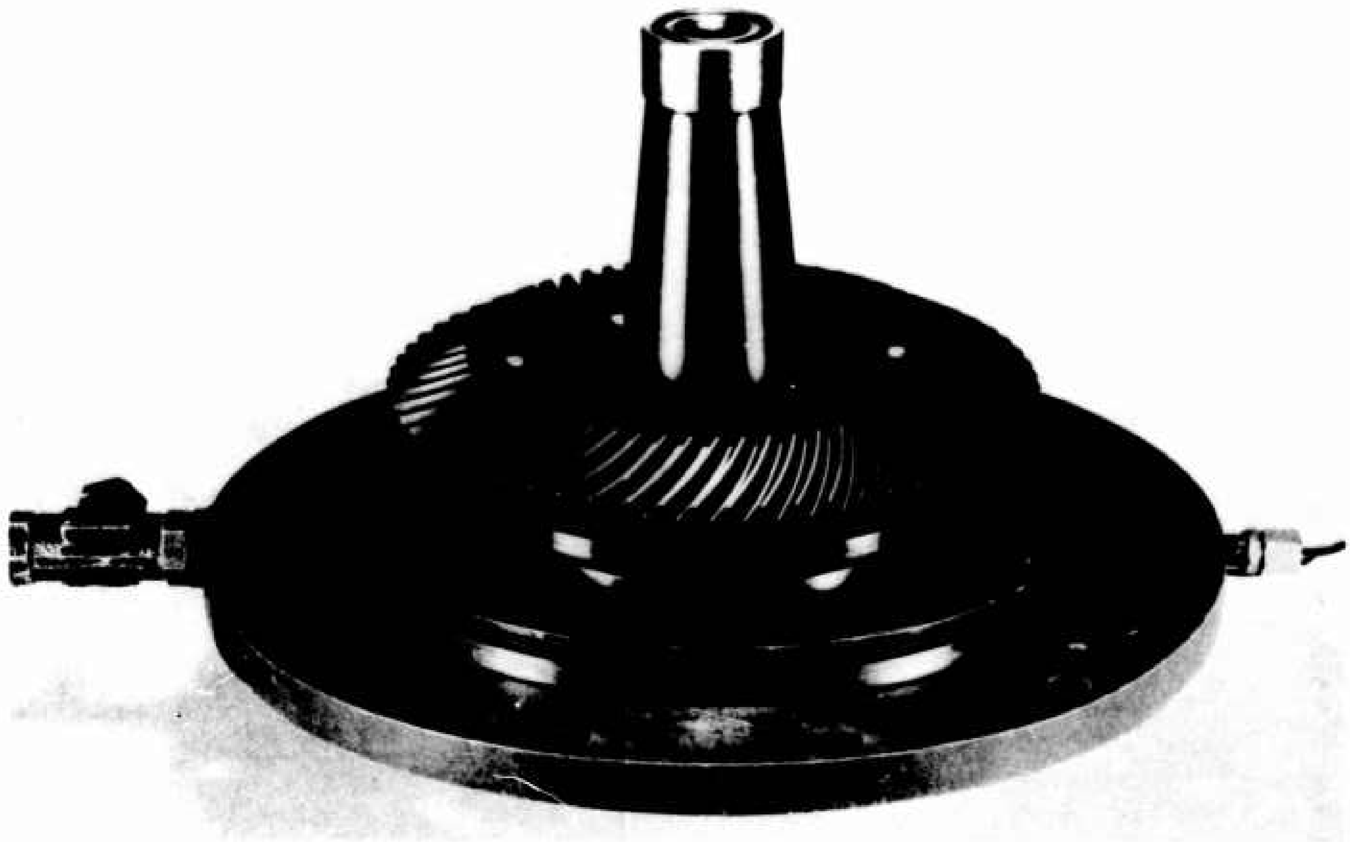


Figure 21. Bevel Gear/Shaft and Bearing Support Assembly.

from 4340 steel, heat-treated to R_c 33-38 hardness. Refer to Table I for gear data.

The right-angle bevel gear housing was made of cast magnesium. The interface with the roller gear housing is at the output shaft bearing support with twelve 3/8-24 studs on a 11.625-in.-diameter bolt circle.

Four cast bosses form the corners of the support plate; 1.125-in.-diameter through-holes in each boss on centers 21.750-in. x 24.750-in. were used for pinning the bevel gear housing to the rig support structure (Figure 17).

Lubrication

Every gear mesh in the roller gear drive was lubricated by a pressure jet, generally oriented in the direction of the mesh and where possible, on the outgoing side of the mesh. The roller contacts were lubricated only from the ensuing oil spray of the gear meshes. A gravity drain was used for each test unit, with electric chip detectors mounted in the drain adapter plugs.

The bevel gear mesh and inboard high-speed and low-speed roller bearings were lubricated by pressure jets. The pinion outer bearing cartridge and driven gear shaft duplex bearings were pressure lubricated through drilled passages.

A pump, pressure regulator, filter (10 micron), flowmeter, pressure shutdown switches, heat exchanger, and sump are used on the basic test rig. The lubricating oil selected for the test was Humble Oil Turbo 35, a commercially available oil similar in many characteristics to oils meeting the MIL-L-7808 specification, but higher in viscosity and with a higher load-carrying capability. Lube system characteristics of the roller gear drive and bevel gear drive are:

1.	System pressure:	30 psi minimum
2.	Flow per drive:	Roller gear drive, 6.2 gpm Bevel gear drive, 3.5 gpm
3.	Heat rejection per drive:	Roller gear drive, 890 Btu/min Bevel gear drive, 465 Btu/min
4.	Inlet temperature maximum:	175°F

TEST METHOD

TEST STAND

The test system employed the principle of back-to-back coupling (four-square loop) of two test units in a closed-power path. The two output shafts of the test units were connected by flexible couplings to the output shafts of the closing gearbox. The closing gearbox is a standard industrial-type reducer unit with a fixed reduction ratio of 5.65:1. The input shafts of the test units were connected to each other with a torque coupling forming a closed-power path. The torque coupling consisted of two plates with holes and slots permitting the attachment of a torque bar with calibrated weights to apply known torque loads to the system. This preloaded torque was applied statically and locked with bolts in the flanges of the torque coupling, after which the torque bar and weights were removed. The input shaft of the closing box was connected to a driving dynamometer, which has a speed control that provides variable speeds and measures the total delivered torque to the closed-power-loop transmission system. Also, a lubricating and cooling system was provided for the transmission system.

Figure 22 is a schematic presentation of the full-load roller gear and bevel assembly test stand.

INSTRUMENTATION SYSTEM

The instrumentation system used for testing was designed to monitor temperature, pressure, oil flow, speed, torque, vibration, chip detectors for the test transmissions, and test stand rotating equipment. This instrumentation consisted of:

1. Temperature recording equipment, including visual indication and a strip chart recorder.

2. Temperature monitor to scan temperatures.

3. Flowmeters, sensors, and visual indicators for display of oil signals.

4. Pressure gages for display of lubrication oil pressures.

5. Tachometer generator and indicator for recording speed.

6. Electric chip detectors to provide a visual signal of foreign material in the test boxes.

7. An electrical dynamometer to provide a torque input reading.

8. Vibration meters (horizontal and vertical).

A complete list of the instrumentation used is included in Table III.

