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USAAVLABS TECHNICAL REPORT 69-69

ADVANCED TECHNOLOGY VTOL DRIVE TRAIN CONFIGURATION STUDY

By

James L. Lastine

Dr. Graham White

January 1970

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

CONTRACT DAAJ02-68-C-0082

SIKORSKY AIRCRAFT

DIVISION OF UNITED AIRCRAFT CORPORATION

STRATFORD, CONNECTICUT

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The efforts reported herein represent part of a continuing USAAVLABS program to investigate advanced technology for VTOL drive trains.

This report presents the results of an investigation to determine the effects of using advanced technology gas turbine engines with both pure and compound helicopter drive trains. Several drive train configurations have been evaluated to determine the influence of higher free turbine speeds upon the design of transmissions for the 1970 to 1975 time frame. The study further considers several new reduction concepts that offer increased redundancy and efficiency over more conventional designs.

Project 1G162203D14414
Contract DAAJ02-68-C-0082
USAAVLABS Technical Report 69-69
January 1970

ADVANCED TECHNOLOGY VTOL DRIVE TRAIN CONFIGURATION STUDY

Sikorsky Engineering Report 50590

By

James L. Lastine

Dr. Graham White

Prepared by

Sikorsky Aircraft
Division of United Aircraft Corporation
Stratford, Connecticut

for

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

SUMMARY

This report presents transmission system concepts that have been developed during a design investigation into single-rotor pure and compound helicopter drive trains.

Advanced technology gas turbine engines which utilize high output shaft speeds require greater speed reduction in helicopter drive trains. This study has considered the effects of these engines upon various drive train configurations. Accepting turbine power at free turbine speed and incorporating all speed reductions in the main gearbox have permitted consideration of several new reduction concepts which offer increased redundancy and efficiency over more conventional gearbox designs.

Specific areas considered in this study include torque dividing mechanisms supplying power to multiple load paths, redundant drives, and the use of epicyclic gearing for initial speed reduction.

This study indicates that the selected pure helicopter transmission system shown in Figures 29 through 34 can be fabricated for approximately 1600 pounds with an efficiency of 97.4 percent. The compound helicopter transmission system shown in Figures 41 through 44 exhibits a minimum efficiency of 96.7 percent with a resulting system weight of 1839 pounds.

Recently, the free-planet planetary concept for parallel axis speed reduction has been advanced for use in helicopter drive trains. A preliminary study incorporating this concept in a main gearbox has been included as Appendix IV.

FOREWORD

This report summarizes the investigation and preliminary design of several drive train configurations which are capable of satisfying the requirements of advanced pure and compound helicopters (Contract DAAJ02-68-C-0082, Project 1G162203D14414). Sikorsky Aircraft, a Division of United Aircraft Corporation, was the contractor for this study. The Gleason Works and the Curtiss-Wright Corporation provided pertinent data upon which portions of the study were based. Data concerning the advanced technology demonstrator engines were provided by the General Electric Company and Pratt and Whitney Aircraft, a Division of United Aircraft Corporation.

Meaningful technical contributions were made by L.R. Burroughs, Supervisor, Mechanical Systems Section, L. Webb, Design Engineer, and A. Korzun, Design Engineer, Mechanical Systems Section. USAAVLABS technical direction was provided by Mr. Wayne A. Hudgins, Mechanical Systems Branch, Aircraft Systems and Equipment Division.

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B_c	pitch diameter of bevel gear driven by planet carrier, in.
B_o	pitch diameter of combining bevel gear, in.
B_r	pitch diameter of bevel gear driven by ring gear, in.
c	rotational speed of planet carrier, rpm
C_p	specific heat, Btu/lb ^oF
F_{hp}	friction horsepower
n_o	rotational speed of combining bevel gear, rpm
P_1, P_2, P_3	power carried by members of differential gear as defined in Reference 6
p_c	power transmitted by planet carrier, arbitrary units
p_r	power transmitted by ring gear, arbitrary units
p_s	power transmitted by sun gear, arbitrary units
PLV	pitch line velocity, ft/min
Q_G	heat generated, Btu/hr
R	pitch diameter of ring gear, in.
R'	basic ratio of a differential gear, defined as R in Reference 6
S	pitch diameter of sun gear, in.
T	torque, lb-in.
T_c	torque transmitted by planet carrier, lb-in.
T_r	torque transmitted by ring gear, lb-in.
T_s	torque transmitted by sun gear, lb-in.
V_m	mean air velocity, ft/sec
W_o	oil flow, gal/min
β	flapping angle, deg

θ	angular displacement, rad
ρ	density of air, lb/ft^3
ρ_o	density of oil, lb/ft^3
ΔT	temperature difference, ^oF

INTRODUCTION

The effects of using advanced technology gas turbine engines in both pure and compound helicopter drive trains have been evaluated to determine the influence of higher free turbine speeds upon transmission system design.

Figures 1 and 2 depict the airframes that were used in this design study. Although these aircraft may not represent optimized configurations, they do provide a realistic basis for developing typical aircraft design data.

This preliminary design study of drive train configurations for single-rotor helicopters has included several different design concepts that were considered for the intended aircraft applications. A comparison of these different transmission systems has been conducted and is presented herein. The concepts that were considered include a conventional main transmission design that serves as a basis for comparison with the other reduction concepts, some of which include gearing arrangements not presently used in any known applications. On the basis of the comparative evaluations, one system has been developed for incorporation in each of the two types of helicopters.

A primary goal of this drive train design study has been to obtain increased component and system reliability and maintainability. The technology to be utilized in fabricating the drive train components is presently available. Thus, the components could be fabricated for use in current helicopter applications. For clarity, pure helicopter drive train configurations are denoted by 1-digit numbers (1 through 6) while the compound helicopter drive trains have 2-digit numbers (11 through 16).

BASIC DATA

DESIGN CHARACTERISTICS

USAAVLABS outlined the general design characteristics of two representative aircraft that would utilize two engines rated at 2000 shaft horsepower each. These general requirements included the following parameters.

Pure Helicopter

Twin engine, 4000 hp (military rated power) at sea level standard day.

Hover capability of 4000 feet, 95°F.

Cruise speed of 150 knots.

Rotor tip speed of 700 fps.

Compound Helicopter

Twin engine, 4000 hp (military rated power) at sea level standard day.

Hover capability of 4000 feet, 95°F.

Cruise speed of 200 knots.

Rotor tip speed of 600 fps.

MISSION REQUIREMENTS

Although the preceding requirements specify hover performance at 4,000 feet, which establishes rotor parameters, maximum horsepower is introduced into the drive train at sea level standard day operating conditions. Therefore, mission analyses should also include sea level standard day operations where 4000 horsepower can be supplied to the transmission system.

As a basis for this design evaluation, it has been assumed that missions at both altitudes will occur with equal frequency, and performance data have been developed for operation at both of these altitudes. Based on current operational practice, this assumption appears realistic, since higher horsepower requirements are imposed on the entire drive train during sea level standard day operation.

VEHICLE DESCRIPTION

Considering the preceding aircraft requirements, two representative vehicles have been designed for use in this design study. The general arrangements of these aircraft are shown in Figures 1 and 2. In addition, the design parameters describing these representative aircraft are given in Tables I and II.

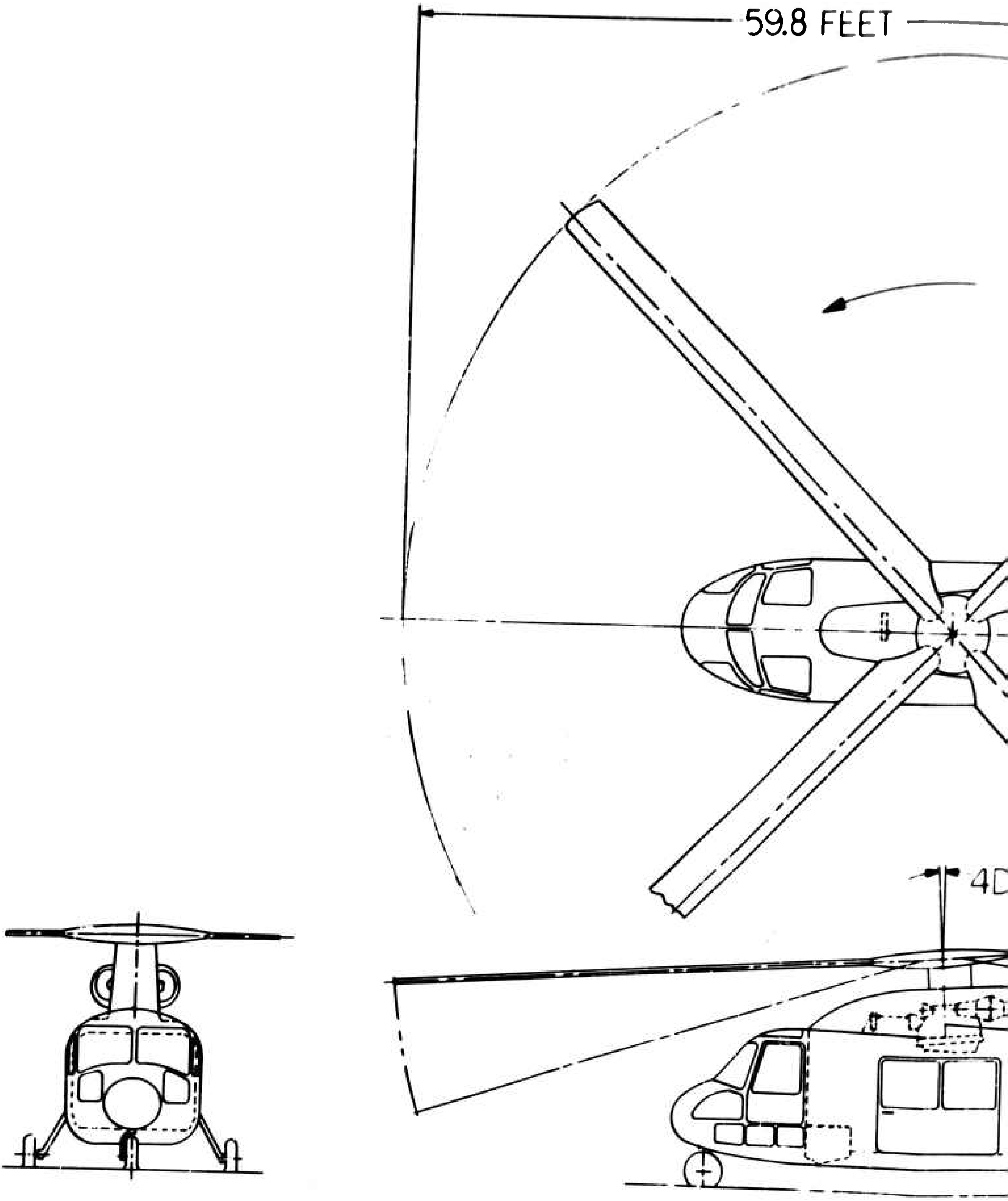
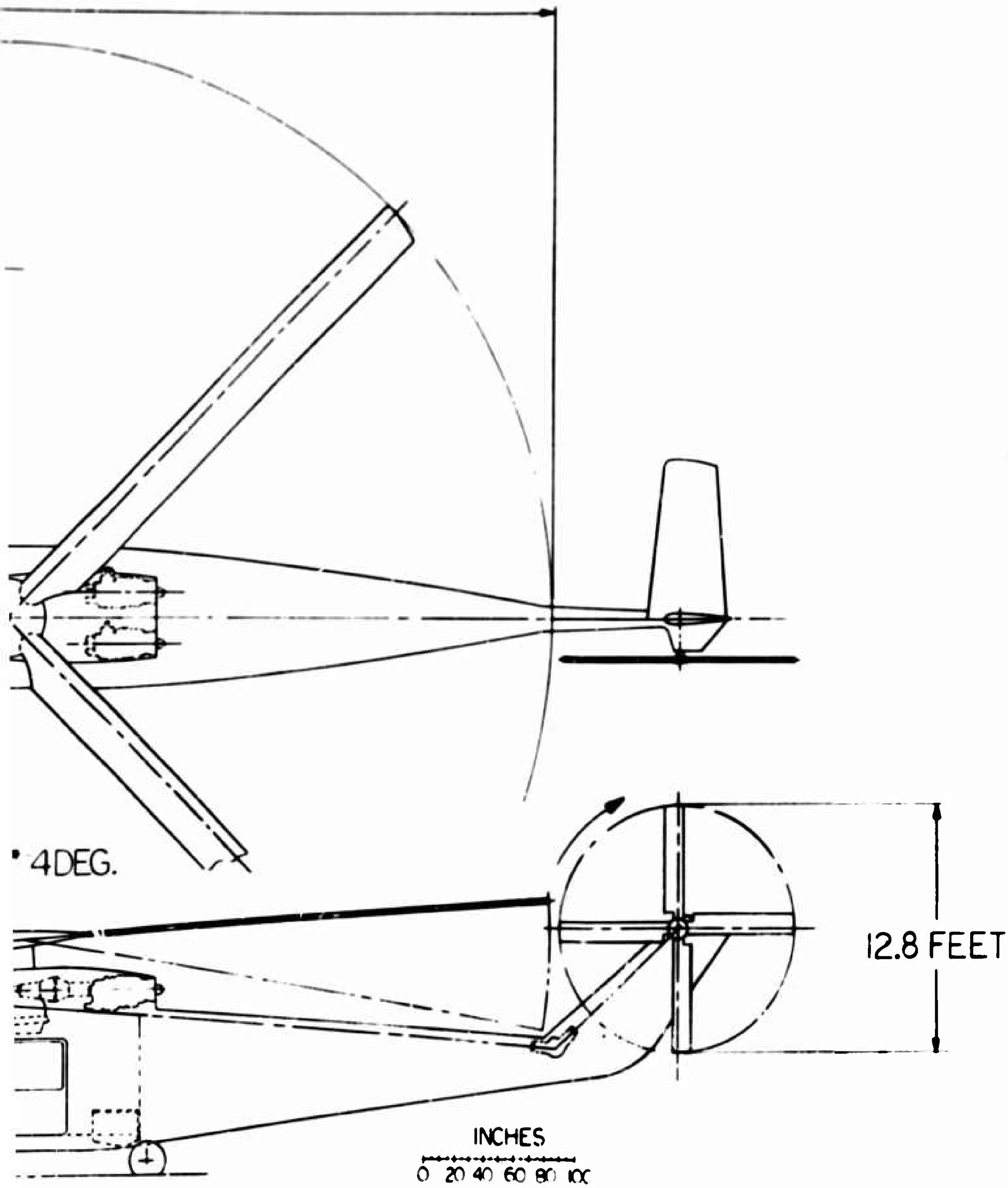


Figure 1. Aircraft Configuration, Pure Helicopter.

A



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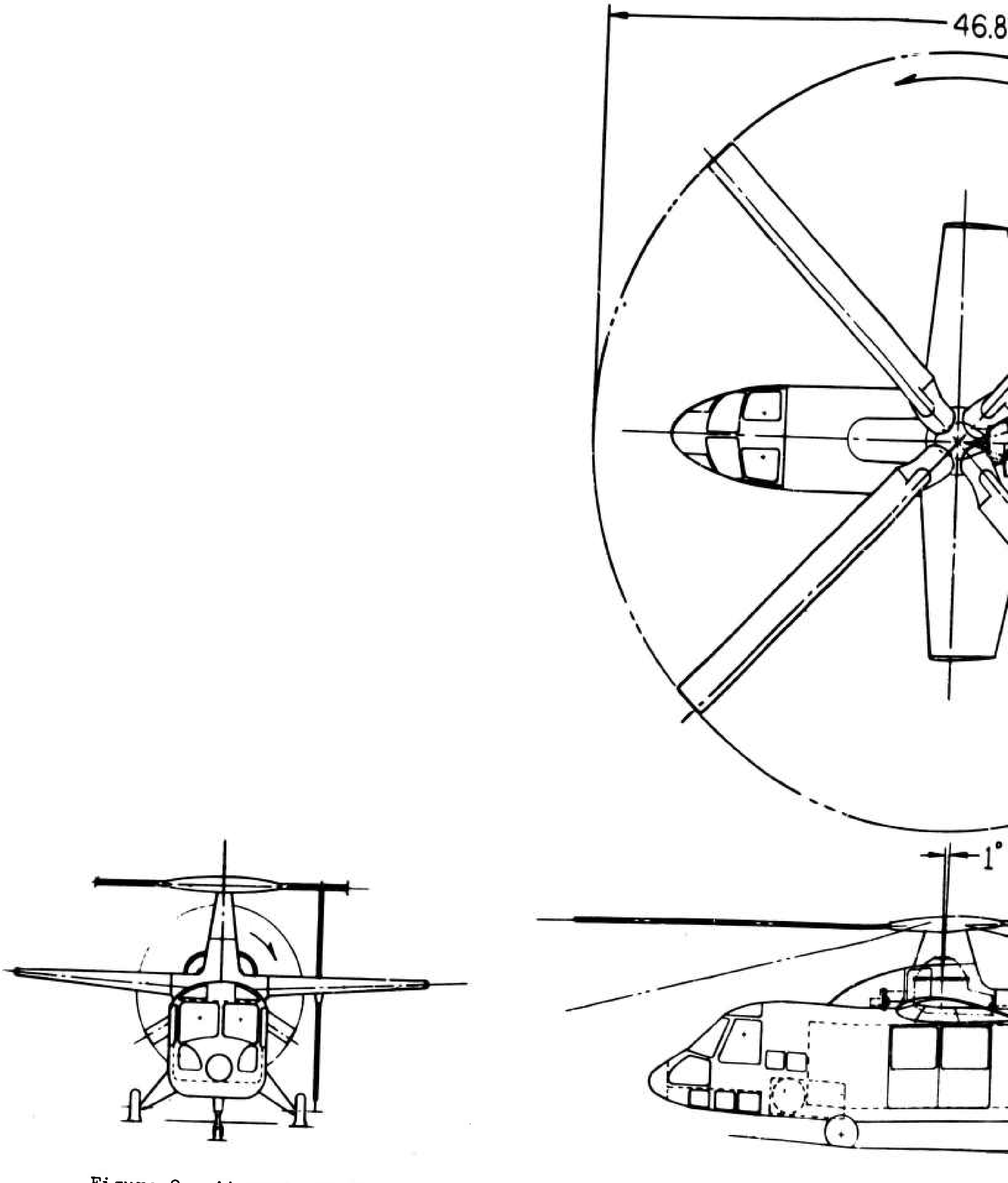
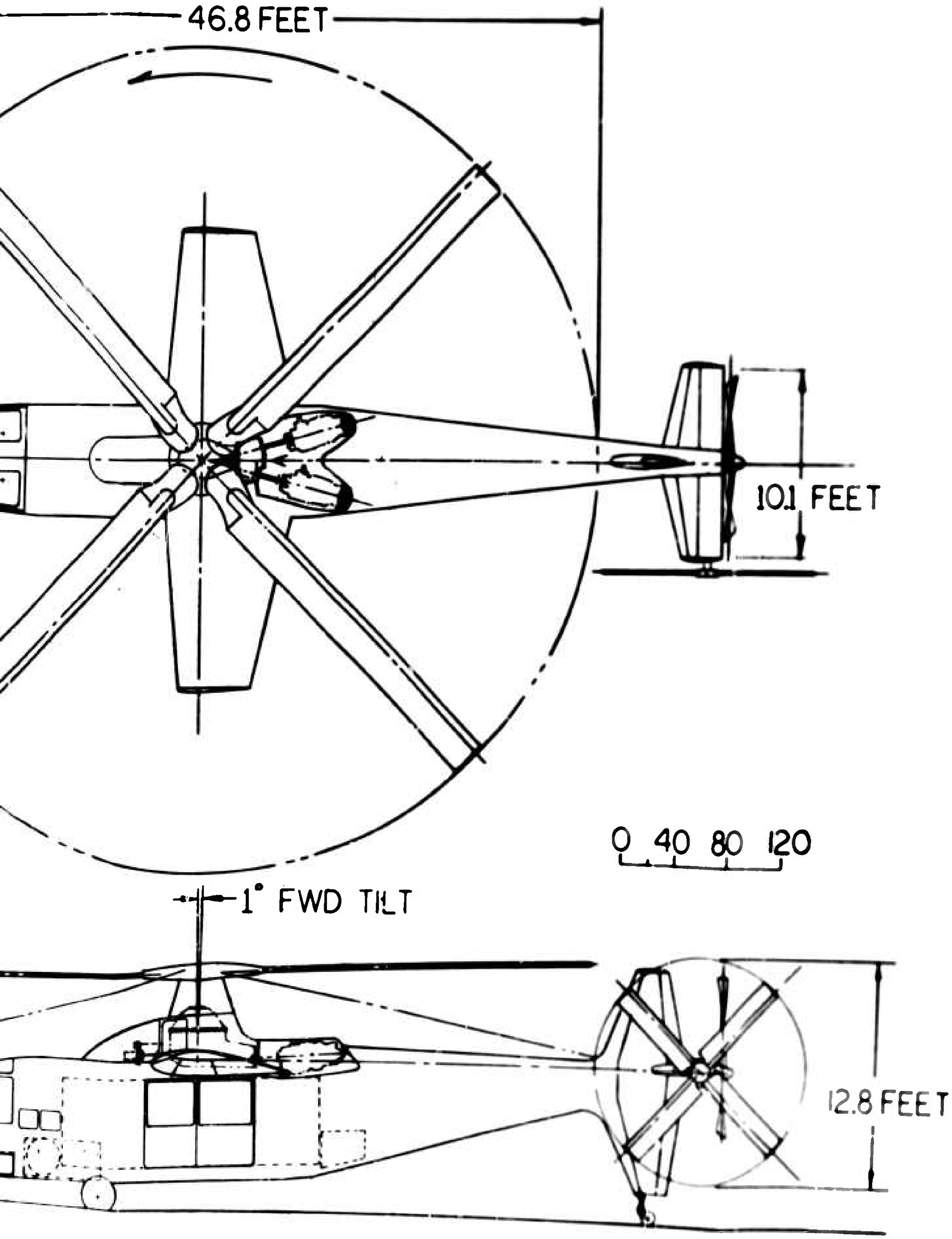


Figure 2. Aircraft Configuration, Compound Helicopter.

A



B

TABLE I
DESIGN PARAMETERS,
PURE HELICOPTER

Item		Value
Weight Empty, lb		11,340
Design Gross Weight, lb		19,620
Alternate Gross Weight, lb		23,550
Fixed Useful Load, lb		630
Mission Fuel, lb		2,588
Payload, lb		5,062
Engines (2)		
Type		Advanced Technology
Mil Pwr, SLS, hp (each)		2,000
Mil Pwr, 4,000 feet 95°F, hp (each)		1,535
Input Speed, rpm		15,000
Alternate Input Speed, rpm		30,000
Cruise Velocity, kn		150
Maximum Velocity, kn		160
Rotor Parameters	Main Rotor	Tail Rotor
Diameter, feet	59.8	12.8
rpm	223.67	1045
Tip Speed, fps	700	700
Disc Loading, psf	7	28
Solidity	0.10	0.248
Number of Blades	4	4
Aspect Ratio	12.73	5.13
Head Moment Constant, ft-lb/degree	2910	160

TABLE II
DESIGN PARAMETERS, COMPOUND HELICOPTER

Item		Value
Weight Empty, lb		10,720
Design Gross Weight, lb		15,420
Alternate Gross Weight, lb		18,500
Fixed Useful Load, lb		630
Mission Fuel, lb		2,528
Payload, lb		1,542
Engines (2)		
Type		Advanced Technology
Mil Pwr, SLS, hp (each)		2,000
Mil Pwr, 4,000 feet 95°F, hp (each)		1,535
Input Speed, rpm		15,000
Alternate Input Speed, rpm		30,000
Cruise Velocity, kn		200
Dash Velocity, kn		210
Propeller Diameter, feet		10.1
No. of Propeller Blades		3
Propeller Tip Speed, fps		900
Rotor Parameters	Main Rotor	Tail Rotor
Diameter, feet	46.8	12.8
rpm	245	1045
Tip Speed, fps	600	700
Disc Loading, psf	9	28
Solidity	0.1238	0.248

TABLE II-Continued		
Item	Value	
	Main Rotor	Tail Rotor
Number of Blades	4	4
Aspect Ratio	10.28	5.13
Blade Twist, degrees	-8^{o}	0^{o}
Head Moment Constant, ft-lb/degree	1970	160
Wing Parameters		
Area, square feet		144
Aspect Ratio/Taper Ratio		4/0.5
Span, feet		24.0
Root Chord/Tip Chord, feet		8.0/4.0
Root Airfoil/Tip Airfoil		63_3A418/63_1A412

MISSION ANALYSIS

Representative power and flapping spectrums have been derived for the pure and the compound helicopters using current typical mission requirements and operational experience as general guidelines. The anticipated frequency of occurrence and distribution of shaft horsepower to the major segments of the drive train of the pure helicopter are given in Tables XXXV and XXXVI. Corresponding flapping spectrums (Tables XXXVII and XXXVIII) outline the anticipated main and tail rotor blade flapping angles and frequency of occurrence. Similar data for the compound helicopter are given in Tables XXXIX through XLII.

Various accessories are also required on both of these aircraft, and drive provisions must be included in the drive train. Accessories considered for this design study are listed in Table XLIII, including the maximum power required for their operation.

These initial power spectrums were derived using estimated drive train efficiencies. No iterative calculations were made to update these preliminary power spectrums following system selection, as such effort appeared unwarranted for this study. Although minor discrepancies exist between the input power and the actual aircraft requirements, the overall effect of these discrepancies on the design of the drive train is of secondary importance and has not been considered in this preliminary drive train study.

DESIGN LOADS

Gearing and Shafting

The required design loads for each dynamic component can be determined by applying the cumulative damage theory to the power spectrums and representative S-N curves, using 10,000 hours as the minimum service objective. Design powers for the various drive train components are as follows.

Pure Helicopter

Component	Design Power, hp
Main Gearbox Input Drive (each engine)	2000
Main Rotor Drive	3400
Tail Rotor Drive	370
Accessory Drive	123

Compound Helicopter

Component	Design Power, hp
Main Gearbox Input Drive (each engine)	2000
Main Rotor Drive	3300
Propeller Drive	2633
Tail Rotor Drive	320
Accessory Drive	123

Bearings

Bearings in helicopter gearboxes are subjected to varying speeds and loads, as summarized by the preceding loading spectrums. To evaluate the effect of variations in speed, load, and percentage of time during which the variable loads and speeds occur, it has been shown in Reference 1 that bearing selection should be based upon a "weighted average loading." Roller bearing life varies inversely as the 3.33 power of the applied load, whereas ball-bearing life is inversely related to the 3.0 power. However, to be conservative and simplify the selection, the 3.33 power will be considered for all bearing applications. The weighted average power (or prorated power) upon which bearing selection is based can be determined using the following equations.

$$T = \frac{63{,}025\ (hp)}{(rpm)}$$

$$\text{Prorated Torque T at } rpm_1 = T_1 \left[t_1 + t_2 \left(\frac{T_2}{T_1} \right)^{3.33} \frac{rpm_2}{rpm_1} + \ldots\ldots + t_n \left(\frac{T_n}{T_1} \right)^{3.33} \frac{rpm_n}{rpm_1} \right]^{0.3} \qquad (1)$$

Assuming an equal number of missions at 4000 feet, 95°F and at sea level standard conditions, the prorated design powers and flapping angles used in this study are given in Table III.

DESIGN REQUIREMENTS

Design Lives

The dynamic components used in these transmission systems have been designed for 10,000-hour service intervals, while bearings have been designed to achieve a minimum unfactored bearing life of 3,000 hours. These design values should permit operation of the drive train with service intervals that approach "on-condition operation", following an adequate testing and "debugging" program at accelerated loading conditions. This entire concept is explained and outlined in Reference 2, which explains testing concepts.

Material Selection

The materials selected for components evaluated in this study are currently used in similar applications on production aircraft. AMS 6265 vacuum processed alloy steel will be used for all primary drive train gearing, while AMS 6260 alloy steel will be used in other gearing applications. Critical bearing applications also utilize vacuum melt 52100 steel, where its higher fatigue strength justifies its use. Housings will be AZ 91C magnesium castings or ZK 60A magnesium forgings. Titanium will be used in fabricating such components as epicyclic gearing planet carrier plates, engine gearbox gimbal mounts, and miscellaneous flanges and spacers.

Design Procedures

The components used in the transmission configurations of this report have been designed by using analytical techniques and formulas similar to those used in Reference 3. The allowable stresses used in these formulas are outlined on the following pages.

TABLE III
PRORATED DESIGN VALUES,
PURE AND COMPOUND HELICOPTERS

	Mission Environment 4,000 Feet $95^{\circ}F$	SLS	Design Value
Pure Helicopter			
Prorated Horsepower			
Main Gearbox Input Drive (each engine)	993	1278	1135
Main Rotor Drive	1687	2173	1930
Tail Rotor Drive	148	175	162
Accessory Drive	65	65	65
Prorated Flapping Angle, Degrees			
Main Rotor	2.89	3.48	3.19
Tail Rotor	2.12	2.53	2.33
Compound Helicopter			
Prorated Horsepower			
Main Gearbox Input Drive (each engine)	1240	1533	1387
Main Rotor Drive	1526	1888	1707
Propeller Drive	1421	1719	1570
Tail Rotor Drive	151	185	168
Accessory Drive	65	65	65
Prorated Flapping Angle, Degrees			
Main Rotor	2.34	2.83	2.59
Tail Rotor	1.97	2.35	2.16

Allowable Design Stresses

Gearing

Material: AMS 6260 or AMS 6265 steel

Spiral Bevel Gears

Bending Stress, $*F_b$ = 30,000 psi

Compressive Stress, $*F_c$ = 210,000 psi

*Note: Allowable bevel gear design stresses are currently being evaluated under a USAAVLABS program. Preliminary analyses indicate that the above allowable stresses represent conservative values for the 1970 to 1975 time frame, especially considering the use of vacuum processed alloy steel.