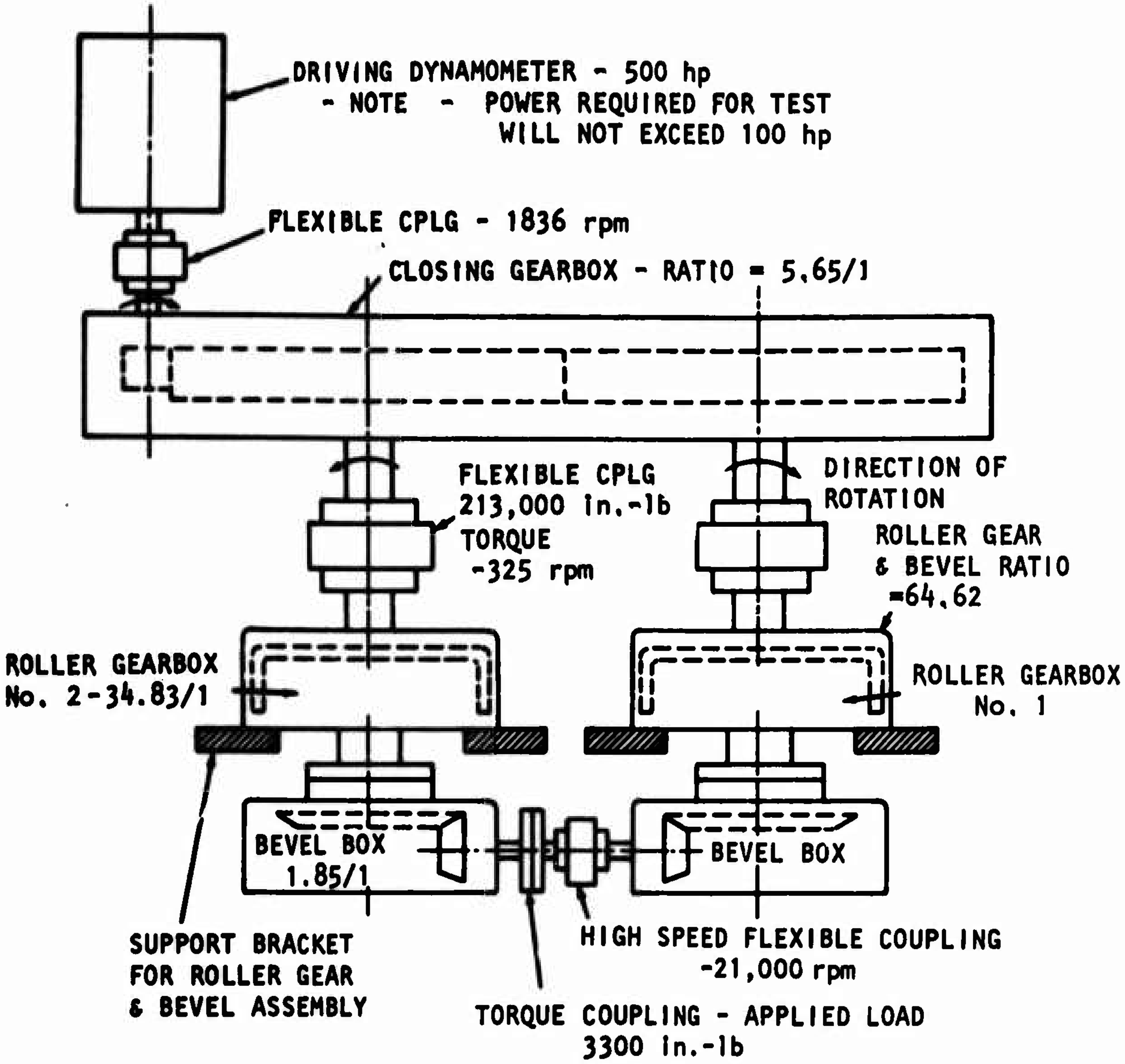


Figure 22. Test Rig Schematic.

TABLE III

INSTRUMENTS USED IN TESTING ROLLER DRIVES

Instrument	Range	Accuracy Full Scale	Frequency of Calibration
Oil Flow Readout	0-100	1/2%	1 Month
Oil Flow Readout	0-100	1/2%	1 Month
Temperature Readout	-75 to +575°F	1/4%	1 Month
Temperature Recorder	-100 - +500	1/4%	1 Month
Speed	0-1MC	±1 Count	1 Month
Torque	0-600 ft.-lbs	±1/2%	1 Month
Dyno	600 hp	-	-
Flow Transducers	1.25-9.5 gpm	1/2%	1 Month
Flow Transducers	0.6-4.0 gpm	1/2%	1 Month
Flow Transducers	1.25-10 gpm	1/2%	1 Month
Flow Transducers	0.3-1.63 gpm	1/2%	1 Month
Flow Transducer	1.2-10 gpm	1/2%	1 Month
Flow Transducer	1.1-3.87 gpm	1/2%	1 Month
Flow Transducer	1.5-9.3 gpm	1/2%	1 Month
Flow Transducer	0.3-1.63 gpm	1/2%	1 Month
Vibration Readout	0-1000-6 Ranges	5%	1 Month
Vibration Readout	0-1000-6 Ranges	5%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Vibration Pickup	0.1 to 1000 G's	±2%	1 Month
Oil Pressure Readout	0-60 psig	2%	Monthly
Oil Pressure Readout	0-600 psig	2%	Monthly
Oil Pressure Readout	0-100 psig	2%	Monthly
Oil Pressure Readout	0-100 psig	2%	Monthly

TEST RIG CALIBRATION

Calibrations of instruments required for testing were performed from one to five days prior to actual test using TRW master calibration equipment and applicable procedures. All calibration equipment is traceable to the National Bureau of Standards.

TEST SEQUENCE AND RESULTS

DEVELOPMENT TEST

Bevel Box Testing

Testing was performed on each of the two bevel boxes. Each box was checked for tooth contact pattern under no-load and full-load conditions. The contact pattern was satisfactory, and the units were subjected to no-load, full-speed testing.

The speed tests were stopped due to excessive roller bearing temperatures on both the spiral bevel pinion and the spiral bevel gear. The difficulty was traced to insufficient diametral looseness in the bearings. This problem was corrected; testing continued and was successfully completed.

Roller Gear Partial-Load Test

Two roller gear drive units were tested in a low-power back-to-back rig for ten hours with a preload torque equivalent to 165 hp at full design speed. The roller gear drive efficiency at these test conditions was calculated to be 98 percent. Subsequent full-power efficiencies were calculated to be 98.8 percent based on measured heat losses.

At completion of the test, one box was completely disassembled and the other was partially torn down. Examination of the units revealed the following items of distress:

1. Magnetic metal particles appeared in the lube oil.

2. There was evidence of what appeared to be slight brinelling of the eccentric pins, caused by the load reaction of the needle bearings.

3. A roller in one of the first-row roller gears had started to fail and pieces had actually broken off and had gone through some of the meshes; they were detected in the lubricating oil of the particular box. The failure was traced to defective rollers which contained cracks caused by faulty grinding operations.

4. An abnormal needle bearing eccentric pin load bearing pattern was noted and attributed to poor lubrication. This problem was eliminated in future tests.

5. All of the small rollers of the first row gears in both boxes showed damage which was confined exclusively to the sharp side edges of the rollers. This problem was attributed to insufficient corner breaks.

6. The most disconcerting area of distress was observed on the sun gear. This gear showed a very distinct wear contact pattern on both flanks of the pinion teeth, suggesting tight mesh conditions with no backlash.

Careful examination of the mating first-row gears showed slightly heavier than normal tip wear on two of these gears. Flank wear was clearly evident on the drive side of the teeth but was very hard to detect on the nondrive side. The only logical explanation for the near absence of wear on both sides of these first-row gears is that there are approximately 18 times as many load cycles on the sun gear; therefore, this behavior is strictly a time function. Involute profile measurements were made on all these first-row gears. The results indicated that the wear could not be measured and that the gears were in excellent condition. The tip wear in no case exceeded .0005 in. and did not extend along the tooth flank. This little wear is within the profile tolerance originally suggested for the manufacture of these first-row gears.

The first mesh has a design backlash of .006 to .008 in., the second mesh has .004 to .006 in., and the last mesh with the ring gear has .004 to .008 in. The ring gear mesh was normal, with tooth contact only on the driven side of the teeth.

The only reasonable explanation to account for the absence of backlash on the first mesh was that this is the only mesh in the entire system which did not contain rollers to insure proper center distance between the meshing gears. It was expected that rollers would not be necessary for the mesh; therefore, these rollers were intentionally omitted. All evidence indicated that the eccentric roller loading arrangement functioned adequately, as witnessed by the uniformity of load marking on the reaction eccentric pins, the uniformity of all other roller gear tooth markings, and the fact that no jamming ever occurred which could be detected by the test instrumentation used.

It was then presumed that a slight angular error could occur for the spacing of the first-row roller gears as a function of the automatic preload device. The error, which was significant, did not interfere with the function of the other meshes, which had even less backlash than the first sun gear mesh, but they had rollers to maintain a control of the meshing center distances. The slight tip wear of the first-row gear could also have been caused by the angular mislocation of the first-row gears and the absence of rollers on this mesh.

A static load rig was used to observe and to confirm the explanations advanced for the areas of distress. This rig permitted static loads to be placed on the roller gear drive up to a maximum of full torque. The construction permitted roller gear position measurements to be taken. As a result of this evaluation, the first modification to the design was made and rollers were provided on the sun gear and three on the first-row gears in each box.

Static Load Test for Roller Gear Drive

For the static test, one complete roller gear drive along with the right-angle drive was used. A special support ring was used to clamp the normally rotating output ring gear to the roller gear housing. A manually operated worm gearbox was attached to the input bevel pinion to apply input torque load. The worm gearbox was calibrated to determine the resulting loading torque on the stationary reaction ring gear. This rig had torque loading capabilities to 100 percent of full output torque. Static torque loads were applied at 15, 50, and 100 percent.

Dimensional readings for the center of each six first-row gears (Figures 4 and 5) from the input sun gear (Figure 3) were recorded along with the chordal dimensions of these gears. Pins were inserted into the six housing shafts, and chordal dimensions were recorded center to center of these shafts. Depth micrometer readings were taken from the special support ring to the outer diameter of the ring gear at twelve equally spaced places before and after torque loading. In addition, visual observations were made on the functioning of the eccentrics and the resulting backlash of the meshing gears.

From these measurements, it was concluded that the eccentrics were not working in unison, causing the roller gears to be unequally loaded and improperly positioned and thus, resulting in the absence of backlash in some of the meshes. A plot of the ring gear deflection indicated that the ring gear deflection was two-mode instead of six-mode. A six-mode deflection curve should exist in an equally loaded six outer planet roller gear system.

Because of the observations, a synchronizer system of linkages was built which adapted to the existing hardware without any significant change. This synchronizer linkage permitted the adjustment of individual eccentrics under load and locked them into position. A test rig that permitted observation of the linkage effectiveness and permitted the same type of measurement data to be taken was designed and built. This rig differed from the previous rig because loads could be applied, and the gears were made to rotate at low speed.

Roller Gear Torque Load Test

Two complete gear drives along with their bevel reductions were loaded at 5, 10, 15, 25, 50, and 75 percent torque load levels. The torques were applied by means of a specially furnished torque load coupling on the high-speed input bevel shaft which connected to form a four-square loop. Power was applied to the output ring gear of one roller gear drive and rotated at approximately 1 rpm. Measurements were taken for the radial positions and spacing of all first-row gears before and after rotation. The sun gear and all first-row gears were painted with red lead marking compound, and the meshing patterns were carefully examined.

It was concluded that the synchronizing linkage was not necessary to insure proper functioning of the roller gear drive, and it was not employed in the test. The positioning of the roller gear elements, which resulted from eccentrics located in the outer row roller gears, appeared to be very satisfactory, and good load sharing appeared to be evident when the meshing patterns were studied.