Spur Gears

Bending Stress, F_b = (35,750 - .704 PLV) psi

Compressive Stress, F_c = 145,000 psi

Planetary Gears

Bending Stress, F_b = (31,500 - .625 PLV) psi

Compressive Stress, F_c = (120,000 to 145,000) psi

Planet Carrier Plates

Material: 6 Al - 4V Titanium

Ultimate Tensile Stress, F_{tu} = 130,000 psi

Steady Bending Stress, F_b = 42,000 psi

Plate Deflection (max) δ = 0.0006 inch per inch

Shafting

Material: AMS 6260 steel

R_c 30 - 45 core hardness

Bending Stress, F_b = 19,500 psi

Shear Stress, F_s = 30,000 psi

Material: AMS 5000 steel

Ultimate Tensile Stress, F_{tu} = 200,000 psi

Bending Stress, F_b = 21,800 psi

Shear Stress, F_s = 35,000 psi

Housings

Material: AZ 91C-T6 Magnesium Castings

ZK 60A-T5 Magnesium Forgings

For design allowables, see Reference 4.

PURE HELICOPTER TRANSMISSION SYSTEM

DESIGN GUIDELINES AND OBJECTIVES

As outlined by the contract and in the Basic Data Section, the primary design objectives to be considered in designing the pure helicopter transmission systems included the following:

Develop lightweight drive train systems incorporating a minimum number of reduction stages.

Consider two engine output speed levels of approximately 15,000 and 30,000 rpm.

Install integral lubrication and cooling systems wherever possible.

Use technology that will be available during the period 1970 to 1975.

Consider component and system maintainability and reliability foremost in all configurations.

The above objectives are self-explanatory guidelines. The need for fabrication in the immediate future, however, indicates that only small improvements in transmission characteristics can be realized from the use of higher stress levels. Greater improvements might be realized either by modifying current gearing arrangements or considering reduction techniques not previously used. For this reason several alternative drive system configurations merit consideration, since new configurations may lead to significant weight savings independently of a change in stress levels.

A baseline configuration has been developed in order to provide a standard against which the other transmission designs can be compared. This baseline configuration incorporates two 15,000-rpm engine inputs, and the design follows current practices and techniques. Meaningful reliability and weights data have been obtained by comparing the other gearbox designs with the baseline configuration and with current production gearboxes.

Since the system efficiency is largely determined by that of the main gearbox, initial effort was devoted to developing efficient engine to main rotor drive trains. To facilitate maintenance and reduce the number of maintenance hours per flight hour, considerable attention has been given to component accessibility, separate subsystem isolation, and fault detection equipment. The arrangement of individual components has been selected to permit more subsystem maintenance to be performed while the primary component remains installed on the aircraft. By isolating the return oil in various portions of the gearbox with appropriate filters and chip detectors, trouble shooting is simplified, gearbox contamination is isolated, and the various subsystems can be examined and overhauled. Such practices will reduce the cost of maintenance by permitting more gearbox subsystems such as freewheel units, individual gear reductions, etc, to be overhauled without necessitating gearbox removal and affecting several related aircraft systems such as the rotor and control systems.

CANDIDATE SYSTEMS

To satisfy the preceding objectives a variety of drive system arrangements were reviewed and compared, bearing in mind the need for an efficient transfer of power from the engines to the main rotor. From these preliminary studies, six different drive train arrangements for the pure helicopter were selected; the primary design parameters of engine speed and number and type of gear reductions are shown in Table IV.

A system description of each different arrangement follows Table IV. Although particular engine arrangements are depicted in Figures 23 through 28, several alternate arrangements could also have been used at the discretion of the designer. These alternate engine locations are discussed following the descriptions of the various configurations.

TABLE IV
PURE HELICOPTER
DRIVE TRAIN CONCEPTS

Configuration Number	Schematic Figure Number	Engine rpm	Number and Type of Gear Reductions - Engine to Main Rotor
1 (Figure 23)	3	15,000	2 bevel 2 stage planetary
2 (Figure 24)	4	30,000	1 bevel 1 spur 2 stage planetary
3 (Figure 25)	6	15,000	1 spur 1 bevel 1 spur 1 stage planetary
4 (Figure 26)	7	15,000	2 bevel 1 compound planetary
5 (Figure 27)	8	30,000 (15,000 alternate)	1 epicyclic 1 bevel 1 spur
6 (Figure 28)	9	30,000 (15,000 alternate)	1 epicyclic 1 bevel 1 spur 1 stage planetary

Configuration Number 1

Using conventional gearing techniques well proven in service, this concept accepts power at 15,000 rpm from two engines located behind the main gearbox. The engines are mounted in a horizontal plane with centerlines located 15.5 inches on either side of the longitudinal axis of the aircraft. As shown in Figure 3, a right-angle spiral bevel gear set is used to accomplish the first speed reduction, supplying power to a horizontal shaft rotating at 7368 rpm. A freewheel unit must also be included between each engine and the rotors in the drive train. Acceptable operation is currently being obtained with such units rotating at more than 6000 rpm. Therefore, this output shaft provides a convenient location for mounting such a unit, since engine speed is beyond the range of current experience with roller-type freewheel units and thereby prevents installation on that shaft in this baseline configuration.

To accomplish the second speed reduction, another set of spiral bevel gears is employed. Input pinions, as shown in Figures 3 and 23, transfer power to a bevel gear that combines the power from both engines and is mounted concentric with the main rotor shaft. This bevel gear supplies power to the tail rotor drive train and to the accessories as well, including during autorotation or engine shutdown.

A two-stage planetary unit is necessary to give 223.67 rpm at the main rotor, the first and second stages having reduction ratios of 3.458 and 2.785 respectively. The first-stage planetary unit and the main rotor shaft diameter are, to some extent, interrelated and influence the design of the transmission configuration. The rotor shaft must be rigid enough to keep the deflection and bearing slopes within acceptable limits and yet be small enough to pass through the internal diameter of the sun gear. This requirement, to some extent, either limits the reduction ratio which can be obtained from the first-stage planetary unit or increases the diameter of the ring gear. Some care must also be taken in the mounting of the planetary units in order that the gear tooth meshes can be, as far as possible, isolated from any load-induced motions of the main rotor shaft and the transmission casing.

This transmission configuration represents what may currently be termed a conventional transmission arrangement. For this reason it serves as a base with which to compare alternative transmission configurations. Data on the gearing used in this concept are listed in Table V.

Configuration Number 2

In an effort to eliminate a bevel gear mesh and combine engine power using spur gearing, Configuration Number 1 was modified so as to replace the combining bevel gear meshes with spur gears. As engine clearance requirements indicate that parallel engines must be separated a minimum of 31 inches, a large-diameter spur gear is used to

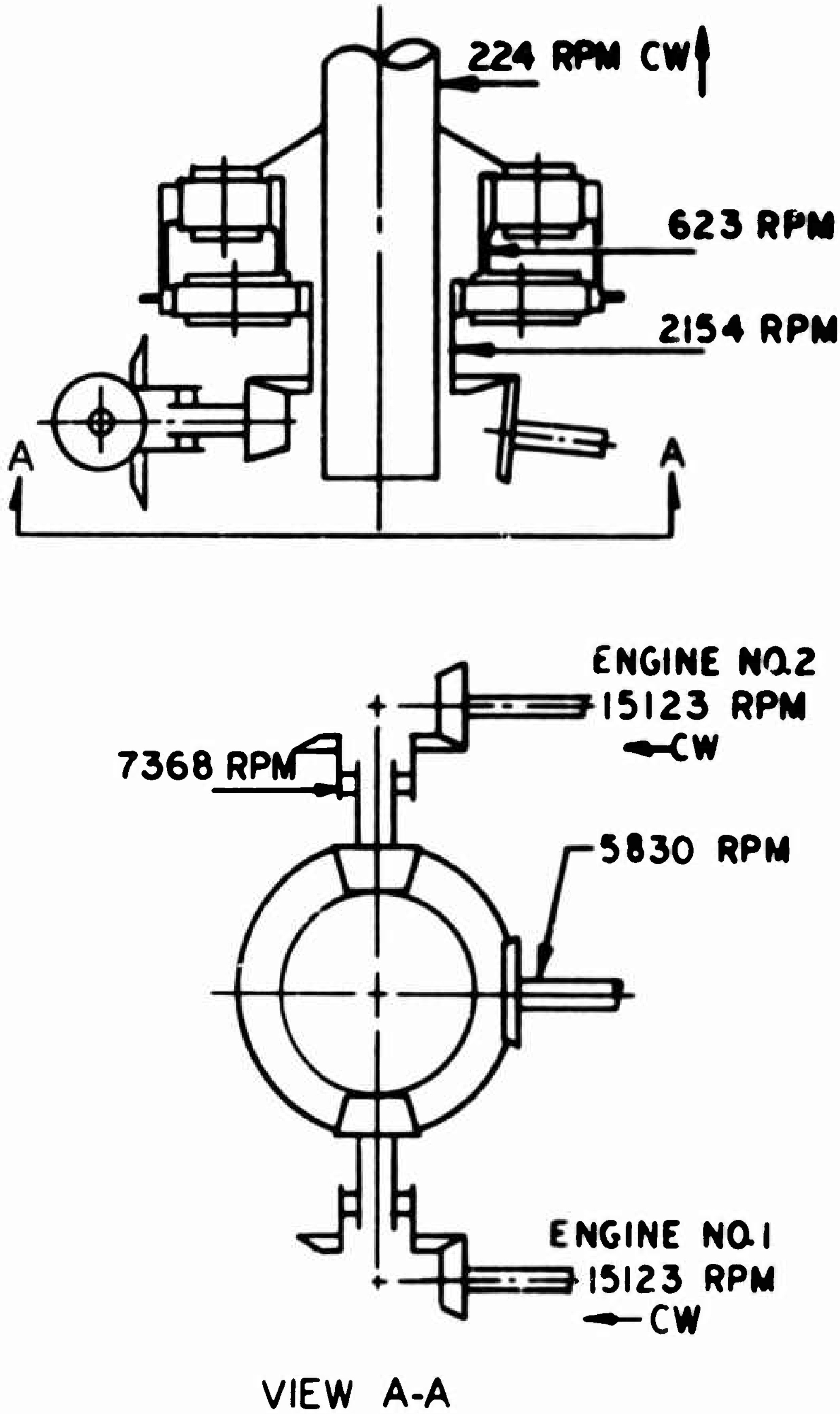


Figure 3. Gearing Schematic of Configuration Number 1.

TABLE V
GEAR DATA,
CONFIGURATION NUMBER 1

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	15,123	19	4.60	1.6	18,200
Input Bevel Gear	7,368	39	9.44	1.6	18,200
2nd Stage Bevel Pinion	7,368	19	4.9	2.65	9,500
2nd Stage Bevel Gear	2,154	65	16.76	2.65	9,500
1st Stage Planetary (3.4576 RR)					
Sun	2,154	59	9.833	1.43	9,680
Pinion (6 req'd)	2,100	43	7.167	1.18	
Ring	0	145	24.167	0.73	
2nd Stage Planetary (2.7848 RR)					
Sun	623	79	13.167	2.23	2,460
Pinion (10 req'd)	1,017	31	5.167	1.98	
Ring	0	141	23.500	1.14	
Carrier	224				

combine engine power as shown in Figure 4. It is convenient to accomplish a large speed reduction at this combining spur gear mesh, and this feature makes the configuration suitable for use with the 30,000-rpm engines. The combining gear also supplies power to the tail rotor drive train and the required accessory drives.

The input bevel gearing provides a reduction ratio of 2.48 to 1 and reduces the speed of the output shaft to 12,068 rpm. A roller-type freewheel unit is located on this vertical output shaft, isolating each engine from the remainder of the drive train in the event of engine shutdown. Although this speed exceeds operational experience with such units, testing indicates that a freewheel unit configuration could be developed to perform satisfactorily at this speed. The spur gear reduction ratio of 4.84:1 provides an output speed of 2493 rpm to the sun gear of the first-stage planetary unit. This combining spur gear is mounted independently of, and concentric with, the main rotor shaft, as is shown in Figures 4 and 24. A two-stage planetary unit provides the final speed reduction in the main rotor drive train, with the main rotor rotating at 224 rpm.

While this transmission arrangement uses conventional gearing techniques, careful attention must be given to the mounting arrangement of the large-diameter combining spur gear and the need to minimize the slope of the rotor shaft at the attachment point of the second-stage planet carrier. Another area requiring special consideration is the bearing arrangement used on the input bevel pinion rotating at 29,907 rpm. At this rotational speed most of the capacity of the bearings is taken up by the centrifugal loads from the balls themselves. In an effort to determine a recommended bearing installation for an input bevel pinion rotating at this speed, several bearing arrangements were considered and analyzed using iterative techniques outlined in Reference 3. The initial gearbox configurations included bevel pinions with a 35-degree cone angle, 35-degree spiral angle, and 20-degree pressure angle. Six of these arrangements are shown in Figure 5 and include different numbers and combinations of bearings and both left- and right-hand spiral bevel pinions. A summary of the resulting bearing lives using standard bearing materials is also given in Figure 5. As shown, the individual bearings of arrangement 3 exhibit the most uniform lives. But the use of vacuum melt steel is required to obtain the required bearing life of 3000 hours, and this material should provide an acceptable bearing installation. Data on the gear reductions used in Configuration Number 2 are contained in Table VI.