A 10-hour, full-speed, back-to-back 15 percent load test of the roller gear drive was completed. All test data indicated a very satisfactory test. (It should be noted that the preceding two paragraphs show remarkable difference between static and dynamic load modes. It seemed that variation in static friction among the contacted parts introduced distortions which were eliminated when the rotation started.)

TEST OBJECTIVE

The primary test objective was to run a 200-hour dynamic load test on each of two roller gear test units, at 100 percent design speed and 100-percent design load. During the entire 200-hour duration, spectrographic analysis of the gear drive lubrication oil was to be made to determine the iron content to detect any gradual wear taking place in the roller gear drives. A minimum of 20 oil samples were to be taken for this purpose at time intervals not to exceed 10 hours. After the 200-hour test, the roller gear drive and bevel boxes were to be dismantled for a complete inspection and evaluation.

Full-Load Test

A full-load, full-speed test was conducted for 1 hour prior to initiation of the 200-hour endurance test. The hardware was inspected and appeared to be satisfactory for additional testing.

There was some distress evident on the 1.800-in.-diameter rollers on the first-row roller gears. A 0.030-in. x 20° chamfer was machined on these parts to provide additional relief before the test was continued. In addition, thermocouples were incorporated into the bevel boxes, and an improvement of instrumentation reliability was incorporated throughout the test rig.

Endurance Test

Full-load testing was terminated when a chip was detected in the right-hand unit. The unit was shut down after accumulating approximately 24.5 hours. Spectrographic examination of the chip indicated that it could be roller or gear material. Chemical lab analysis of the oil was made every 10 hours, but there had been no significant rise in the elements prior to shutdown.

The right-hand gearbox was completely disassembled. A piece of material approximately 3/8 in. x 1/4 in. had spalled away from a first-row roller gear at the 1.800-in.-diameter roller surface. The remainder of the parts were Magnaglo inspected, and a crack was discovered in another first-row roller gear. This crack extended from the side of the roller onto the load-carrying roller surface.

A design study was made to confirm what appeared to be an example of case crushing failures on the spalled first-row gear roller. The load on the roller surface was 2750 lb, and it had an effective length of 0.178 in. The calculated Hertz stress was 237,000 psi and was normally considered safe for the application. The contact deflection was calculated to be 0.0015 in., and the depth to the point of maximum shear stress was calculated to be 0.015 in. The drawing requirement called for a 0.020- to 0.030-in. effective case depth. The design study was followed by a complete analytical and metallurgical investigation, which concluded that the specified case depth of 0.020 to 0.030 in. was inadequate.

The second unit was also disassembled and completely inspected. The outer planetary gears had indications of surface peeling on the rollers. Chemical inspection revealed that a chrome plating on the roller surface was breaking down. Magnaglo inspection of all gears revealed no additional flaws.

From the above analysis, a second design modification was made to incorporate roller contacts at all gear meshes in contact with the sun gear and to increase the case thickness on the first-row roller gears.

Testing of the redesigned unit was initiated after alignment checks had been made on the high-speed coupling and careful balancing (.015 oz-in. or less) of the torque coupling had been completed. The applied load at the 21,000-rpm coupling was 275 ft-lb (100 percent load).

The first run was made at 40 percent speed, and a complete set of readings was recorded. Both boxes were listened to, and no extraneous noise was detected. All recorded temperatures were satisfactory for the continuation of testing. At the end of approximately 15 minutes of running at 40 percent speed, the drives were brought to 60 percent speed and a complete set of readings was recorded. The drives were kept at this speed for about 15 minutes. Since everything looked and sounded satisfactory, the units were brought to 100 percent speed.

At 100 percent speed, satisfactory flows and temperatures were recorded. The vibration levels were observed, and the right-hand horizontal vibration pickup read 60 G's. Inside the cell, the right-hand box was listened to and sounded definitely rough. A shutdown was immediately ordered. The last set of oil pressure readings indicated a low oil pressure condition in the right-hand unit. Investigation revealed that one of the stainless steel oil tubes had broken off from the main oil gallery. A complete roller gear drive disassembly was necessary.

Inspection of the roller gear drive components revealed:

1. The oil tube had gone through the gear meshes. The probable cause of failure was overheating of the 304 stainless steel during fabrication and embrittling of the 304 by the copper braze. Subsequently, the part failed because of vibration levels during testing.

2. A piece of shiny magnetic material was found in the drive and traced to the sun gear flange.

3. One second-row roller gear had a tooth slightly bent.

4. There were numerous abrasions of varying intensity which were not considered serious.

5. Magnaglo inspection of all gears revealed small indications on the load-carrying roller of a second-row roller gear. Analysis indicated that grind checks existed on the nonload side of the roller and were propagated onto the roller surface after running under load. Magnaglo inspection which was used to establish the soundness of the part did not detect the grind checks. The mechanical construction of this particular roller and gear (Figure 9) prohibited the use of a nital-etch inspection which would have detected this problem area.

6. Magnaglo inspection revealed small indications at the machining centers on a first-row roller gear. Microscopic examination traced the indication to foreign material inclusions. The part was considered satisfactory for future use.

7. Most of the rollers in the first-row gears had some circumferential movement. Previous experience with this type of problem (reference Report AD 471-437) indicated that it did not cause any detrimental affect, and it was decided to continue the use of these first-row components for future tests.

8. A first-row gear had what appeared to be chatter marks on a small roller. Roundness was checked, and the maximum out-of-roundness indications were .000020. The part was considered to be satisfactory for future testing.

The right bevel box was checked; the bevel gearing and bearings were in excellent condition. The high-speed coupling was checked, and the only evident area of distress was on the input spline hub teeth to the left-hand bevel box. The distress was very minor, and a total of 30 minutes running time was too short to adequately evaluate the high-speed coupling.

The test was restarted after careful reassembly of roller gearbox No. 2. Full-load torque (1100 hp) was applied to the boxes after pre-established procedures had been carried out for the high-speed coupling shaft.

Test runs were made at various speed levels at full-load torque. Several times during this period the chip detector light for the right box (No. 2) came on. Examination by the Spectrographic Lab indicated materials that were neither roller nor gear material, so testing was permitted to continue. Eleven hours at full load and full speed were accumulated during this period.

Testing was suspended when spectrographic analysis of a chip picked up by a detector on the right roller box (No. 2) indicated the material to be 52100 steel which was used for some of the rollers. Disassembly of this box revealed that a roller on the sun gear nearest the drive coupling had a flange completely disintegrated. The remaining three rollers on this same part were not damaged.

All roller gears which had rollers contacting the broken flange showed abrasion. The damage was not considered to be excessive, and the rollers were chamfered to remove most of the distressed areas. Some of the metal from the sun roller flange went through the meshes, and a second-row roller gear had a damaged tooth. For the rebuild, this gear was replaced. There were numerous abrasion marks on the other gears, but these were almost completely removed by careful stoning.

All of the roller gears, including the ring gear which had the most gear teeth damaged, were Magnaglo inspected and were found to be free of any indication of cracks.

At this time the box failure was attributed to the presence of an unexpected axial force which reacted on the roller flanges and broke out the rather thin sun roller flange. The flanges of the sun roller were made only .065-.070 in. thick because of space limitations brought about in this design, which was considerably compromised to incorporate roller contacts on all gear contacts. The stiffness of this system is considerably different from the original system, which did not employ roller contacts of the sun roller, and this may have added significantly higher thrust forces to the roller gear elements.

In the preceding test run, the sun gear had also broken the end roller flange nearest the drive coupling. This failure had previously been explained by the lube line failure. After stripping this damaged roller from the sun gear, a roller from a spare sun gear was used as a replacement. After this roller was measured and shimmed, the sun gear for proper running clearances, it was used for the rebuild.

To make up for the lack of suspected flange strength on this sun roller gear, a larger diameter was incorporated in the drive spline and on the outboard washer to provide a strong backing for the flanges. There was no possible way to increase the strength of the two inner flanges, but previous running experience indicated that there was no problem with these flanges.

Box No. 2 was reassembled and testing was again initiated.

Full-torque load was applied, and the system was run for 1 hour at 54-percent speed. One set of readings was taken, and nothing unusual was observed on the instrumentation during this 1-hour run. The chip detector on the right roller gearbox (Serial No. 2) shorted twice, and because of the suspicious appearance of the metal chip, testing was suspended. Spectrographic analysis confirmed the presence of 52100 steel, and a teardown was ordered again for Box No. 2.

The only serious damage discovered in this teardown was a broken roller flange in the second-row roller gear on the side opposite the drive coupling. This roller flange was replaced, and the gear was considered to be satisfactory for further testing.

Box No. 2 was reassembled and testing was again initiated. Testing was suddenly suspended at the end of 52 minutes due to the failure of the high-speed flexible coupling, which resulted in considerable damage to the bevel boxes and minor damage to the roller gear boxes.

The failure of the high-speed shaft was attributed to two causes. The first cause was the lack of test rig stiffness, and the second cause was insufficient rigidity of the bevel boxes.

To correct these deficiencies, an improved stiffened test rig was erected and the product bevel gearboxes were supported by brackets attached directly to the wall of the roller gear housing which carried the six load bearing reaction pins. With this arrangement, the high-speed coupling was not affected by deflections resulting from the output torque loads.

The roller gear drives were in no way connected with this failure. The only apparent damage to these boxes was attributed to inadequate strength in the flanged rollers.

All rollers in the outer row roller gear assemblies were replaced. The flange thicknesses were increased. The roller gear drives and bevel boxes were repaired and reassembled for test.

Testing was again initiated and terminated after approximately 6 hours of running. Two interruptions during the 6-hour test were the result of a high-stress condition at the high-speed coupling preload disk and dynamometer stator trunnion bearing problems. Testing was discontinued as a result of a metal chip which was discovered in the roller gear drive lubrication fluid. The analysis showed that the chip was roller material. Disassembly of the unit indicated that it had come from the sun gear. All other damage was minor in nature and easily repaired.

The roller gear drives were repaired and testing was again initiated. For a period of 2 hours and 43 minutes, the roller gear drives performed at full-load conditions. Testing was terminated at this time because of an unusual noise. The metal chip detector light for the south gearbox indicated that metal was present in the oil return line. The south gearbox was removed for disassembly. Rotation of the south gearbox was not possible. Disassembly of the unit revealed that three rollers were completely disintegrated. Both large and small segments of the rollers were scattered throughout the box along with fine particles of metal.

Although the primary cause of failure was not specifically identified, the evidence indicated abnormal loading of the roller rings. Several potential causes of the abnormal loading exist, the most probable of which is the rotation of the roller ring during operation. Furthermore, it is apparent that the roller ring and gear assembly must be redesigned and a better method devised to retain the ring. At this point, testing was discontinued.

<u>Efficiency Data</u>

The example used is based on full-speed and full-load conditions:

Applied load = 275 ft-lb = 3300 in.-lb

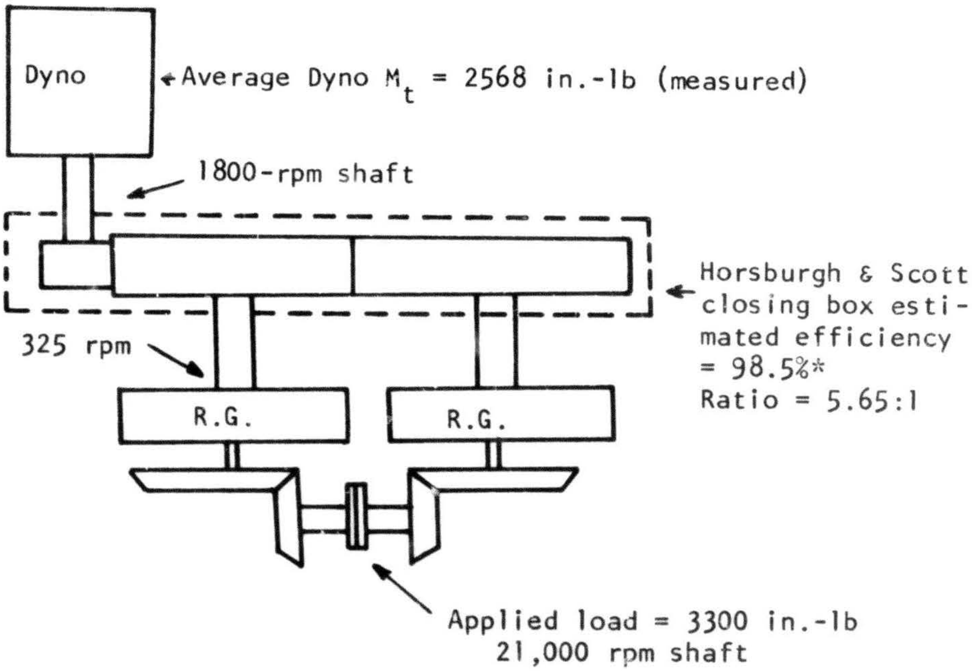
Load is applied at the 21,000-rpm shaft.

$$M_t = \frac{63{,}000 \times hp}{N} \; ; \quad hp = \frac{21{,}000 \times 3{,}300}{63{,}000} = 1100 \text{ hp}$$

Average dyno torque (based on 25 readings) = 214 ft-lb = 2568 in.-lb

Dyno speed = 1,836 rpm

$$\text{Input hp} = \frac{N \times M_t}{63{,}000} = \frac{1{,}836 \times 2{,}568}{63{,}000} = 74.84 \text{ hp}$$



$\therefore$ Loss in commercial box = .015 x 1100 = 16.5 hp

Total measured loss for the two roller gear drives and two bevel gear drives = 74.84 - 16.5 = 58.34 hp.

It is reasonable to assume that the losses of the roller gear drives and their bevel boxes will be proportional to their measured heat losses (these losses are based on 25 readings).

Roller Gear Drive

Average oil in = 160°F

Average oil out = 194°F

ΔT = 34°F

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*Horsburgh & Scott calculated efficiency between speeds of 300 and 1,800 rpm is 98.5 percent.

Average flow rate = 3.75 gpm/box

1 gal. turbo #35 oil wt. = 7.7 lb/gal

S.G. = .924

Coefficient of spec. heat C_p = .49

$Q = MC_p \, \Delta T$

$Q = 3.75 \times 7.7 \times .49 \times 34 = 481$ Btu/min/box

$Q = \frac{481}{42.4} = 11.34$ hp/box

Bevel Gear Drive

Average mesh flow	=	1.28 gpm
Average cartridge flow	=	1.10 gpm
Average mix flow	=	.90 gpm
Total average flow	=	3.28 gpm
Average oil in	=	160°F
Average oil out	=	207°F

$\Delta T = 47^\circ F$

$Q = MC_p \, \Delta T$

$Q = 3.28 \times 7.7 \times .49 \times 47 = 581.62$ Btu/min/box

$Q = \frac{581.62}{42.4} = 13.72$ hp

Let total roller gear loss = A
Total bevel gear loss = B

$$A + B = 58.34$$

$$\frac{A}{B} = \frac{11.34}{13.72}; \quad A = \frac{11.34}{13.72} B = .8265B$$

$$1.8265\,B = 58.34$$
$$B = 31.94$$
$$A = 26.40$$

$\therefore$ Total loss for roller gearbox $= \dfrac{26.40}{2} = 13.20$

$$\% \text{ loss} = \frac{13.20}{1{,}100} \times 100 = 1.2\%$$

$\therefore$ Roller gear drive efficiency is $= 98.8\%$

Measured heat loss $= 11.34$ hp

$$\% = \frac{11.34}{13.20} = 85.9\%$$

Total loss for the bevel box $= \dfrac{31.94}{2} = 15.97$

$$\% \text{ loss} = \frac{15.97}{1{,}100} \times 100 = 1.45\%$$

Bevel drive efficiency is $= 98.55\%$

Measured heat loss $= 85.9\%$

The measured results indicate good correlation.

CONCLUSIONS

The roller gear drive was subjected to a total operating test time of 76.5 hours, of which 44.5 hours were at 100-percent design speeds and 100-percent design load. A total of 19.0 hours at full-load condition was accumulated on the final configuration. The test objective of 200 hours was not achieved.

The testing system employed a back-to-back coupling (four-square loop) of two test units in a closed-power path with an input shaft speed of 21,000 rpm and an output shaft speed of 325 rpm. An instrumentation system measured loads, speed, oil flows, pressures, temperatures, and vibration levels. During the 200-hour load tests, the lubrication oil was analyzed frequently to determine the iron content in parts per million.

The test program encountered difficulties due to test rig malfunction and minor design deficiencies which became apparent during the full-load portion of the test program. These difficulties resulted in a number of test interruptions and two design modifications. The design modifications were made between the first-row planet gears and the sun gear, wherein the roller contact was increased from zero roller in design A to 100 percent roller contact in design C.

The following hours of running time were logged for each design:

Design	Partial Load and Speed	Full Load Full Speed
A	10.0 Hours	-
B	10.0 Hours	25.5 Hours
C	12.0 Hours	19.0 Hours

LOAD SHARING

An open-face test proved that all gear contacts had good load sharing from 10 to 100 percent of design load.

EFFICIENCY

Drive efficiency calculated by torque balances or by heat balances showed approximately 98.8 percent efficiency or 1.2 percent loss. Other comparable ratio drives at similar speed and load ranges have at least 2.75 to 3.0 percent loss.

DRIVE FAILURES EXPERIENCED

Except for the first failure (caused by insufficient depth-case hardening of the rollers), other failures were caused by shoulder failures resulting from generally weak shoulder design. At the time that the drive was designed, the value of the thrust forces on "ideal" rolling surfaces was not known. These forces were experimentally evaluated as a percentage of radial load forces only. A more conservative design approach would be to increase the roller width to reduce the contact stresses.

The last failure (the exact cause of which cannot be determined) showed that the roller gear fastening methods are not 100 percent reliable and must be improved. The roller width should be extended radially inward below the gear line, or the roller and gear body contact should be electron beam welded.

The detailed failure analyses performed in this task indicate that the basic design concept of the roller gear drive is sound and offers a practical solution for long-life, high-speed, high-power, and high-ratio aerospace transmissions with excellent efficiency.

APPENDIX

LOAD STRESS CALCULATIONS, ROLLER GEAR DRIVE

1.0 Basic drive requirements - based on full speed and load conditions:

Input speed = 21,000 rpm

Input horsepower = 1100

Output speed = 325 rpm

Overall ratio $= \frac{21,000}{325} = 64.61$

Drive ratio $= \frac{X_1}{A} \times \frac{C}{Y_1} = \frac{66}{24} \times \frac{228}{18} = 2.75 \times 12.667$

$= 34.83$

Bevel gear ratio $= \frac{63}{34} = 1.853$

Input speed $= \frac{21,000}{1.853} = 11,333$ rpm

Sun gear input torque $= \frac{63,000 \times 1100}{11,333} = 6114.9$ in.-lb

First-row gear torque $= 6114.9 \times 2.75 = 16,816$ in.-lb

Second-row gear torque $= 16,816 \times \frac{72}{18} = 67,264$ in.-lb

Output torque $= 67,264 \times \frac{228}{72} = 213,000$ in.-lb

W_T (Sun) $= \frac{6114.9}{1.200 \times 6} = 849.3$ lb (There are 6 tooth contacts.)