Configuration Number 3

Configurations 1 and 2 each utilize a two-stage planetary gear reduction to drive the main rotor. Since these reductions account for more than 20 percent of main gearbox weight, this was a logical area in which to make changes and suggest alternate speed reduction techniques. Greater flexibility in the location of the engines could also be obtained by using spur gears to accomplish the initial speed

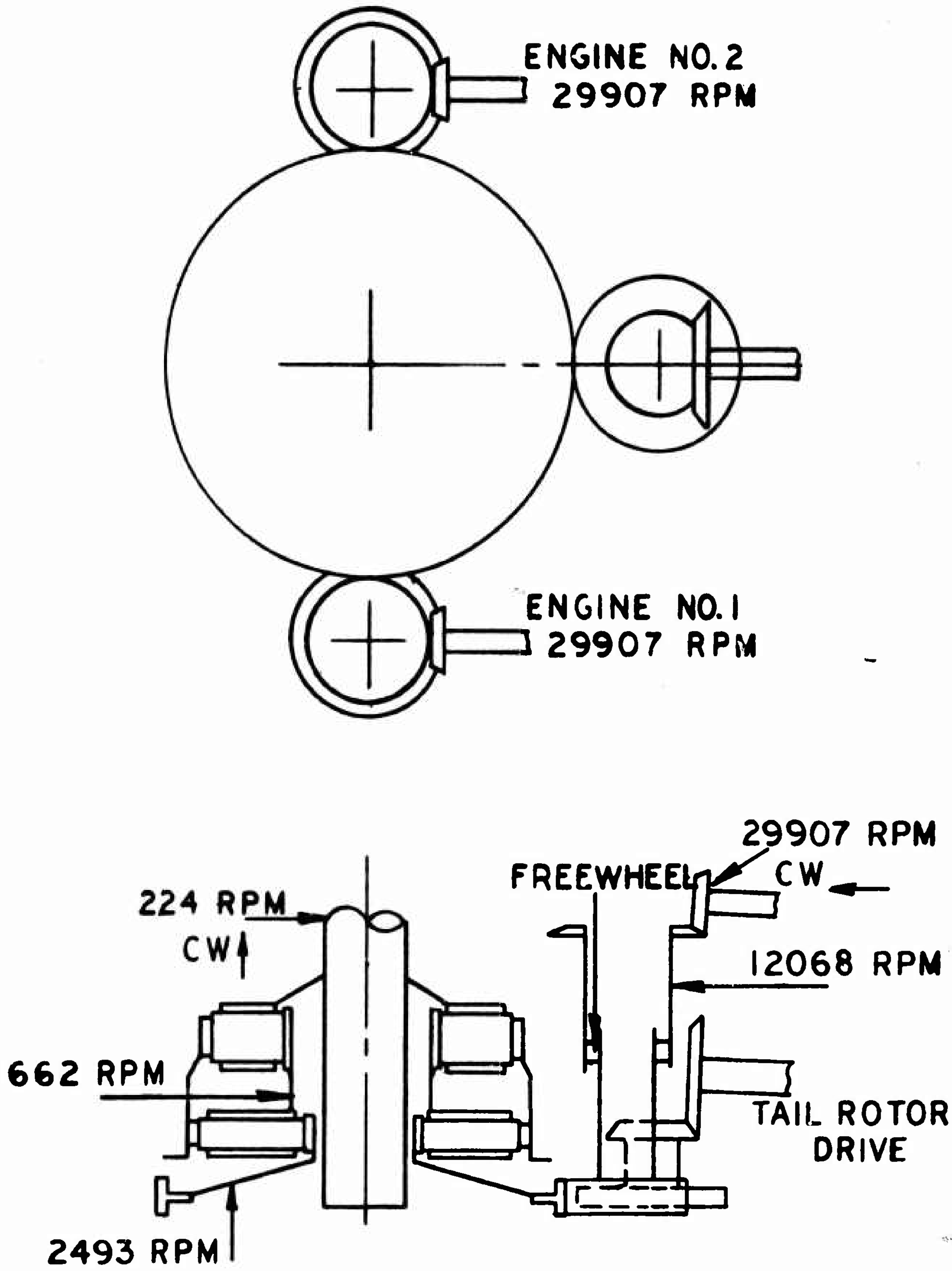
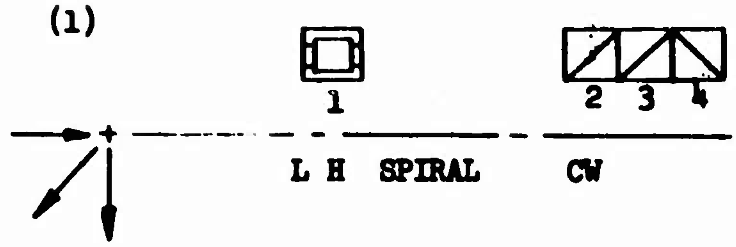


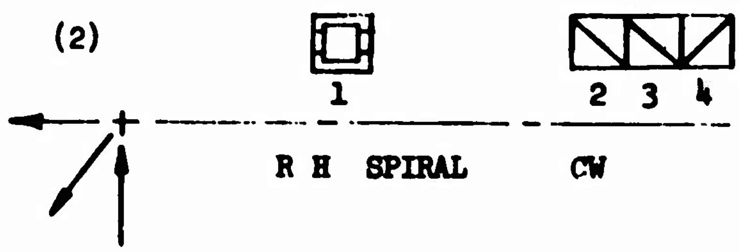
Figure 4. Gearing Schematic of Configuration Number 2.

(1)



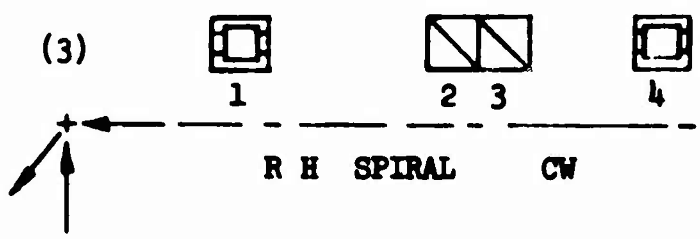
Brg No	Brg Size	Life (hours)
1	214	2099
2	211	332
3	211	816
4	211	319

(2)



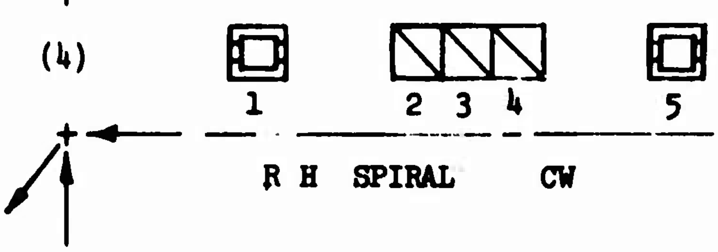
1	215	972
2	214	283
3	214	341
4	214	243

(3)



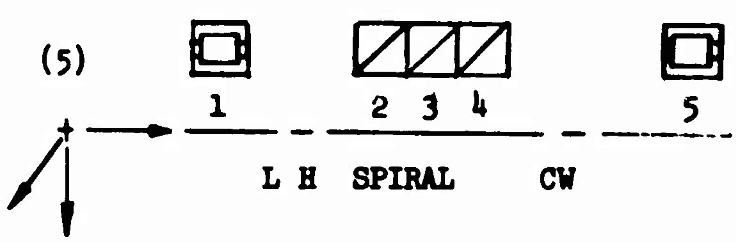
1	314	1288
2	211	1712
3	211	1712
4	110	2098

(4)



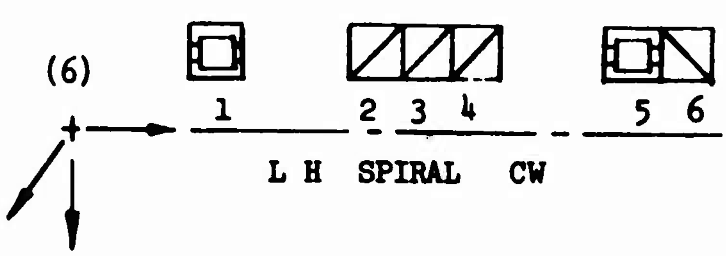
1	214	370
2	211	887
3	211	887
4	211	625
5	110	2128

(5)



1	214	937
2	211	1678
3	211	1678
4	211	1678
5	110	4239

(6)



1	214	962
2	211	1052
3	211	1052
4	211	1052
5	110	4691
6	211	524

Figure 5. Bearing Arrangements, Input Pinion.

TABLE VI
GEAR DATA,
CONFIGURATION NUMBER 2

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	29,907	23	3.82	1.33	29,900
Input Bevel Gear	12,068	57	9.467	1.33	29,900
Spur Pinion	12,068	44	5.50	1.6	17,400
Spur Gear	2,493	213	26,625	1.5	17,400
1st Stage Planetary (3.7627 RR)					
Sun	2,493	59	7.375	1.85	3,540
Pinion (6 req'd)	2,077	52	6.500	1.60	
Ring	0	163	20.375	1.20	
2nd Stage Planetary (2.962 RR)					
Sun	662	79	9.875	3.44	1,130
Pinion (9 req'd)	912	38	4.750	3.19	
Ring	0	155	19.375	2.25	
Carrier	224				

reduction, as shown in Figures 6 and 25.

Thus, this configuration was designed with 15,000-rpm input shafts supplying power to a 1.97 reduction ratio spur gear mesh. This speed reduction permits a roller-type freewheel unit, similar to that of Configuration Number 1, to operate at 7594 rpm and to be installed prior to the spiral bevel gear mesh which, in turn, provides an input speed of 2690 rpm to the spur pinion. The output combining spur gear is mounted concentric with the main rotor shaft and transfers power to the sun gear of the one-stage planetary. This 3.449 reduction ratio planetary provides the final speed reduction prior to the main rotor. The combining spur gear also provides the required power to the accessory and the tail rotor drive trains. The elimination of the first-stage planetary unit of Configurations 1 and 2 means that, in the present configuration, the diameter of the main rotor shaft is not influenced by the sun gear diameter or vice versa, and the rotor shaft can be sized to obtain the required stiffness at a minimum weight. Data on the gearing used in this concept are included in Table VII.

Configuration Number 4

While investigating and developing various gearing concepts and gearbox configurations, a planetary unit with compound planet pinions was considered for the final speed reduction in the main rotor drive train. A gearbox with this type of reduction exhibited several apparent advantages, and the concept was examined in detail.

Initial designs were based on a 9 to 1 reduction ratio with 4 compound planet pinions, the fixed ring gear meshing with the smaller diameter pinions and the planet carrier driving the main rotor. The large diameter swept out by the rotating planet gears, however, made it necessary to modify the concept to a fixed planet carrier design with the output taken from the rotating ring gear.

This latter arrangement is shown in Figures 7 and 26, from which it is evident that the drive train from each engine consists of 2 bevel gear reductions. The output shaft from the first bevel reduction is positioned between the fixed lay shafts and inclined toward the center of the rotor shaft. If the planet carrier were rotating, as on the earliest design, the first reduction drive shaft would have to pass outside of the planet pinions and a very large diameter combining gear would then be necessary.

The first and second bevel gear reductions of 2.51 and 3.17 respectively give a speed of 1914 rpm to the combining bevel gear. A further reduction of 8.55 in the compound planetary section gives a rotor speed of 224 rpm. The freewheel is conveniently mounted between the first and second bevel gear reductions at a speed of 6069 rpm, this speed being in the region of current helicopter operational experience.

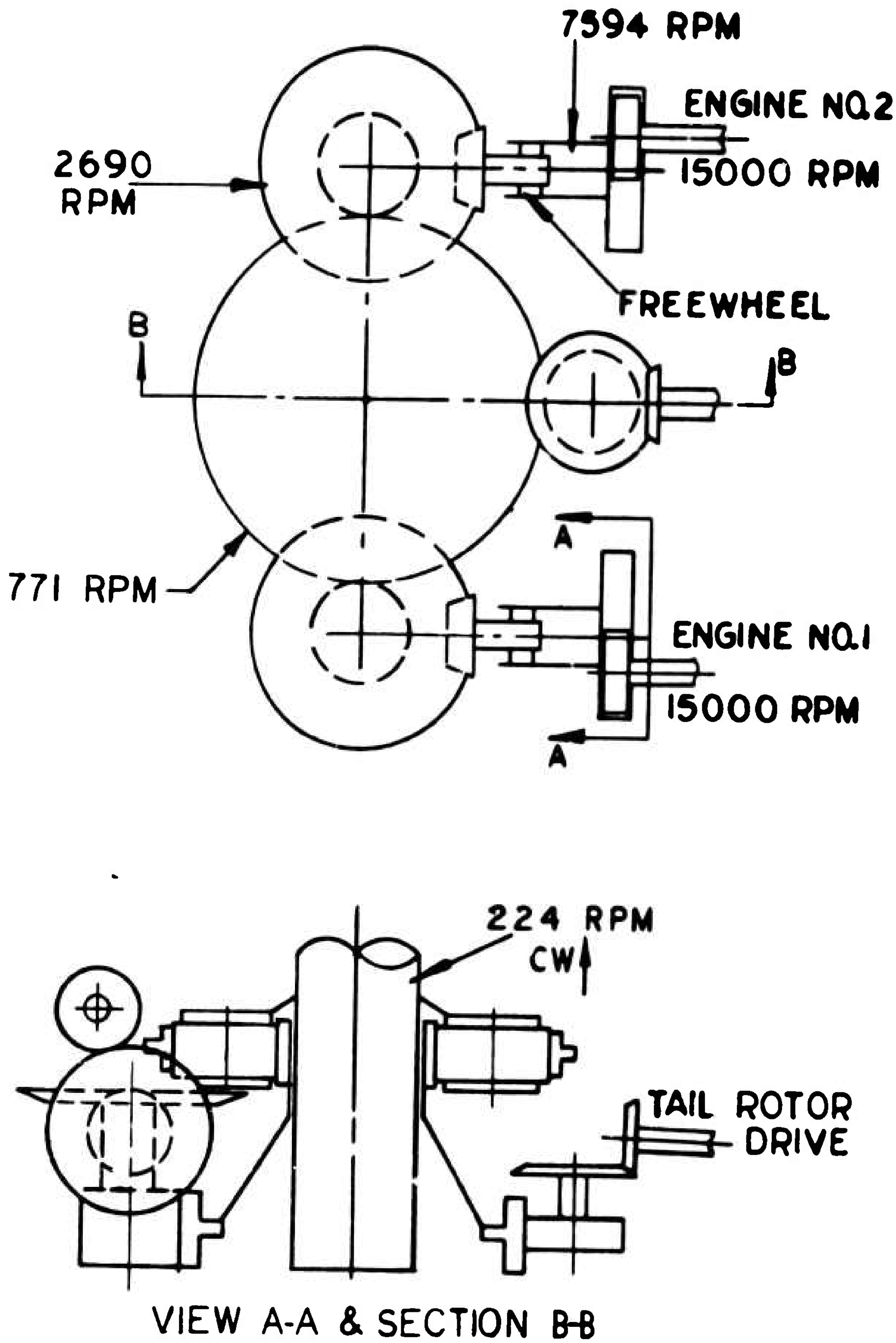


Figure 6. Gearing Schematic of Configuration Number 3.

TABLE VII
GEAR DATA,
CONFIGURATION NUMBER 3

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Spur Pinion	15,000	40	5.00	1.90	19,650
Input Spur Gear	7,594	79	9.875	1.90	19,650
Bevel Pinion	7,594	17	5.00	1.8	9,950
Bevel Gear	2,690	48	14.11	1.8	9,950
Spur Pinion	2,690	37	6.167	4.6	4,350
Combining Spur Gear	771	129	21.500	4.6	4,350
Planetary (3.449 RR)					
Sun	771	69	9.857	2.98	1,410
Pinion (7 req'd)	756	50	7.143	2.73	
Ring	0	169	24,143	1.97	
Carrier	224				

Planetary reduction units with compound planet pinions are known, from turbo-prop experience, to present difficulties in the design of the planet carrier, the main problem being that of obtaining adequate stiffness at an acceptable weight. Another potential problem area is the mounting of the large rotating ring gear which exceeds current design experience. Data on the gearing used in this concept are included in Table VIII.

Configuration Number 5

This transmission seeks to obtain a large speed reduction by using a combination of conventional gearing and a split-power concept not previously incorporated in helicopter, or other, transmission systems. The use of engines with output shaft speeds of 30,000 rpm dictates that the associated gearbox must provide a reduction ratio of 134 to 1. The difficulties associated with obtaining an acceptable bearing arrangement for high-speed spiral bevel gears, as outlined in Configuration Number 2, are avoided in this concept by connecting the sun gear of an epicyclic unit to the engine drive shaft as shown in Figures 8 and 27. By allowing the ring gear of the input epicyclic unit to rotate backwards relative to the sun gear, while the planet carrier rotates in the same direction as the sun gear, a large speed reduction is obtained. In particular, reduction ratios of 11.04 and 12.80 are obtained in the ring gear and the planet carrier drive trains respectively. If this same hardware were utilized in a conventional planetary reduction unit with a fixed ring gear, a reduction ratio of 6.48 would be obtained, which would necessitate an additional gear mesh elsewhere in the drive train. Further, the use of a fixed ring gear would be impracticable, for then the planet carrier bearings would experience large centrifugal loads in addition to the tooth mesh loads and an acceptable planet pinion configuration could not be obtained within the gear envelope.

By providing counterrotating output shafts from the epicyclic unit, the power from each engine (2000 hp) is split into two paths so that the rotating ring gear and the rotating planet carrier each transmit approximately 1000 horsepower. A detailed description of counterrotating epicyclic units, showing how the power split proportions affect the reduction ratios, is contained in Appendix III.

Following the epicyclic unit, spiral bevel gear meshes, with reduction ratios of 2.26 and 1.95 in the ring gear and planet carrier drive trains respectively, reduce the speed to 1199 rpm so that a final combining spur gear mesh to the rotor shaft is sufficient to achieve a rotor speed of 224 rpm. The combining gear shown in Figure 8 is driven by four spur gears, and the gear ratios in the transmission are arranged so that each of the four spiral bevel and spur gear meshes transmits 25 percent of the power demanded by the main rotor. Figure 8 also shows how the engines are arranged to achieve a minimum housing weight while maintaining adequate engine separation for accessibility during maintenance.

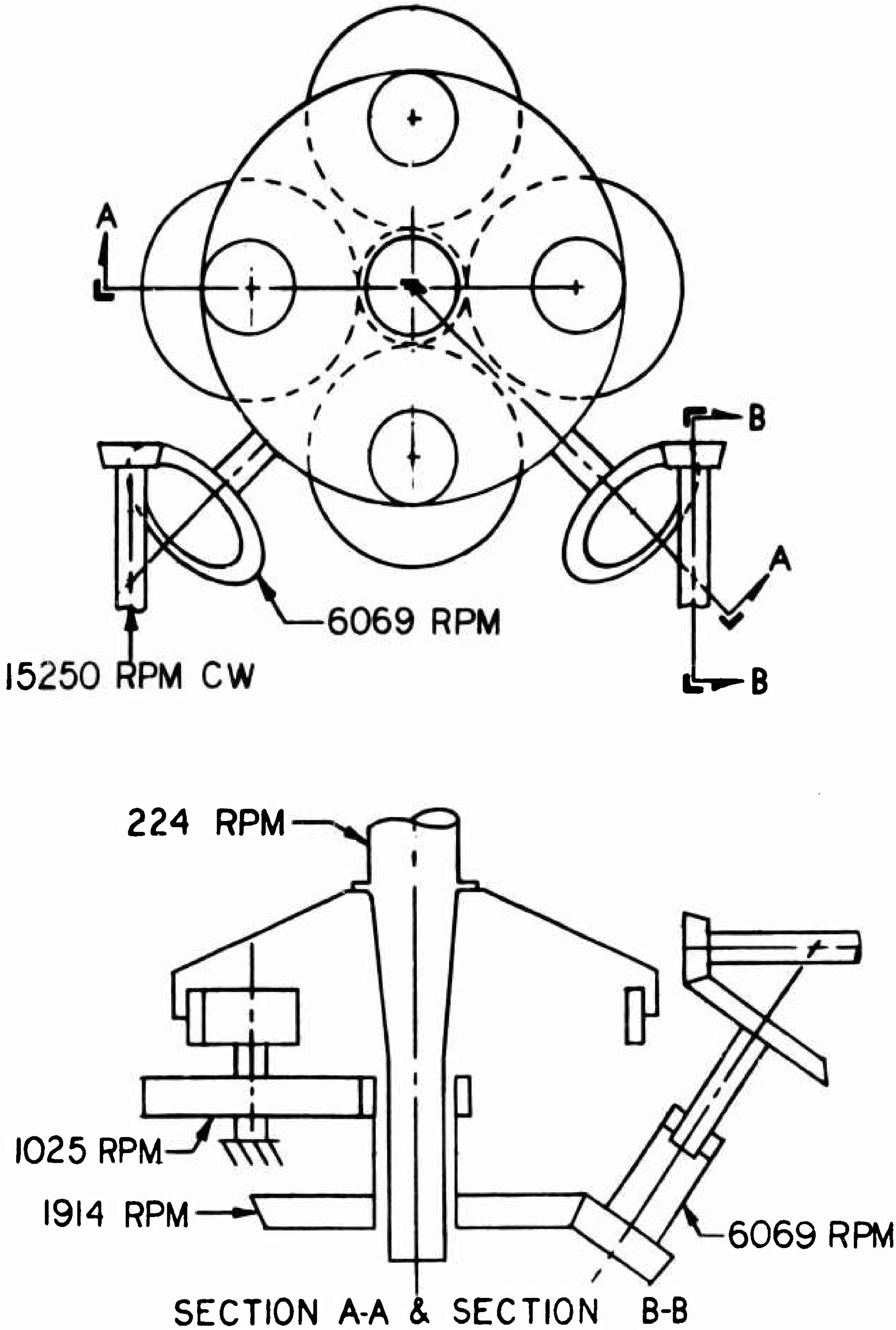


Figure 7. Gearing Schematic of Configuration Number 4.

TABLE VIII
GEAR DATA,
CONFIGURATION NUMBER 4

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	15,250	39	4.50	1.67	18,000
Driven Bevel Gear	6,069	98	11.30	1.67	18,000
2nd Bevel Pinion	6,069	35	7.00	2.16	11,100
2nd Bevel Gear	1,914	111	22.20	2.16	11,100
Compound Planetary					
Sun Gear Mesh					
Sun	1,914	45	7.5	2.50	3,760
Pinion (4 req'd)	1,025	84	14.0	2.46	3,760
Ring Gear Mesh					
Pinion (4 req'd)	1,025	24	6.0	3.54	1,610
Ring	224	110	27.5	3.50	1,610

The low weight and high efficiency achieved are thus a direct result of reducing the equivalent number of gear meshes carrying engine power to a total of three; i.e., a counterrotating epicyclic unit, one spiral bevel gear reduction, and one spur (or helical) gear reduction. In addition, low weight results from taking a much larger speed reduction (5.36 to 1) in the final spur reduction than the usual planetary unit can provide. The combining gear arrangement, however, must be used in conjunction with the split power concept, for only in this way can the torque to the main rotor be distributed between four gear meshes while also maintaining acceptable face widths on the combining gearing. Other features made possible by the split power concept include ideal load sharing between gear tooth meshes as a result of an inherent load-balancing effect, drive redundancy as far as the combining gear which drives the main rotor, the facility for placing an oil cooler inside of the large-diameter rotor shaft, and the use of freewheels sized for one-half engine power. Although engine-mounted freewheel units would provide the lightest freewheel installation, no experience is available on units operating at 30,000-rpm. For this reason and bearing in mind the design guidelines and objectives outlined previously (page 15), a 30,000-rpm freewheel installation could not be recommended at the present time. Freewheel units subject to lower overrun speeds can be used by placing the freewheel unit after the epicyclic unit in either the ring gear or the planet carrier drive trains of each engine. On engine shutdown,the sun gear of the epicyclic unit is brought to rest and the freewheel unit, designed for one half engine power, overruns as explained in Appendix III. Such a freewheel position involves some weight penalty when compared with an engine-mounted freewheel unit, but acceptable operation is assured. Gearing data used in this concept are included in Table IX.

Configuration Number 6

The transmission arrangement shown in Figures 9 and 28 incorporates a split-power concept, and the input drive section has much in common with that of Configuration Number 5. Thus, 30,000-rpm engines can be accepted by each counterrotating epicyclic unit, and a large speed reduction is obtained from this unit while the input power is divided between the ring gear and the planet carrier drive paths. The purpose of power splitting at the input section, therefore, is to allow the use of high-speed engines and at the same time split the input power of each engine into two approximately equal parts so that the bevel gears downstream of the counterrotating epicyclic unit can be sized for half engine power. Each of the counterrotating bevel gears meshes with a combining bevel gear which can be positioned to achieve the required angular turn between the engine and rotor shaft axes. It follows that the spur gear pinion integral with the combining bevel must be sized for engine power.

A compact design is obtained by positioning the input epicyclic unit between the counterrotating bevel gears, and initial design studies were based on such a layout, But then it was found that, to clear the ring gear, bevel gears of somewhat larger diameters and narrower

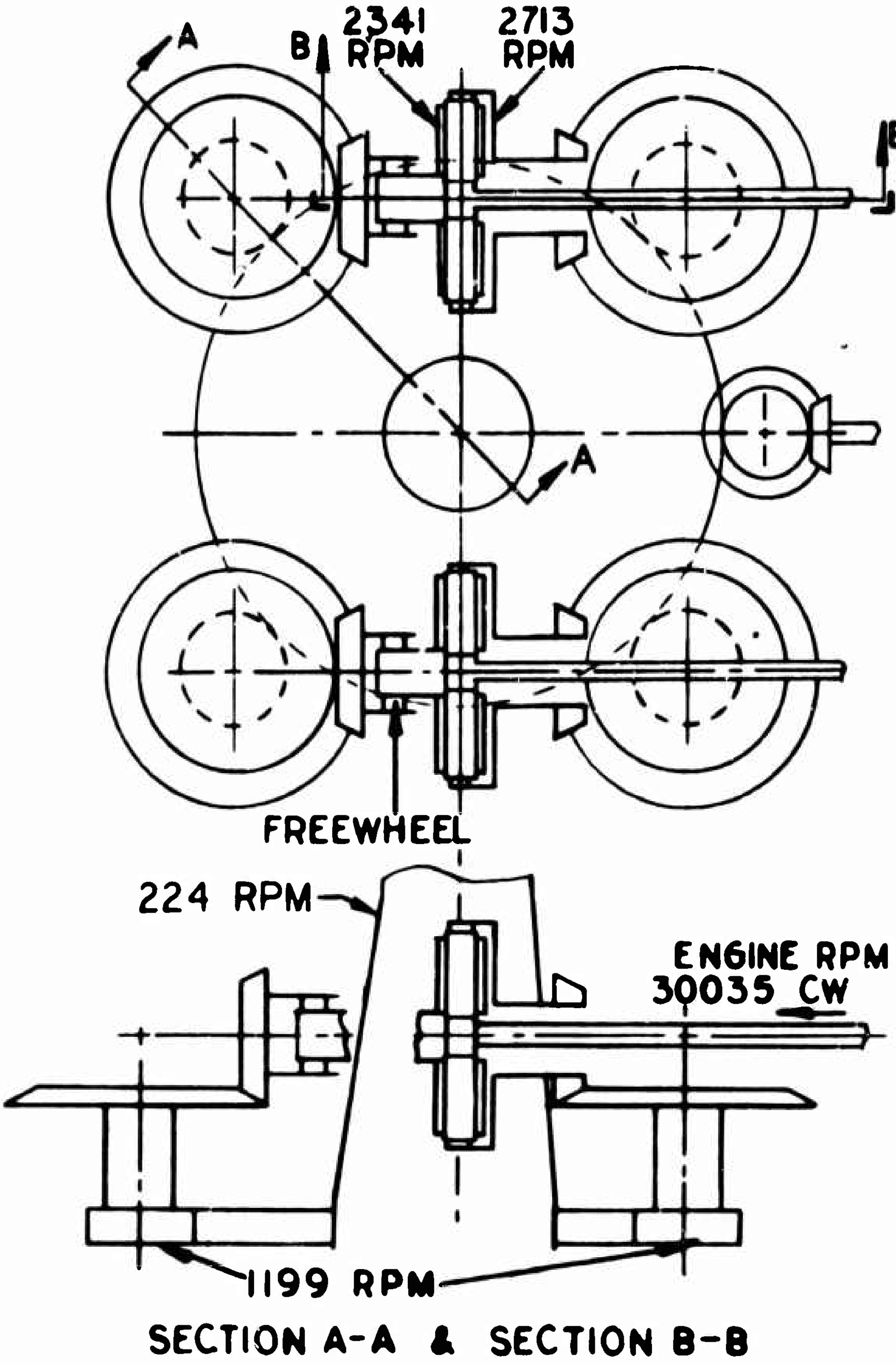


Figure 8. Gearing Schematic of Configuration Number 5.

TABLE IX
GEAR DATA,
CONFIGURATION NUMBER 5

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Epicyclic Unit					
Sun	30,035	25	2.08	1.80	15,070
Pinion (3 req'd)	-10,022	56	4.67	1.59	
Ring	-2,713	137	11.42	1.20	
Carrier	2,341				
Bevel Pinion (Driven by Ring Gear)	2,713	19	6.50	2.3	4,620
Bevel Gear	1,199	43	14.72	2.3	4,620
Bevel Pinion (Driven by Carrier)	2,341	21	7.40	2.4	4,530
Bevel Gear	1,199	41	14.45	2.4	4,530
Spur Pinion	1,199	25	6.25	4.19	1,960
Spur Gear	224	134	33.50	4.09	1,960

face width than normal were required which, in turn, necessitated larger diameter housings and larger bearing spans for the bevel gears. A convenient way of avoiding the above problems is to place the planetary unit outside the bevel gear cluster as shown in Figure 9; then there is no restriction on the size of the epicyclic unit, and the counterrotating bevel gears can be of a diameter consistent with the torque transmitted. The gearbox arrangement shown in Figure 9 also allows a short, high-speed input drive shaft between the engine and the sun gear, together with the facility of removing the epicyclic gear and its bearings as a unit for maintenance or overhaul without affecting the rest of the gearbox.

Each combining bevel gear of Figure 9 carries the power of one engine to a combining spur mesh, this mesh providing a speed reduction of 3.06. A final reduction of 3.58 in the output planetary unit gives a rotor speed of 224 rpm. It is seen from Table XII that the efficiency and weight of the transmission are slightly inferior to Configuration Number 1, a conventional arrangement.

The input epicyclic section warrants further consideration and comment, since either the components or their arrangements are somewhat different than those usually utilized in helicopter gearboxes. Figure 9 shows that a large speed reduction can be taken at the input epicyclic unit, at the same time achieving a power split and avoiding high centrifugal loads on the planet carrier bearings. It will be realized that the input section of the transmission is virtually identical with that of the alternative split-power transmission arrangement of Configuration Number 5; for this reason all the characteristics and arguments put forward in describing that system also apply. The 30,000-rpm engine speed can be accepted with this design scheme, and it will be seen from Figure 9 that the input epicyclic unit gives a planet carrier speed of 2292 rpm and a ring gear speed of 2631 rpm from a 29,963 rpm sun gear speed. The combining bevel, which is integral with the combining spur pinion, rotates at 2450 rpm. Engines with 15,000 rpm can be accommodated by altering the proportions of the input epicyclic unit as dictated by the expressions derived in Appendix III. It will be found that lower engine speeds involve a smaller ring-gear-to-sun-gear diameter ratio.

It is convenient to mount the freewheel unit inside one of the bevel gears. Then, with the arrangement and speeds shown in Figure 9, the freewheel must carry the ring gear torque at 2631 rpm. On shutdown of an engine, the freewheel experiences an overrun speed of 5331 rpm, as explained in Appendix III.