W_T (Y_1 contact) $= \frac{16,816}{.900 \times 12} = 1557$ lb (There are 12 tooth contacts.)

W_T (ring contact) $= \frac{67,264}{3.600 \times 6} = 3114$ lb (There are 6 tooth contacts.)

2.0 Gear stresses - first mesh, A and X_1 gears:

$$S_T = \frac{W_T P_d K_e}{F\left(\frac{Y}{K_f}\right) C_V}$$ = bending stress (based on Lewis formula)

Sun input torque = 6115 in.-lb

W_T = 849.3 lb

P_d = diametral pitch = 10

K_e = load distribution factor = 1.2

F = face width = .872

$\frac{Y}{K_f}$ = .35

C_V = 1.0

$$S_T = \frac{849.3 \times 10 \times 1.2}{.872 \times .35 \times 1.0} = 33{,}394 \text{ psi (tooth bending)}$$

$$\text{Hertz stress} = S_c = 5235 \sqrt{\frac{W_T\ K_e\ (M_g + 1)}{F \times d \times C_v\ M_g}}$$

$$M_g = \text{gear ratio} = \frac{66}{24} = 2.75$$

$$\frac{M_g + 1}{M_g} = \frac{2.75 + 1}{2.75} = 1.36$$

$$S_c = 5235\sqrt{\frac{K \times K_e}{C_v}};$$

C_v = 1.0

d = pitch diameter = 2.40

K_e = 1.2

$$K = \frac{W_T}{F \times d} \times \frac{M_g + 1}{M_g} = \frac{849.3 \times 1.36}{.872 \times 2.40} = 552$$

$$S_c = 5235 \sqrt{\frac{552 \times 1.2}{1.0}} = 5235 \sqrt{662.4}$$

S_c = 134,749 psi

3.0 Gear stresses - second mesh, Y_1 and X_2 gears:

W_T = 1557 lb

P_d = 10

K_e = 1.2

$$F = 2.0$$

$$\frac{Y}{K_f} = .34$$

$$C_V = 1.0$$

$$S_T = \frac{1557 \times 10 \times 1.2}{2.0 \times .34 \times 1.0} = 27{,}500 \text{ psi (tooth bending)}$$

$$M_g = \frac{72}{18} = 4; \ \frac{M_g + 1}{M_g} = \frac{5}{4} = 1.25$$

$$K = \frac{W_T}{Fxd} \times \frac{M_g + 1}{M_g}; \ \begin{array}{l} F = 2.0 \\ d = 1.80 \end{array}$$

$$K = \frac{1557 \times 1.25}{2.0 \times 1.80} = 541$$

$$S_c = 5235 \sqrt{541 \times 1.2} = 5235 \sqrt{649}$$

$$S_c = 133{,}335 \text{ psi (Hertz stress)}$$

4.0 Gear stresses - third mesh, X_2 and C gears:

$$W_T = 3117 \text{ lb}$$

$$P_d = 10$$

$$K_e = 1.2$$

$$F = 2.0$$

$$\frac{Y}{K_f} = .385$$

$$C_V = 1.0$$

$$S_T = \frac{3117 \times 10 \times 1.2}{2.0 \times .385 \times 1.0} = 48{,}500 \text{ psi (tooth bending) } X_2 \text{ gear}$$

$$M_g = \frac{228}{72} = 3.167$$

$$\frac{M_g - 1}{M_g} = \frac{3.167 - 1}{3.167} = \frac{2.167}{3.167} = .684$$

$$K = \frac{W_T}{Fxd} \times \frac{M_g - 1}{M_g}; \ \begin{array}{l} F = 2.0 \\ d = 7.20 \end{array}$$

$$K = \frac{3117 \times .684}{2.0 \times 7.20} = 148$$

$$S_c = 5235 \sqrt{148 \times 1.2} = 5235 \sqrt{178}$$

S_c = 70,000 psi (Hertz stress) X_2 and C gears

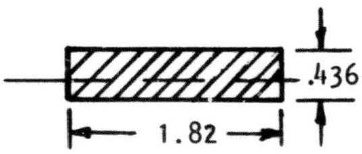
5.0 Bending stress - C ring gear:

$$\begin{array}{lr} OD = & 23.930 \\ ID = & \underline{23.058} \\ & .872 \end{array}$$

$$\text{Wall thickness} = \frac{.872}{2} = .436$$

$$\text{Inside radius} = \frac{23.058}{2} = 11.529$$

$$\begin{array}{lr} \text{1/2 wall thickness} = & .218 \\ \text{Mean radius 'R'} = & \overline{11.747} \end{array}$$



$$I = \frac{bh^3}{12} = \frac{1.82 \times .436^3}{12} = .0126$$

W_T (ring gear) = 3117 lb

Separating force $W_S = 3117 \text{ Tan } 25^\circ = 3117\ (.46631) = 1453$ lb

Moment at load = KWR = .089 x 1453 x 11.747 = 1519 in.-lb

$$S = \frac{Mc}{I} = \frac{1519 \times .218}{.0126} = 26{,}300 \text{ psi (ring gear bending)}$$

6.0 Hertz stress on A-X_1 roller contact:

Load = 1720 lb

Dia (D_1) of A = 2.400

Dia (D_2) of X_1 = 6.600

Effective roller face width = .130 each roller

.130 x 2 = .260 total width

$$\text{Hertz stress} = S_c = .591 \sqrt{\frac{P}{t} \times E \times \frac{D_1 + D_2}{D_1\ D_2}}$$

$$S_c = .591 \sqrt{\frac{1720}{.260} \times 30 \times 10^6 \times \frac{2.4 + 6.6}{2.4 \times 6.6}}$$

$$S_c = .591 \sqrt{6615 \times 30 \times \frac{9.0}{15.84}}$$

$$S_c = .591 \sqrt{112,755} = 591 \times 336$$

$$S_c = 198.576 \text{ psi}$$

7.0 Hertz stress on Y_1-X_2 roller contact:

Load = 2750 lb

Dia (D_1) of Y_1 = 1.800

Dia (D_2) of X_2 = 7.200

Effective roller face width = .184 each roller

.184 x 2 = .368 total width

$$\text{Hertz stress} = S_c = .591 \sqrt{\frac{P}{t} \times E \times \frac{D_1 + D_2}{D_1 \; D_2}}$$

$$S_c = .591 \sqrt{\frac{2750}{.368} \times 30 \times 10^6 \times \frac{1.8 + 7.2}{1.8 \times 7.2}}$$

$$S_c = .591 \sqrt{7473 \times 30 \times \frac{9.0}{12.96}}$$

$$S_c = .591 \sqrt{155,687} = 591 \times 394.6$$

$$S_c = 233,209 \text{ psi}$$

8.0 Gear stresses - high-speed bevel gear:

Engine = 1100 hp at 21,000 rpm

$$M_T = \frac{63,000 \times 1100}{21,000} = 3300 \text{ in.-lb}$$

Pinion P.D. = 4.25 R = 2.125

$$W_T = \frac{3300}{2.125} = 1553 \text{ lb}$$

$$S_T = \frac{W_T \times K_o \times P_d \times K_s \times K_m}{K_v \times F \times J} = \text{bending stress (Gleason formula)}$$

W_T = 1553 lb

K_o = 1.25 (overload factor)

P_d = 8 (diametral pitch)

K_s = .595 (size factor)

K_m = 1.15 (both members straddle-mounted) - load distribution factor

K_v = 1.0 (dynamic factor)

F = 1.125 (face width)

J = .354

$$S_T = \frac{1553 \times 1.25 \times 8 \times .595 \times 1.15}{1.0 \times 1.125 \times .354} = 26{,}800 \text{ psi}$$

$$\text{Hertz stress} = S_c = C_p \sqrt{\frac{W_T \times C_o}{C_V} \times \frac{1}{Fxd} \times \frac{C_s C_m C_f}{I}}$$

C_p = 2800

W_T = 1553 lb

C_V = 1.25 (overload factor)

C_s = 1.0 (size factor)

C_m = 1.15 (load distribution factor) both members straddle-mounted

C_f = 1.0 (surface condition factor)

F = 1.125 (face width)

d = 4.25 (pinion P.D.)

I = .118 (geometry factor)

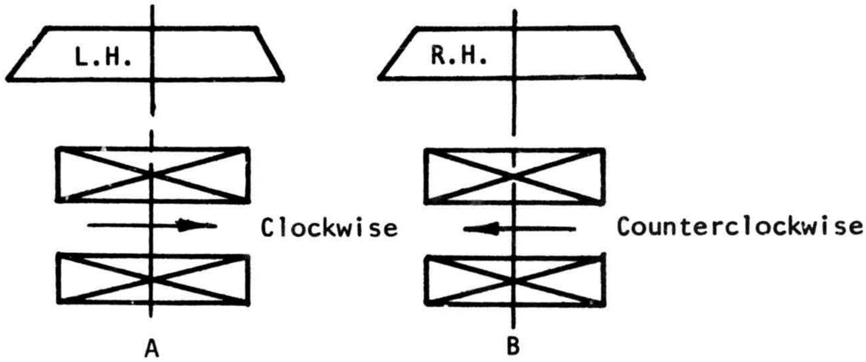
$$S_c = 2800 \sqrt{\frac{1553 \times 1.25}{1.0} \times \frac{1}{1.125 \times 4.25} \times \frac{1.0 \times 1.15 \times 1.0}{.118}}$$

$$S_c = 2800 \sqrt{\frac{1553 \times 1.25 \times 1.15}{1.125 \times 4.25 \times .118}}$$

$$S_c = 2800 \sqrt{3960} = 176{,}000 \text{ psi}$$

9.0 High-speed bevel pinion bearing loads:

$$F = \frac{\text{Input Torque}}{\text{MPR (pinion)}} = \frac{3300}{1.858} = 1776 \text{ lb}$$



γ_p = arctan $\frac{34}{63}$ = arctan .53968

γ_p = 28.35^o (approx.) (1/2 cone angle, pinion)

γ_g = 90-28.47^o = 61.65^o (1/2 cone angle, gear)

$$\gamma_p = F\left(\tan B \cos \gamma_p + \frac{\tan a \sin \gamma_p}{\cos B}\right.$$

tan B = tan 30^o = .57735

cos γ_p = cos 28.35^o = .88006

tan A = tan 20^o = .36397

sin γ_p = sin 28.35^o = .47485

cos B = cos 30^o = .86603

$$T_p = 1776\ (.57735 \times .88006 + \frac{.36397 \times .47485}{.86603})$$

$$T_p = 1776\ (.50810 + \frac{.17283115}{.86603})$$

T_p = 1776 (.50810 + .19957)

T_p = 1776 (.70767) = 1257 lb

MPR_p = 1/2 (Pinion PD - FW sin γ_p)

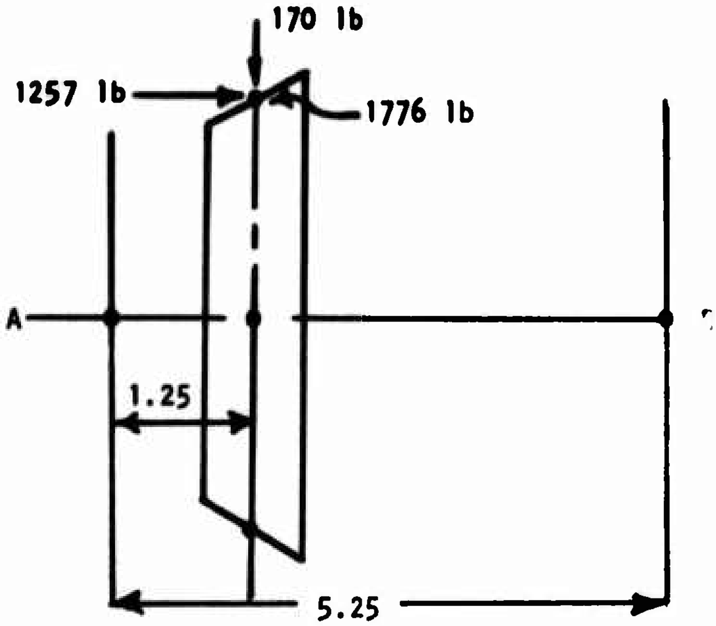
= 1/2 (4.25 - 1.125 x .47485) = 1/2 (4.25 - .53421)

= 1/2 (3.7158) = 1.858 (approx.)

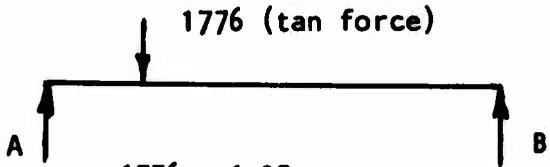
$$T_G = F\ (-\tan B \sin \gamma_p + \frac{\tan A \cos \gamma_p}{\cos B})$$

$$T_G = 1776\ (-.57735 \times .47485 + \frac{.36397 \times .88006}{.86603})$$

$$T_G = 1776\left(-.27415 + \frac{.320315}{.86603}\right)$$
$$= 1776\,(-.27415 + .36987)$$
$$= 1776\,(+.09572)$$
$$= 170 \text{ lb } (30^\circ \text{ spiral angle})$$

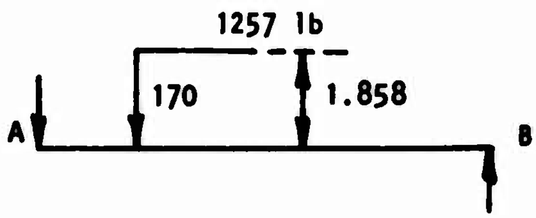


<u>Horizontal Plane</u>



$$B = \frac{1776 \times 1.25}{5.25} = 423 \text{ lb}$$
$$A = 1776 - 423 = 1353 \text{ lb}$$

<u>Vertical Plane</u>



$$\Sigma M_A = 0 \quad 1257 \times 1.858 + 170 \times 1.25 = 5.25B$$

$$2335.5 + 212.5 = 5.25B$$

$$B = \frac{2548}{5.25} = 485 \text{ lb}$$

$$\Sigma M_B = 0 \quad 1257 \times 1.858 - 170 \times 4.0 - A \times 5.25 =$$

$$2335.5 - 680 - 5.25A = 0$$

$$A = \frac{1655.5}{5.25} = 315 \text{ lb}$$

Resultant bearing force A $= \sqrt{1353^2 + 315^2} = 1390$ lb

Resultant bearing force B $= \sqrt{423^2 + 485^2} = 645$ lb

For bearing A,

Use MR-306-EX (ABEC 5) Roller Bearing

Rating = 4130 at 1000 rpm

Rating at speed = 4130 (.40) = 1653

Load = 1390

$$SF = \frac{1653}{1390} = 1.19$$

$$\text{Life} = 500\ (1.19)^{3.33} = 890 \text{ hours}$$

For bearing B (radial load),

Use MR-209-C (ABEC 5) roller bearing

Rating = 4855 at 1000 rpm

Rating at speed = 4855 (.40) = 1942

Load = 645

$$SF = \frac{1942}{645} = 3.0$$

$$\text{Life} = 500\ (3.0)^{3.33} = 18{,}000 \text{ hours}$$

For bearing B (thrust load),

Use matched set (2) 209 RDT and (1) 209RDB

Bearing rating = 2260 at 1000 rpm

Rating at speed = 2260 x 1.9 x .362 = 1560

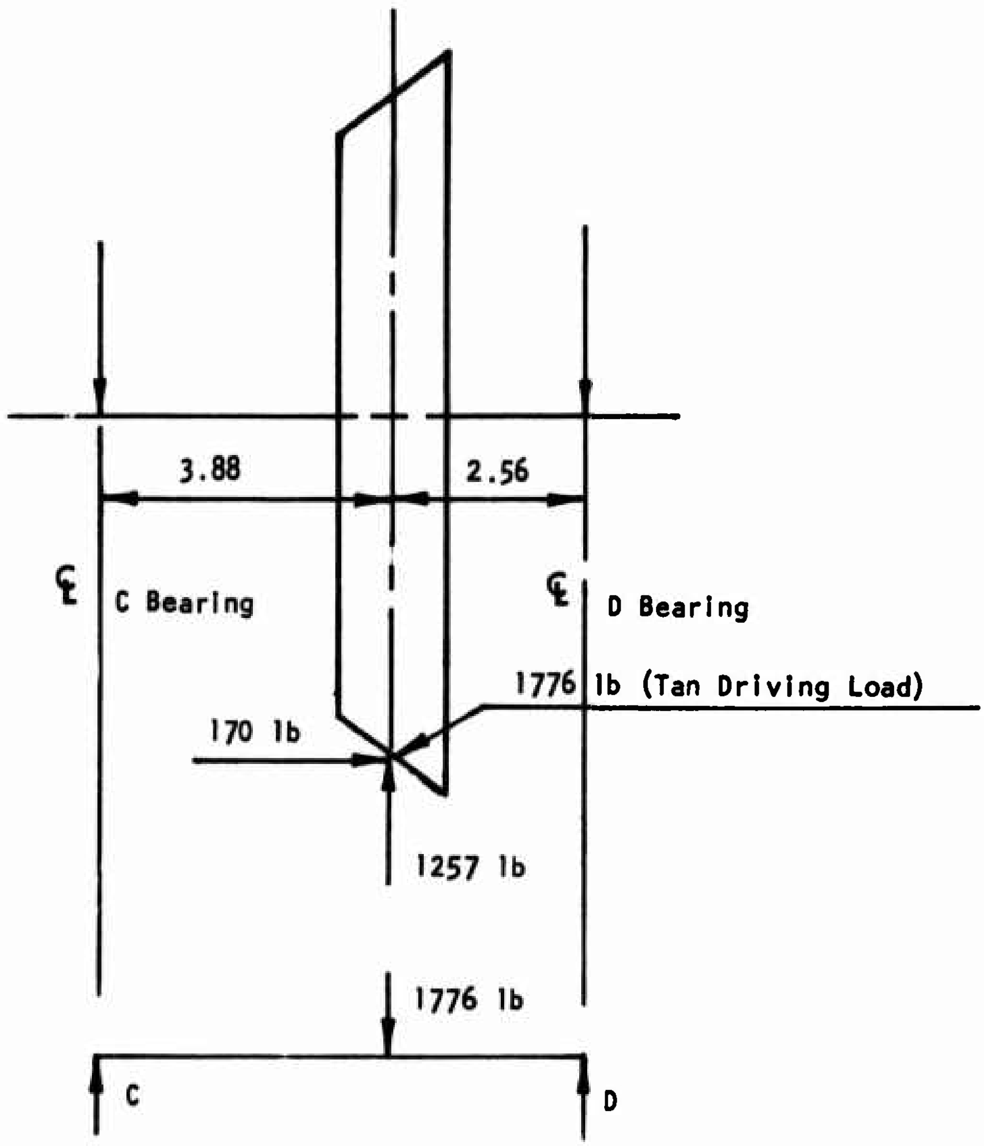
$$SF = \frac{1560}{1257} = 1.24$$

$$\text{Life} = 500\ (1.24)^{3.33} = 950 \text{ hours}$$

10.0 High-speed bevel gear bearing loads:

$$
\begin{aligned}
\text{MPR (gear)} &= 1/2\ (\text{gear PD} - \text{FW} \times \sin \gamma\ G) \\
&= 1/2\ (7.875 - 1.125 \times \sin 61.65) \\
&= 1/2\ (7.875 - 1.125 \times .88006) \\
&= 1/2\ (7.875 - .99006) = 1/2\ (6.885) = 3.443
\end{aligned}
$$