An advantage of the layout shown in Figure 9 is that the ring gear torque passes into the freewheel unit directly underneath the freewheel roller path and then into the bevel gear under the gear teeth. In this way the freewheel and bevel gear shafts enclosing the freewheel unit carry no torque. A freewheel of slightly higher torque capacity (27,800 in-lb versus the present 23,600 in-lb) can be placed between the planet carrier and the bevel gear driven by the planet carrier. A slight weight penalty then is incurred, but there is a marked advantage in that the freewheel unit can be removed for either inspection or maintenance without disturbing the bevel gears.

The set of three bevel gears seen in Figure 9 serves to convert the counterrotating motions of the planetary unit into a single direction drive. At the same time an angle turn is achieved which places the output drive parallel to, but offset from, the rotor. An advantageous feature is that engine inclinations in any plane can be accepted by suitable proportioning of the bevel gear diameters. In the arrangement shown in Figure 9, care has been taken to insure that the unequal diameters of bevel gears necessitated by the rotor shaft inclination are exploited in such a way that the larger diameter bevel gear carries the torque from the planet carrier while the smaller diameter bevel gear carries the smaller torque from the ring gear. In this way a virtually equal power split can be achieved, i.e., half engine power through each counterrotating bevel gear. This implies that the tangential loads on a diameter of the combining bevel gear are equal, thus obtaining a balance of radial loads.

Engine power is split between the counterrotating bevel gears in a proportion which depends on the epicyclic gear proportions, the engine inclination and the relative size of bevel gears. Expressions enabling the power distribution to be calculated are developed in Appendix III. Gearing data used in this concept are included in Table X.

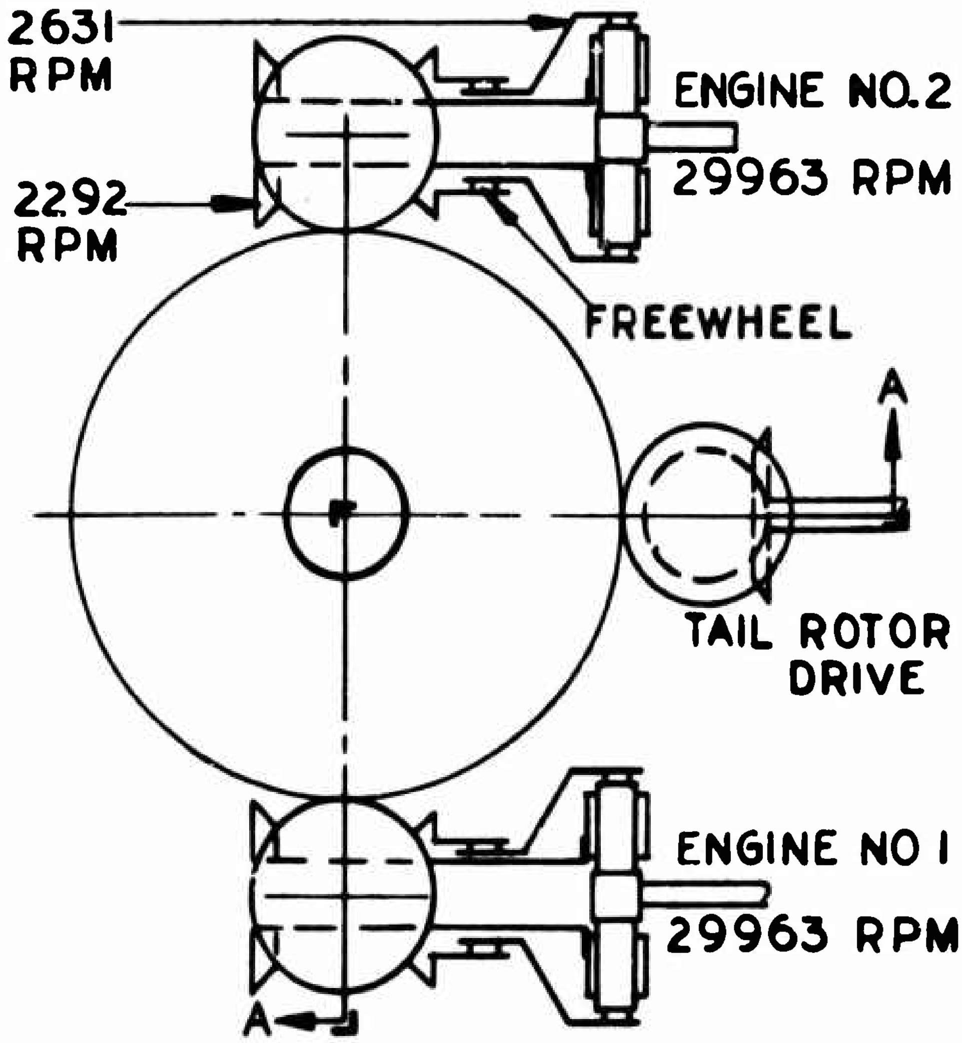
ALTERNATE ENGINE INSTALLATIONS

The preceding drive train configurations depict six engine arrangements which were selected to complement the various drive train installations. In addition, alternate engine locations were considered that could also be accommodated by modification to the main gearbox configurations shown in Figures 23 through 28. Several of these alternate engine arrangements, which could also be used in these representative aircraft, are presented in Table XI. The associated gearbox modifications required to accommodate the engine installation, together with appropriate comments, are included in Table XI.

ENGINE MOUNTING ARRANGEMENTS

Ideal engine installations are seldom obtained in practice as a result of turbine engines being used for a variety of applications. It is usual to find that certain design features incorporated in the engine cause interface problems in certain installations. The demonstrator engines considered in this study, however, have been specifically developed with helicopter applications in mind; for this reason it would appear reasonable to believe that interface and installation problems should be minimized.

In addition to the basic engine, together with its associated fuel and control systems, several subsystems of the engine also influence the drive train and airframe design. These additional requirements are imposed by



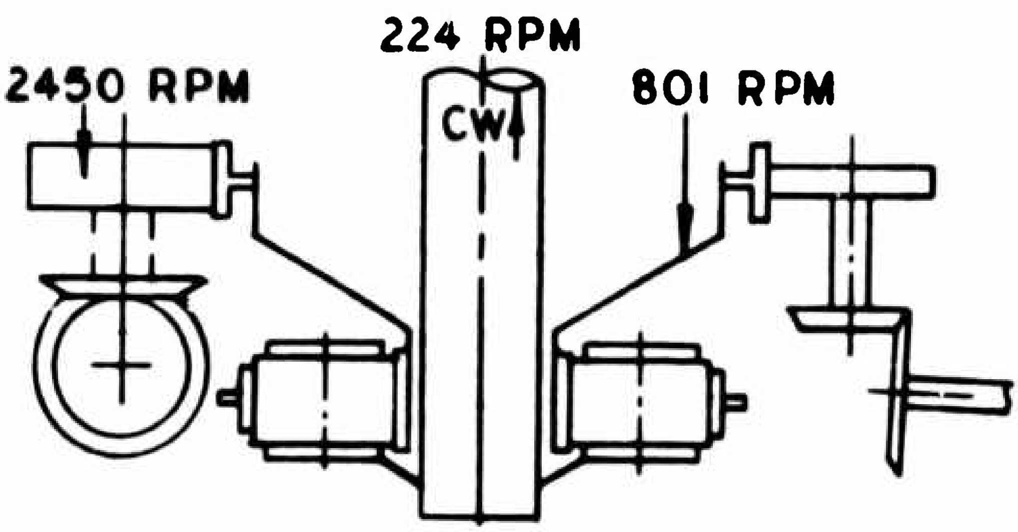


Figure 9. Gearing Schematic of Configuration Number 6.

TABLE X
GEAR DATA,
CONFIGURATION NUMBER 6

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Epicyclic Unit					
Sun	29,963	29	2.071	1.70	15,000
Pinion (3 req'd)	- 9,685	67	4.786	1.45	
Ring	- 2,631	163	11.643	1.20	
Carrier	2,292				
Bevel Pinion (Driven by Ring Gear)	2,631	27	7.60	1.50	5,240
Bevel Pinion (Driven by Carrier)	2,292	31	8.72	1.50	5,240
Bevel Gear (Output)	2,450	29	8.16	1.50	5,240
Spur Pinion	2,450	52	8.667	3.2	5,560
Spur Gear	801	159	26.50	3.0	5,560
Planetary (3.582 RR)					
Sun	801	67	8.375	4.28	1,270
Pinion (6 req'd)	730	53	6.625	4.03	
Ring	0	173	21.625	3.12	
Carrier	224				

TABLE XI
ALTERNATE ENGINE ARRANGEMENTS
OF ADVANCED TECHNOLOGY ENGINES

Configuration Number		Alternate Engine Location	Gearbox Modifications Required	Comments
1 & 2	(a)	Approx. horizontal & parallel to longitudinal axis of A/C but 10 to 12 inches fwd or aft of present location.	Fwd - acute shaft angle on 1st bevel mesh; aft - obtuse shaft angle on 1st bevel mesh.	Alternate engine locations give small weight variations due to changes in gearbox housing and length of drive shafts.
	(b)	Approx. horizontal but moved as in (a) above and fanned outward to obtain required engine clearance.	Same as (a) above.	Eliminate previous weight increase with an overall increase in width of engine installation.
3		Limited inclination in any plane and moved fwd or aft as desired.	Bevel gears can accept changes in vertical plane. Pinions meshing with combining spur gear can be moved closer together on back of gear box to compensate for fanned engine arrange - ment.	Input spur gears permit engines to be moved freely about input bevel pinion. Engines can be fanned outboard in horizontal plane to obtain adequate clearance.
4		In plan view, one engine installed to right & behind gearbox. Other engine reversed & installed fwd of gearbox. Engine inputs 180° apart.	In plan view, rotate fixed lay shafts approx 20° CW (or CCW if apposite engine arrangement is desired). Inputs would be similar to present configuration.	Low frontal area obtained. Engine ingestion problems could be minimized by placing inlets on one side of A/C & exhaust of fwd engine on opposite side.

TABLE XI-Continued

Configuration Number	Alternate Engine Location	Gearbox Modifications Required	Comments
5	Limited angular variation of engine centerline in vertical and horizontal planes	Change shaft angles of bevel gears. In plan view, min. clearance for fwd bevel gearing and rotor shaft determines max. angle engines can be fanned out.	Easily accommodated. No weight penalties. Engine clearance requirements dictate spacing of engines, but looking fwd, could be fanned inboard as desired to minimize frontal area.
6	Same as 5 above	Change shaft angles of bevel gears.	Engine slope in vertical plane affects split proportion of input power. Apex of bevel gear must be in same station plane as rotor shaft or slightly fwd to provide adequate clearance for gearing.

infrared suppression devices, air particle separators, mounting requirements, and the output drive shaft configuration. The first two of the above factors influence the position of the engine relative to the gearbox and airframe, but they have little direct bearing on the design of the drive train. The mounting arrangement and drive shaft configuration, however, both influence the design of the input section of the associated gearbox. It is not unusual, in fact, to incur a weight penalty of several pounds when adapting an existing engine to a particular helicopter arrangement; early liaison between airframe and engine manufacturers could, of course, minimize such weight penalties and mounting problems.

The interface and installation problems arising from the use of the 30,000-rpm and 15,000-rpm demonstrator engines are considered in the following paragraphs.

(1) Demonstrator Engine, 30,000-rpm Output

With this engine the suggested helicopter mounting includes a torque tube, concentric with the output drive shaft and connected to the front frame of the engine, and two additional mounts which are connected to the engine casing. The torque tube prevents engine motion in three directions, while the other mounts restrain the gearbox in one and two directions. Unless the engine-gearbox combination is mounted as one unit, a gimbal mount is usually provided on the forward end of the torque tube to connect the engine to the gearbox and to provide angular freedom in two directions while restraining the engine axially. This is the arrangement considered with these aircraft.

As the aft engine mounts move independently of the gearbox, resulting deflections and motion at the mount impose angular misalignment on the gimbal. This same angle must be accommodated by a coupling on the drive shaft, pivoting about the same axes as the gimbal. However, high thrust loads are also impressed upon the input drive shafting for varying periods of time due to thermal expansion in the engine; these forces cannot be imposed upon the coupling at the gimbal mount. To absorb these loads, therefore, thrust bearings must be installed prior to the coupling to transfer the loads into the gimbal mount. As can be seen in Figure 30, this requires an additional shaft and associated mounting hardware which imposes a weight penalty and added complexity upon the drive train.

(2) Demonstrator Engine, 15,000-rpm Output

The mounting arrangement suggested for this engine includes separate systems for both the engine and the gearbox. The engine can be restrained in three directions using the forward mount, while three attachments are made to the aft mounting ring. Thrust loads from thermal expansion and power fluctuations are primarily contained within the turbine and its mounting arrangement, and as such do not cause large deflections on the drive shaft and associated couplings. The engine and gearbox do not move in concert, however, and for this reason the drive shaft installation must absorb the resulting axial and angular motion across the two couplings. An engine gearbox torque tube would minimize this motion and facilitate the drive shaft coupling design.

The drive shaft engine interface is well defined with this engine as a flange is mounted on the turbine output drive shaft. Separate and completely enclosed lubrication systems are obtained in both the gearbox and the engine.

DESIGN EVALUATION - PURE HELICOPTER DRIVE TRAIN

Six different drive train configurations were presented in the preceding pages that satisfy the drive train requirements of the pure helicopter. The various facets of these designs must now be analyzed, evaluated, and compared to determine the drive train system recommended for installation in the pure helicopter.

Design effort has centered on the main gearbox, which must accept full engine power and distribute it to the main rotor, tail rotor drive system, and various accessories required for aircraft operation. The efficiency of the drive train is largely determined by the main gearbox since approximately 90 percent of input power is absorbed by the main rotor.

The tail rotor drive system dictates that a minimum of 2 bevel gear meshes must be used, regardless of the exact arrangement or reductions involved. The tail rotor power is relatively low, and as such the losses are correspondingly low. In addition, the final arrangement of the tail rotor drive system is determined more by the influence variations have upon the balance of the aircraft and system weight than by efficiency. System selection has thus centered upon the primary drive train, as more than 80 percent of the drive train weight is determined by the arrangement of this unit.

It follows that the highest efficiency will result from a transmission which includes a minimum number of gear meshes between the engine and the main rotor, although this tendency will be modified by the type of gear reduction that is considered. The complete elimination of a gear mesh, however, more than compensates for any minor variation in efficiency between the type of gear mesh; except in unusual cases, the highest efficiency can be expected from transmissions which minimize the number of meshes.

A further factor is that the number of gear meshes bears a direct relation to the overall weight of a transmission system. The type of mesh has an influence on weight, but, as in the case of efficiency, the complete elimination of a gear mesh, together with its associated bearings and support structure, produces a greater weight savings. The generally accepted adage that planetary reduction units, spur gear trains, and bevel gear trains involve progressively larger pounds per horsepower ratios, while generally true, must be applied with some caution to transmission systems involving multiple power inputs which are not in line with the output drive. In such cases it is often preferable to take a single large reduction on a combining spur gear as in Configuration Number 5 rather than take a smaller reduction on a combining gear and then add a planetary unit; the former case then gives higher efficiency with lower weight.

Reliability and maintainability were also major considerations in designing and evaluating the preceding drive train concepts. Each configuration was reviewed in detail during layout design, and changes were incorporated to reduce maintenance man-hours per flight hour and accomplish increased reliability in the drive train. In particular, a component analysis and a failure mode analysis were conducted to establish estimated removal failure rates and total failure rates of the gearboxes. Removal failure rate is defined as the rate of failure causing gearbox removal, while total failure rate is the rate of all failures including those requiring gearbox removal plus those not requiring removal. This latter item in cludes such failures as leakage of field replaceable seals, component wear or corrosion detected during (but not the cause of) overhaul, etc. The removal failure rate can be reduced by providing greater accessibility and more field replaceable components and by including fault detection equipment in various sections of the gearbox, thereby locating as well as indicating the malfunction. The data bank used during the preceding evaluations included that obtained from operational experience with the H-3 and H-53 series of production helicopters.

The use of layout designs and the nature of this phase of the program indicated that comparative data were more useful than absolute values, since the final system would be developed based upon the overall evaluation of the various designs. Thus, the removal failure rates and the total failure rates in Table XII are given relative to the baseline configuration.

To permit a comparative evaluation, Table XII presents pertinent data concerning the six pure helicopter main gearboxes. Further comments are presented in Table XIII, which summarizes some of the advantages and disadvantages of the various systems.

As shown in Table XII, the weight and efficiency of the various transmissions differ. To better compare the 6 transmissions, the gearbox weight plus the effect of gearbox losses on aircraft gross weight can be combined to determine the total equivalent weight of the transmission system.

Useful power applied to the rotor system determines the lifting capability of the rotor system while losses adversely affect aircraft performance, since they comprise lost energy which is converted to heat and must be rejected by a cooling system. In this aircraft with an alternate gross weight of 23,550 pounds and an average useful power of 3,875 horsepower, each horsepower lost due to inefficiency represents an aircraft weight loss of approximately 6 pounds. Figure 10 depicts the equivalent weights of the various transmission systems. These equivalent weights consist of the actual weight of the transmission system plus the potential rotor lift that is lost due to gearbox losses, and therefore provide a realistic basis for comparing the drive train configurations.

TABLE XII
DESIGN DATA,
PURE HELICOPTER TRANSMISSIONS

Configuration Number	Weight (lb)	lb per hp	Efficiency (percent)	Comparative Removal Failure Rate	Comparative Total Failure Rate
1	1460	0.365	96.7	1.00	5.22
2	1440	0.36	96.2	1.05	5.40
3	1470	0.367	97.0	1.03	5.28
4	1600	0.40	97.2	0.61	4.30
5	1330	0.342	97.5	0.74	5.17
6	1540	0.385	96.7	0.94	4.92

Note: The efficiency of the above transmission was determined by considering friction losses as follows:

1/2 percent for spur and bevel meshes

3/4 percent for an epicyclic reduction

3/4 percent for churning

As a further comparison, a growth version of a current production aircraft of equivalent horsepower rating has also been evaluated. However, due to rotor rpm, this gearbox supplies 10 percent more torque to the main rotor and therefore exhibits a higher weight than the nominal horsepower rating would imply. With an efficiency of 96.75 percent, the total effective weight of this transmission is also shown in Figure 10.

From Table XII and Figure 10, it can be seen that Configuration Number 5 is clearly superior to the other candidate systems, while Table XIII lists several inherent advantages of the gearbox arrangement. This new reduction concept conveniently accepts the 30,000-rpm input speed by the use of a counterrotating epicyclic unit. The use of three planet pinions assures equal load sharing, and conservative tooth stresses are obtained with a minor increase in weight. The torque in each bevel and spur gear mesh is accurately determined by the epicyclic unit itself. Thus, all gear meshes are subjected to known loading, permitting accurate sizing of all gears.

As previously noted, the data bank used in establishing the comparative reliability figures given in Table XII was obtained from operational experience with H-3 and H-53 helicopters. The use of such data in evaluating gearbox concepts such as Configurations 5 and 6 is extremely conservative, as the stress levels used in designing these new arrangements are markedly different from those used in designing these production aircraft. Although equal tooth loading is obtained in the input epicyclic units, the allowable compressive stress, which determines the face width of the

TABLE XIII
COMPARATIVE COMMENTS,
PURE HELICOPTER DRIVE TRAINS

Configuration Number	Advantages	Disadvantages
1	Components similar to those currently in use. A minimum development program required.	Two-stage output planetary affected by slope of rotor shaft and load sharing considerations; weight penalty incurred to maintain acceptable stress levels.
2	Most components similar to those currently in use.	2(a) Same as (1) above. 2(b) Input bevel pinion must be overhung. 2(c) Analysis indicates special materials must be used to accommodate speeds and loads imposed on input pinion bearings. 2(d) Freewheel unit requires development.
3	3(a) Engine location can be varied and yet remain parallel to longitudinal axis of aircraft. 3(b) Uses 1-stage output planetary. 3(c) No undue restriction on rotor shaft diameter.	
4	Large reduction ratio obtained in compound planetary. Planet pinion bearings carry no centrifugal load.	Planet carrier must withstand large twisting couple developed by compound pinions. Extensive test and development program required to develop proper tooth patterns.

TABLE XIII - continued

Configuration Number	Advantages	Disadvantages
5	5(a) In the epicyclic gearing, load sharing is assured by using 3 pinions. Lower tooth stresses are achieved by small weight penalty.	Planet pinion bearings are subjected to centrifugal loads.
	5(b) Freewheel units designed for one half engine power.	
	5(c) Redundant to last reduction stage.	
	5(d) Loads on all gearing are accurately established.	
6	6(a) Same as 5(a).	6(a) Output bevel gear must be overhung.
	6(b) Same as 5(b).	6(b) In comparison with Configuration Number 5, spur mesh transmits full engine power.

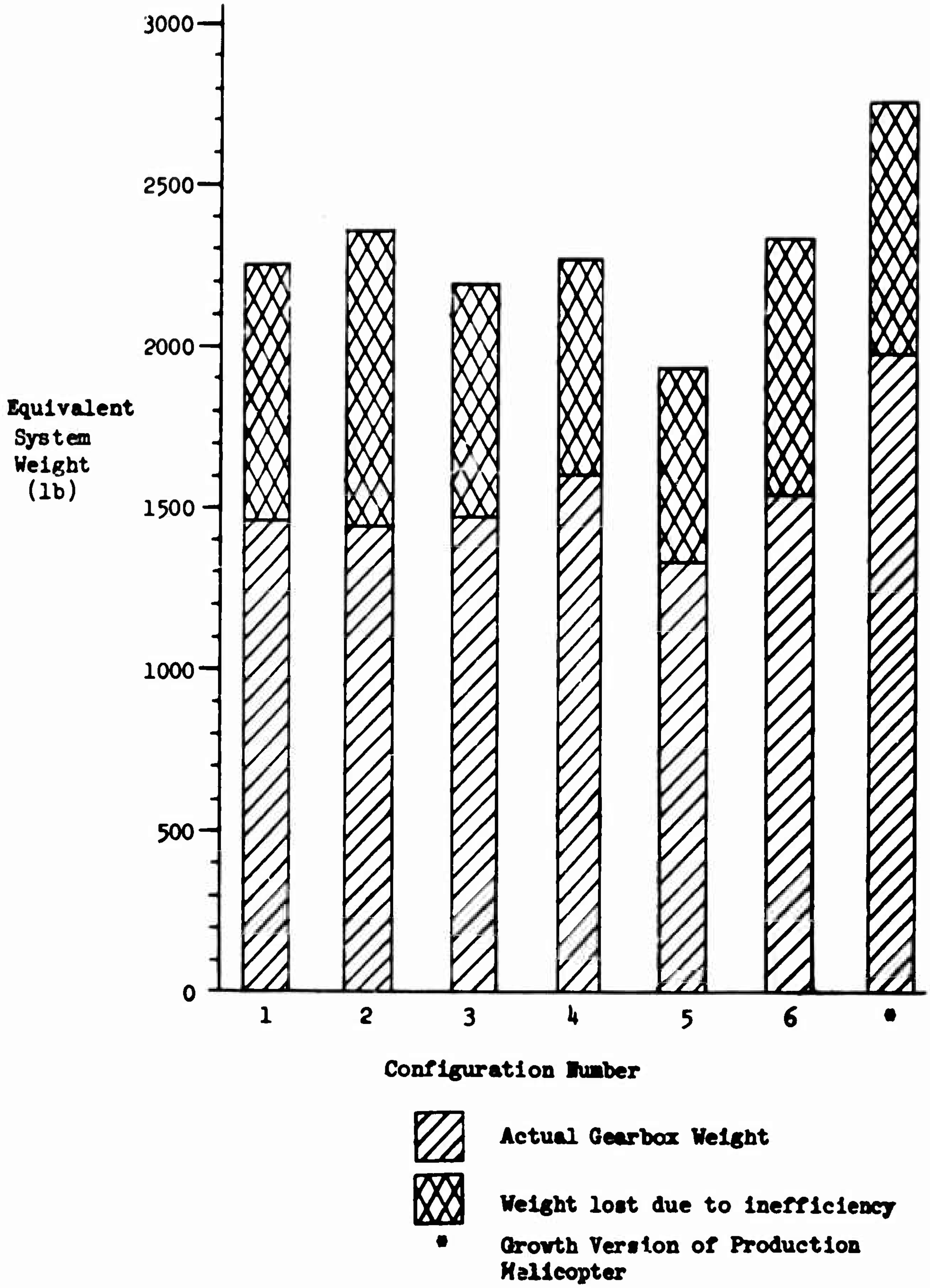


Figure 10. Weight Comparison, Equivalent System Weights.

gearing, was reduced to approximately 120,000 psi to assure acceptable operation at 30,000 rpm and to increase the reliability of the overall system. It seems probable that the decreased stress levels and predictable loading will yield an appreciable increase in reliability over current transmission drive trains.

Thus, Configuration Number 5 was selected for installation in the pure helicopter. Being least in weight and exhibiting several inherent advantages, further design effort to develop a pure helicopter drive train based on this system appeared warranted.

DETAILED DESCRIPTION OF SELECTED SYSTEM

Introduction

A comparative review of all the pure helicopter main transmission systems demonstrated that a new type of split-power transmission, with the rotor driven by a single combining gear instead of the usual planetary unit, had the greatest overall advantage.

The main features of the final transmission (shown in Figures 11 and 29, and developed from Configuration Number 5 (Figures 8 and 27)) can be summarized as follows:

(1) The transmission system can be fabricated to a weight of 0.33 pound per horsepower using stress levels not greater than those achieved in current production helicopters. This weight to power ratio is approximately 25 percent smaller than that achieved in current transmission systems.

(2) High efficiency is obtained (97.4 percent), since the loss sources are equivalent only to 1 counterrotating epicyclic unit, 1 spiral bevel gear mesh, and 1 spur (or helical) mesh, each carrying installed engine power. The reduction ratio obtained in the counterrotating epicyclic unit has eliminated one mesh in comparison with a conventional gearing arrangement.

(3) The freewheel for each engine can be sized for half engine power as a result of splitting power from each engine into two paths. Further, the freewheels are externally accessible for inspection and removal.

(4) Each engine drive train is independent of the other, or is redundant, as far as the spur gear mesh where power is combined to drive the main rotor.

(5) The drive paths from each engine are inherently load-balancing, so that problems of tooth loading arising from uneven load-sharing are avoided.

(6) A counterrotating epicyclic unit accepts high engine speeds (30,035 rpm) while keeping gear pitch line velocities down to

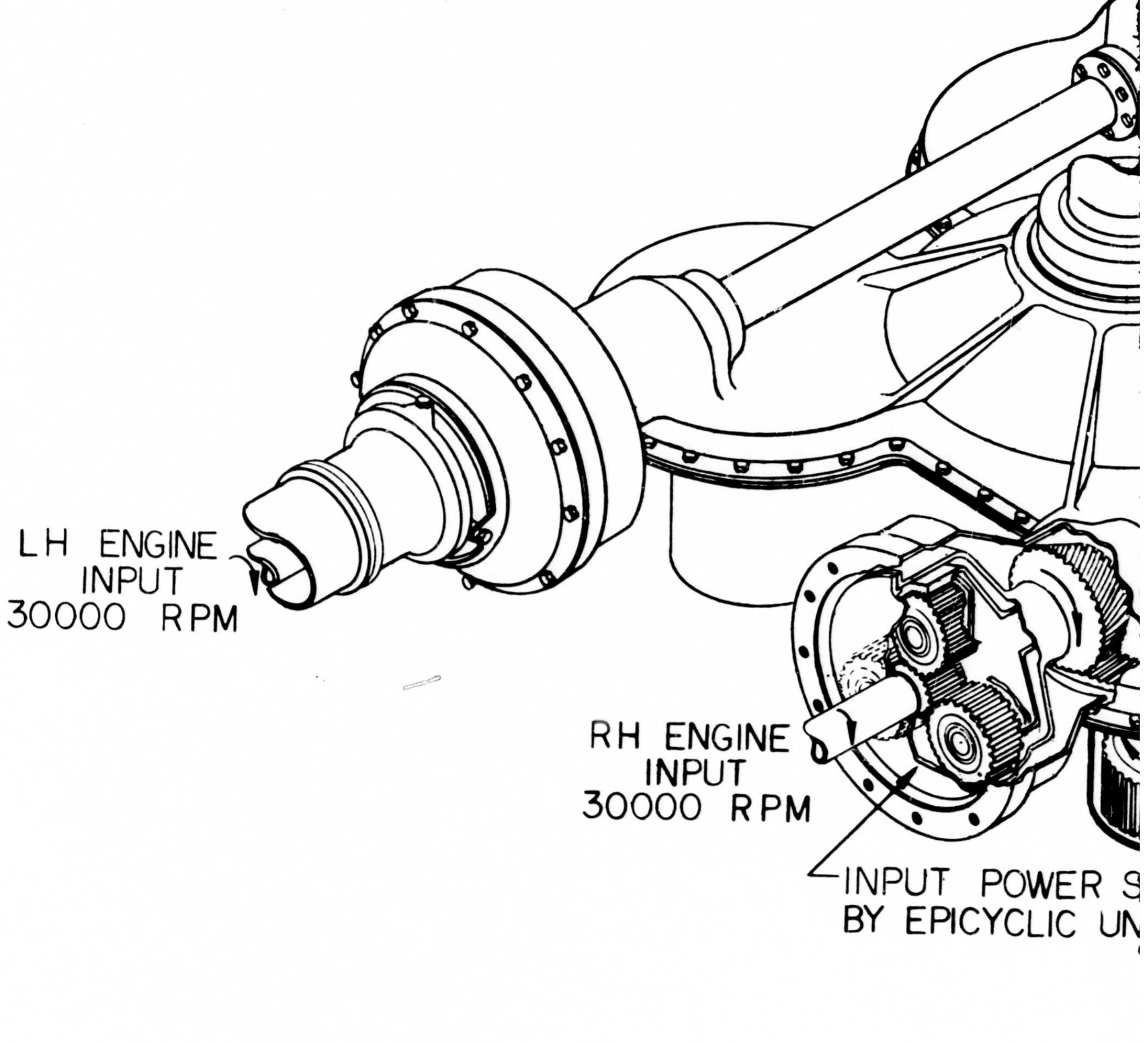
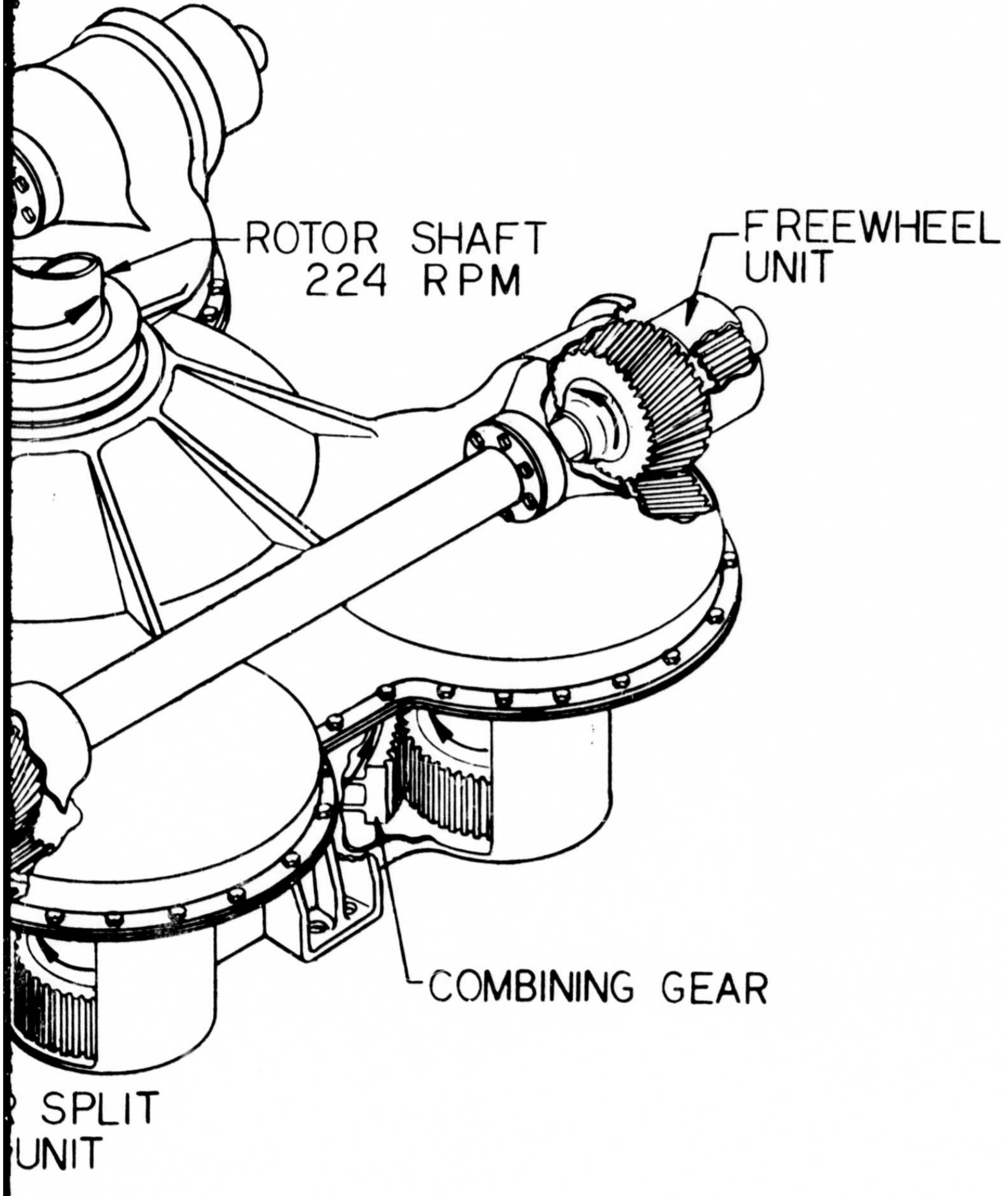