<u>Vertical Plane</u>



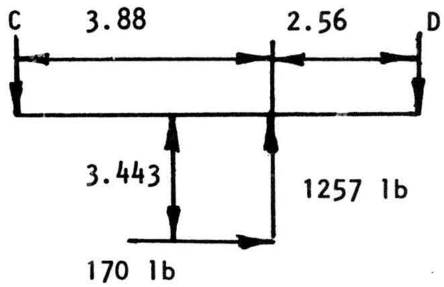
$\Sigma M_C = 0$

$$1776 \times 3.88 = 6.44D$$

$$D = \frac{6890.88}{6.44} = 1070 \text{ lb}$$

$$C = 1776 - 1070 = 706 \text{ lb}$$

<u>Horizontal Plane</u>



$\Sigma C = 0$

$$6.44D = 170\ (3.443) + 1257\ (3.88)$$

$$6.44D = 585.3 + 4877.1 = 5462.4$$

$$D = \frac{5462.4}{6.44} = 848 \text{ lb}$$

$\Sigma D = 0$

$$6.44C + 170\ (3.443) = 1257 \times 2.56$$

$$6.44C = 3217.9 - 585.3 = 2632.6$$

$$C = \frac{2632.6}{6.44} = 409 \text{ lb}$$

Resultant bearing force C = $\sqrt{706^2 + 409^2} = 815$ lb

Resultant bearing force D = $\sqrt{1070^2 + 848^2} = 1360$ lb

Rpm of gear = $\frac{21,000}{1.853} = 11,333$

For bearing D,

Selecting 211RDF (ABEC 3)

$$P = .62\ (1370) + .74\ Y \times 170$$

$$K = 4.43;\ \frac{T}{K_i} = \frac{170}{4.43} = 38;\ Y = 2.1$$

$$P = .62\ (1370) + .74 \times 2.1\ (170)$$
$$= 849 + 264 = 1113 \text{ lb}$$
$$SF = \frac{1360}{1113} = 1.22$$
$$\text{Life} = 500\ (1.22)^3 = 900 \text{ hours}$$

For bearing C,

$$\text{Speed factor} = \left(\frac{1000}{11,333}\right)^3 = (.0882)^{.3} = .484$$

Bearing load = 815 lb

Selecting MR-207 EX (ABEC 3)

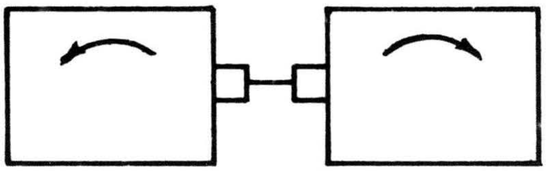
Bearing rating = 3490 lb at 1000 rpm

Rating at speed = 3490 (.484) = 1685 lb

$$SF = \frac{1685}{815} = 2.07$$
$$\text{Life} = 500\ (2.07)^{3.33} = 5890 \text{ hours}$$

11. Stress and deflection under maximum loading in bevel gear housing:

Output M_T = 213,000 in.-lb



$$ID = \frac{9.185}{9.065} \qquad \text{Wall thickness} = .325 - .425$$

Min. ID = 9.065

Min. OD = 9.065 + .650 = 9.715

$$J = \text{polar moment of inertia} = \frac{\pi\ (D^4 - d^4)}{32}$$
$$J = \frac{\pi\ (8907.82 - 6752.60)}{32} = \frac{\pi\ (2155.22)}{32} = 211.6$$
$$\theta = \frac{TL}{JG};\quad L = 4.0 \text{ in.};\quad G = 2,400,000 \text{ psi}$$
$$\theta = \frac{213,000 \times 4}{211.6 \times 2,400,000} = .001677 \text{ radians}$$

$$\theta = \frac{\delta}{R}$$

$$\delta = .001677 \times 12 = .020 \text{ in. (max. deflection)}$$

$$S_s = \frac{M_c}{J} = \frac{213{,}000 \times 4.86}{211.6} = 4892 \text{ psi}$$

12. Determination of cam travel at load through operating temperature range:

At room temperature when assembled:

℄ (center of drive ⟶ center of ecc. pin) = 7.7526

℄ (center of drive ⟶ center of X_2 gear) = 7.7942

(normal assembly position)

℄ (center of drive ⟶ center of X_2 gear, top dead center)

$$\begin{array}{r} = 7.7526 \\ + .0600 \\ \hline 7.8126 \end{array}$$

Therefore, X_2 gear center travels outward when brought to top dead center

$$\begin{array}{r} = 7.8126 \\ - 7.7942 \\ \hline .0184 \end{array}$$

Backlash at ring gear = .004 to .008

$.004 = 2\,\ell \tan 25$; $.004 = 2\,\ell\,(.46631)$

$$\ell = \frac{.004}{93262} = .004 \quad \gamma \quad \ell = .008$$

Therefore, .004 to .008 of this travel to top dead center can be accommodated by taking up the B.L. of the mesh. Under full load the ring deflects .010; consequently, it is quite feasible for the eccentrics to be assembled incorrectly and to have them corrected automatically by simply loading the system to full torque load. The separation force for .010 ring deflection is 1453 lb.

At temperatures of 220°F and at full load:

℄ (center of drive ⟶ center of ecc. pin) = 7.76946

℄ (center of drive ⟶ center of X_2 gear) = 7.79980

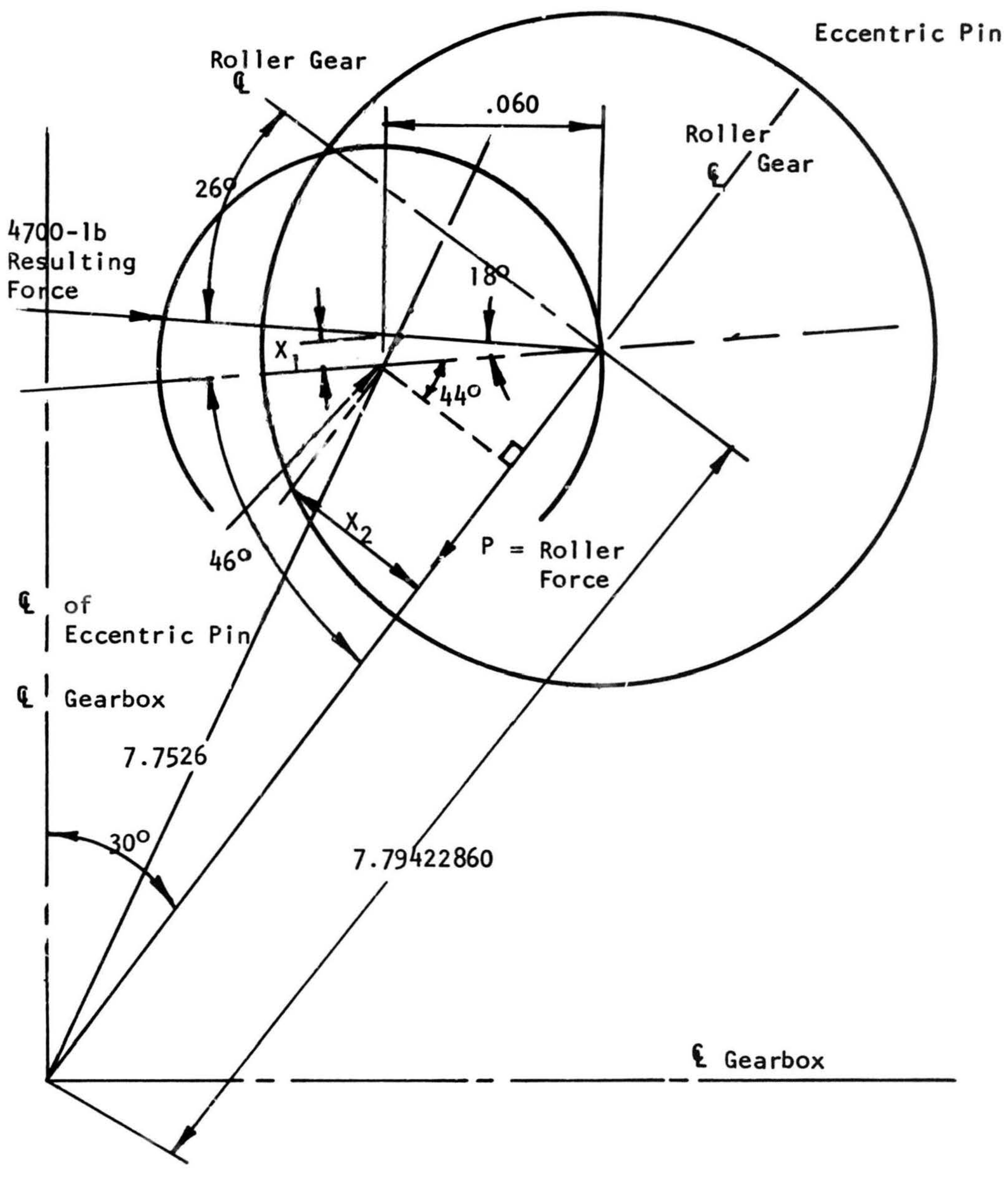
(operating position)

℄ (center of drive ⟶ center of X_2 gear top dead center)

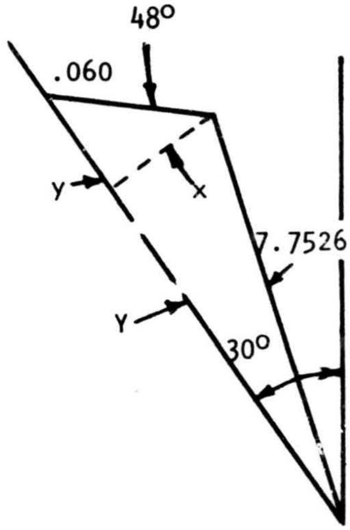
$$\begin{array}{r} = 7.76946 \\ + .06000 \\ \hline 7.82946 \end{array}$$

$$
\begin{array}{ll}
\text{Therefore, } X_2 \text{ gear center travels outward} & = 7.82946 \\
\text{when brought to top dead center} & \underline{-\ 7.79980} \\
 & .02966
\end{array}
$$

.004 to .008 of this travel can be accommodated by taking up the B.L., but it is doubtful that this could ever be accomplished at full load and at a temperature of 220°F.



This represents the eccentric when assembled with no preload but with roller gears on center locations ready for automatic preload.



Assuming that the eccentric is limited to an outward travel of 48° from the theoretically correct 44° by a pin limiting outward travel of gear center by 4°,

$$y = .060 \sin 48^\circ = .060(.74314) = .0445884$$

$$x = .060 \cos 48^\circ = .060(.66913) = .0401478$$

$$Y = \sqrt{7.7526^2 - .0401478^2}$$

$$= \sqrt{60.10280676 - .00161184}$$

$$= \sqrt{60.10119492}$$

$$Y = 7.7525$$

Gear center = y + Y = 7.7525
+ .0446
7.7971

With the stop pin located within an accuracy of 4°, the center of X_2 roller gear goes from theoretical correct C.D. of 7.7942 to 7.7971 = +.0029.

When the roller gear is loaded, the center becomes 7.7942 - .00188 = 7.79232 (this is roller gear center at 70°) (inward compression).

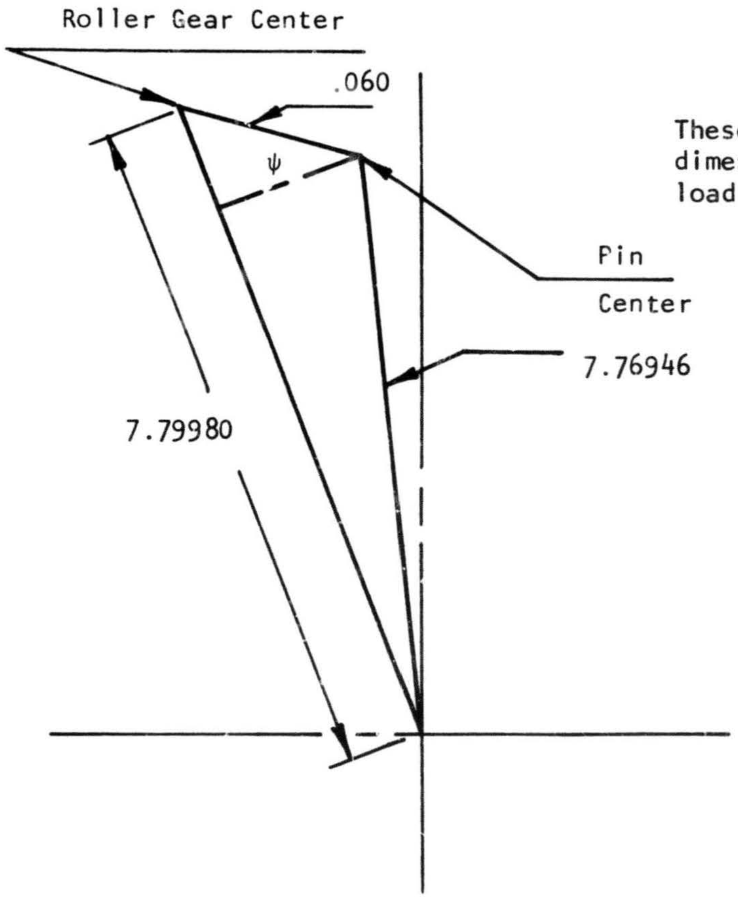
Assuming 70°F as starting condition, allow box and gear to go up to 220°F.

Roller gear center (stud) = expansion = 7.79232 $(6.4 \times 10^{-6}) \times 150$
= .00748

Pin center (magnesium housing) = expansion = 7.7526 $(14.5 \times 10^{-6}) \times 150$
= .01686

Roller gear center (at 220°) and load = 7.79232
+.00748
7.79980

Pin center (at 220°) = 7.75260
+.01686
7.76946



These are the dimensions under load and temperature.

$$\text{Cos } \theta = \frac{7.79980^2 + .060^2 - 7.76946^2}{2(7.79980)(.060)}$$

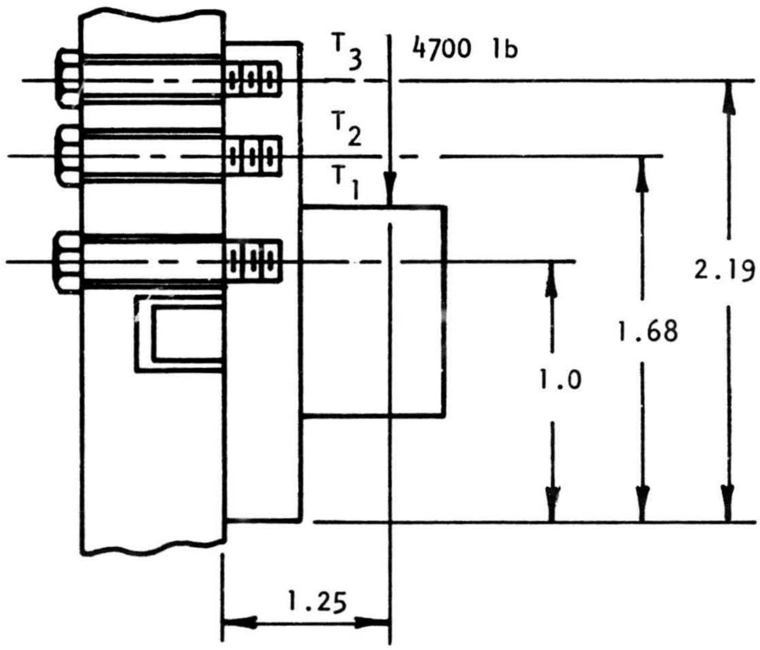
$$\text{Cos } \theta = \frac{60.83688004 + .0036 - 60.36450869}{.935976}$$

$$\cos\theta = \frac{.47597135}{.935976} = .508529$$

$$\theta = 59^{o} \quad 26'$$

$$\psi = 30^{o} \quad 34'$$

13.0 Determination of clamping force on eccentric cam pin plate:



$$\frac{T_1}{T_3} = \frac{1.00}{2.19}; \; T_1 = .45\,T_3$$

$$\frac{T_2}{T_3} = \frac{1.68}{2.19}; \; T_2 = .77\,T_3$$

$$T_1 \times 1.00 + T_2 \times 1.68 + T_3 \times 2.19 = 4700 \times 1.25$$

$$.45\,T_3 + 1.29\,T_3 + 2.19\,T_3 = 5900$$

$$T_3 = \frac{5900}{3.93} = 1500 \text{ lb}$$

The bolt size is 3/8 - six required - use 3/8 - 24 thread.

M_T = .2 D x L (approx.)

M_T = tightening torque = 300 in.-lb

L = tension force, lb

$$L = \frac{300}{.2 \times .375} = 4000 \text{ lb}$$

Total clamp force = 4000 x 6 = 24,000 lb (approx.)

Friction force = 24,000 (.2) = 4800 lb - bearing force
= 4700 lb

14.0 Determination of loads on eccentric cam roller bearing:

$$\text{Output torque} = M_t = \frac{63,000 \times 1100}{325} = 213,000 \text{ in.-lb}$$

$$\text{Input torque} = M_t = \frac{63,000 \times 1100}{12,350} = 5611 \text{ in.-lb}$$

$$\text{Reaction torque on pins (6 pins)} = \frac{213,000 - 5611}{6 \times 7.79} = \frac{207,389}{6 \times 7.79} = 4430 \text{ lb}$$

Layout analysis on forces in the roller drive confirms this horizontal torque vector.

Bearing load - 4700 lb from vector diagram.

$$\text{rpm of } X_2 \text{ roller} = 4117 \times \frac{1.8}{7.2} = 1030 \text{ rpm}$$

Selecting SKF bearing 22213C

P = 1.2 x 4700 - 5650 lb

Basic dynamic rating for bearing = 26,200 lb

$$\frac{C}{P} = \frac{26200}{5650} = 4.64$$

Life - 2500 hours

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13. ABSTRACT

This report reviews the dynamic load tests performed on two roller gear test units. The primary purpose of this task was to conduct a 200-hour dynamic load test at 100-percent design speed and 100-percent design load. Test units did not complete the scheduled testing because of test rig malfunction and minor design deficiencies which became apparent during the full-load portion of the test program. A total of 76.5 hours of testing was logged, of which 44.5 hours were at full-power condition of 1100 hp with an input shaft speed of 21,000 rpm and an output shaft speed of 325 rpm. Two design modifications were necessary during the testing program.

The testing system employed the principle of back-to-back coupling (four-square loop) of two test units in a closed-power path. An instrumentation system was provided to measure loads, speed, oil flows, pressures, temperatures, and vibratio levels. In addition, frequent analysis of the lubrication oil was made to determine the iron content in parts per million to provide a warning of incipient failure. The testing has confirmed that the roller gear drive design is sound and is feasible for helicopter speed reductions. It has further identified critical design parameters which must be considered in future designs.

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