Figure 11. Main Transmission Arrangement, Pure Helicopter.

A


ROTOR SHAFT
224 RPM
FREEWHEEL
UNIT
COMBINING GEAR
SPLIT
UNIT

B

15,000 fpm. In this way the difficult bearing problems associated with bevel gears operating at high rotational speeds are avoided.

(7) As a result of counterrotation, the input epicyclic unit splits the engine power into two paths, at the same time achieving reduction ratios of 11.07 and 12.83 to the ring gear and the planet carrier respectively. These speed reductions ensure that the planet pinion bearings are not unduly affected by centrifugal loads.

(8) The input epicyclic units, which are field replaceable, are designed with floating sun and ring gears and only 3 planet pinions in order to achieve equalization of gear tooth loads.

Main Gearbox

Arrangement

The final arrangement of this transmission is shown in Figures 29 and 30, from which it is seen that each engine drives the sun gear of a counterrotating epicyclic unit. It is most convenient to place a bevel gear mesh after the counterrotating planetary unit for, by meshing the 2 input spiral bevel pinions on opposite sides of their respective output bevel gears as shown in Figure 29, the transmission is converted to a single-rotation system while achieving the angle turn necessitated by horizontal engines and a vertical rotor shaft. In this way each of the two output drive shafts from the epicyclic unit carry one-half engine power, with the result that the spiral bevel gears and the combining spur gears can be sized for one-half engine power.

Several alternate transmission configurations can be devised which incorporate the principles outlined above. For instance, Figure 27 shows internal, or back-to-back, mounting of the spiral bevel pinions with the epicyclic unit and the freewheel unit between the bevel pinions. This arrangement results in a compact transmission, but bevel gear assembly and bearing support problems then are present, together with the need for a long, high-speed, input drive shaft to the sun gear.

As a result of these disadvantages, externally mounted bevel pinions were used in the final design scheme as shown in Figure 30. External mounting of the bevel pinions also allows easy adjustment of the bevel mesh and external access to each freewheel unit, which is an important feature for maintenance. Figure 29 also illustrates how the initial offset of the engines from the center line of the helicopter is utilized to space the 4 spiral bevel sets around the gear driving the rotor to essentially balance the radial loads on the large-diameter spur gear. Gear and bearing data are summarized in Appendix II.

Power Distribution From the Input Epicyclic Unit

Each engine drives the sun gear of an epicyclic gear, and the directions of rotation of the ring gear and planet carrier must be carefully chosen to avoid any recirculation of input power. It is shown in Appendix III that this condition is achieved when the planet carrier rotates in the same direction as the sun gear and the ring gear rotates in an opposite direction to the sun gear. In general, it is desirable that each drive path carry one-half the engine power, for then the combining spur gears driving the rotor can be of uniform face width. The relation between the speeds of epicyclic members necessary to achieve any required proportioning of the two power flows is explained in detail in Appendix III.

The transmission arrangement shown in Figure 30 was adopted as a result of preliminary design studies, and minor design modifications were then undertaken in order to determine what proportion of the overall 134 to 1 speed reduction should be taken at each gear mesh. The main factors influencing the choice of gear ratios are:

(a) The final speed reduction is limited by the acceptable diameter of the combining gear.

(b) Speed reductions from the input epicyclic unit must be large enough to keep the centrifugal load of the planet pinions within acceptable limits. Too large a speed reduction at the epicyclic unit, however, leads to a weight penalty from the ensuing bevel gears.

A reduction ratio of 5.36 was selected for the combining spur gear mesh, which with a main rotor speed of 223.67 rpm gives a speed of 1199 rpm on each of the four pinions. The desirability of transmitting the same power through each of the spur gear pinions means that equations (2), (13), and (14) of Appendix III must be satisfied. Since all gears must use an integer number of teeth, however, an exact solution to the equations noted is rarely possible, but close approximations can easily be obtained.

A ring-gear-to-sun-gear ratio, R/S, of 137/25 proved convenient when used in conjunction with the bevel gear sizes given in Table XXXI, these sizes resulting in rotational speeds of -2713 rpm and 2341 rpm for the ring gear and the planet carrier, respectively. Substitution of these numerical values in equation (2) of Appendix III shows the speed of the sun gear, and hence that of the engine, to be 30,035 rpm. Equations (5) and (6) of Appendix III then give

$$\frac{P_c}{P_s} = 0.505 \text{ and } \frac{P_r}{P_s} = 0.495.$$

The rated input power from the engine to the sun gear, p_s, is 2000 horsepower; hence, the above equations show that the planet carrier and the ring gear supply 1010 horsepower and 990 horsepower respectively to the main rotor. The error from an equal division of input power is thus only 1 percent, and the error between actual turbine speed and desired turbine speed is only 0.12 percent.

It is seen that the design involves a relatively large speed reduction at the input epicyclic unit, this being a principal advantage contributed by counterrotation of the epicyclic output members. In order to achieve equal load sharing, and on acount of high input speeds, only 3 planet pinions are used in the epicyclic unit. Details of the gear stresses, pitch line velocities, and bearings are contained in Table XXXII. Conservative gear tooth compressive stresses (120,000 psi) have been used in the epicyclic unit. On account of the low weight of each unit, these allowable stress levels involve a weight penalty of only 3 pounds for each epicyclic unit.

Rotation of the planet carrier means that additional loads are imposed on the planet bearings as a result of the centrifugal action of the planet gears. If the ring gear of the epicyclic unit were fixed, which is usually the case, the planet carrier would rotate at 4630 rpm with a sun gear speed of 30,000 rpm. In this case the total load on each planet bearing would be 3.3 times the prorated bearing load due solely to tooth forces, and this high additional load could not be accommodated with a conventional planetary design using roller bearings. As a result of counterrotation of the ring gear, however, the planet carrier of Figure 30 rotates at 2341 rpm, giving a total load on the planet gear bearings only 1.3 times the prorated bearing load which would exist with no rotation of the planet carrier. It will be realized, of course, that the centrifugal load component is constant and cannot be prorated. It follows that designs incorporating counterrotation of the ring gear become progressively more attractive when centrifugal loads in fixed ring gear designs necessitate planet pinion bearings of unduly high capacity.

Mounting of Epicyclic Unit

The epicyclic unit can be placed inside the transmission housing and between the counterrotating bevel gears, as shown in Figure 27. This arrangement results in a compact design, but accessibility is then limited and a rather long, high-speed input shaft must pass through the bevel gear nearest the engine. These disadvantages can be most conveniently avoided by placing the epicyclic unit immediately downstream of the engine, as shown in Figure 30. In this way the high-speed drive to the sun gear is as short as possible and the complete epicyclic unit can be removed intact from the transmission housing for inspection or replacement should it be required.

It will be noted from Figure 30 that care has been taken to ensure equal tooth loading in the epicyclic gear by floating both the ring gear and the sun gear. Then the sun gear and the ring gear can take up their natural positions around the planet pinions as dictated by the tooth forces. This arrangement, together with the use of only 3 planet pinions, assures equal load sharing in all the tooth meshes of the epicyclic unit. Further, the only bearings subjected to engine rpm are those which position the engine drive shaft in the gimbal mount, as shown in Figure 30.

Conventional directions of rotation for the engine and the main rotor necessitate that the ring gear and the planet carrier be connected to the spiral bevel sets, as shown in Figure 30. These directions of rotation dictate that the drive from the planet carrier must pass over the ring gear, resulting in a small weight penalty. From this point of view it would be an advantage to have the opposite direction of rotation on the input shaft.

Combining Gear With Helical Teeth

The four spur gear pinions meshing with the large-diameter combining gear can be spaced so that the resultant radial load on the rotor shaft is very low. Uneven loading would still occur to some extent, however, if one engine were shut down. But if these pinions had helical teeth, the axial thrusts of the pinion and the spiral bevel gear could be opposed, thus easing the loads on the thrust bearings adjacent to the spiral bevel mesh (see Figure 30). An important additional feature, however, is that the resulting thrust on the combining gear will act downwards, thereby relieving the rotor bearings from the total lift imposed by the main rotor. Then the component of rotor lift carried by the helical thrust of the combining gear is distributed to the transmission casing in the four regions above the spiral bevel mesh points. If this unusual effect is exploited, the rotor bearings can be relieved of any desired proportion of the lift load imposed on the rotor. In addition, these bearings must still be sized to accommodate the loads on the rotor shaft for the number of cycles amassed during autorotation or single-engine operation.

Torque Balance and Tooth Loading

A well-known property of an epicyclic gear is that the torque distribution between the sun gear, the planet carrier, and the ring gear is unaffected by the rotational speed of each member. It follows that with a given input torque to the sun gear of Figure 30, the torques carried by the bevel gear meshes of Figure 30 are known. The input epicyclic unit thus functions as a torque-splitting device, ensuring that each bevel mesh and combining gear mesh is subjected only to its designed torque level. At the same time the input epicyclic unit will absorb internally, by small relative motions of the planet carrier and the ring gear, any effects arising from eccentric mounting of gears, structural deflections, or drive lines of differing stiffness.

Previous investigations of helicopter transmission systems (Reference 5) have stressed the importance of equal load sharing between gear tooth meshes as a means of obtaining increased power/weight ratios.

No quantitative results have been quoted for the variation of tooth loading which exists, for instance, in planetary units with a large number of planet pinions, but it is evident that equalization of tooth loads would allow some increase in gear tooth capacity and bearing capacity.

Load sharing such as is achieved in this transmission means that all of the gear meshes in the main rotor drive train will operate at predictable tooth loads. The nominal spur gear stress levels designed into the transmission, however, are those which have proven to be acceptable in production helicopter transmissions. A conservative estimate is that, as a result of the inherent load-sharing effect, a 10-percent increase in compressive stress or a 20-percent increase in torque capacity can be accommodated without the true stresses rising above those experienced in current production transmission systems which involve planetary units with a large number of pinions. This potential gain in power/weight ratio is an additional attraction of this transmission and can provide either for engine growth or for increased life and reliability.

Rotor Shaft Design

With a conventional helicopter transmission system such as is shown in Figure 23, some design problems are experienced if the rotor shaft passes through the planetary gear units. These difficulties arise from the conflicting requirements of using a small-diameter sun gear to obtain a high reduction ratio and requiring a large-diameter rotor shaft for stiffness. Inadequate diameter of a rotor shaft means that, although the head moment and transmitted torque can be contained from a stress viewpoint, excessive deflection and slope of the rotor shaft will prevent acceptable bearing and gear installations.

In the design under discussion (Figure 30), there are no restrictions on rotor shaft diameter, so it becomes possible to incorporate a rotor shaft which is flared out at the lower end. In this way a great increase in stiffness of the rotor shaft is obtained while additional, and protected, space is made available inside the rotor shaft for an oil cooler and blower, or for internal controls. Such a rotor shaft does not necessarily involve a weight penalty, since the diameter can be increased to the machining limit associated with thin wall sections. In this case the material is fairly uniformly stressed and, therefore, is used effectively.

A further factor in the design shown in Figure 30 is that the rotor shaft slope at the attachment of the combining gear must be sufficiently small to avoid uneven tooth loads on the spur gear meshes. The rotor shaft proportions shown in Figure 12 give slopes of 0.0008 and 0.0003 inch per inch at the upper and lower bearings respectively. Thus, the slopes through the rotor bearings are negligible while the maximum tooth slope across the combining gear mesh, taking into account any change in position of head moment, will always be less than 0.00024 inch per inch of face width. The critical shaft section is

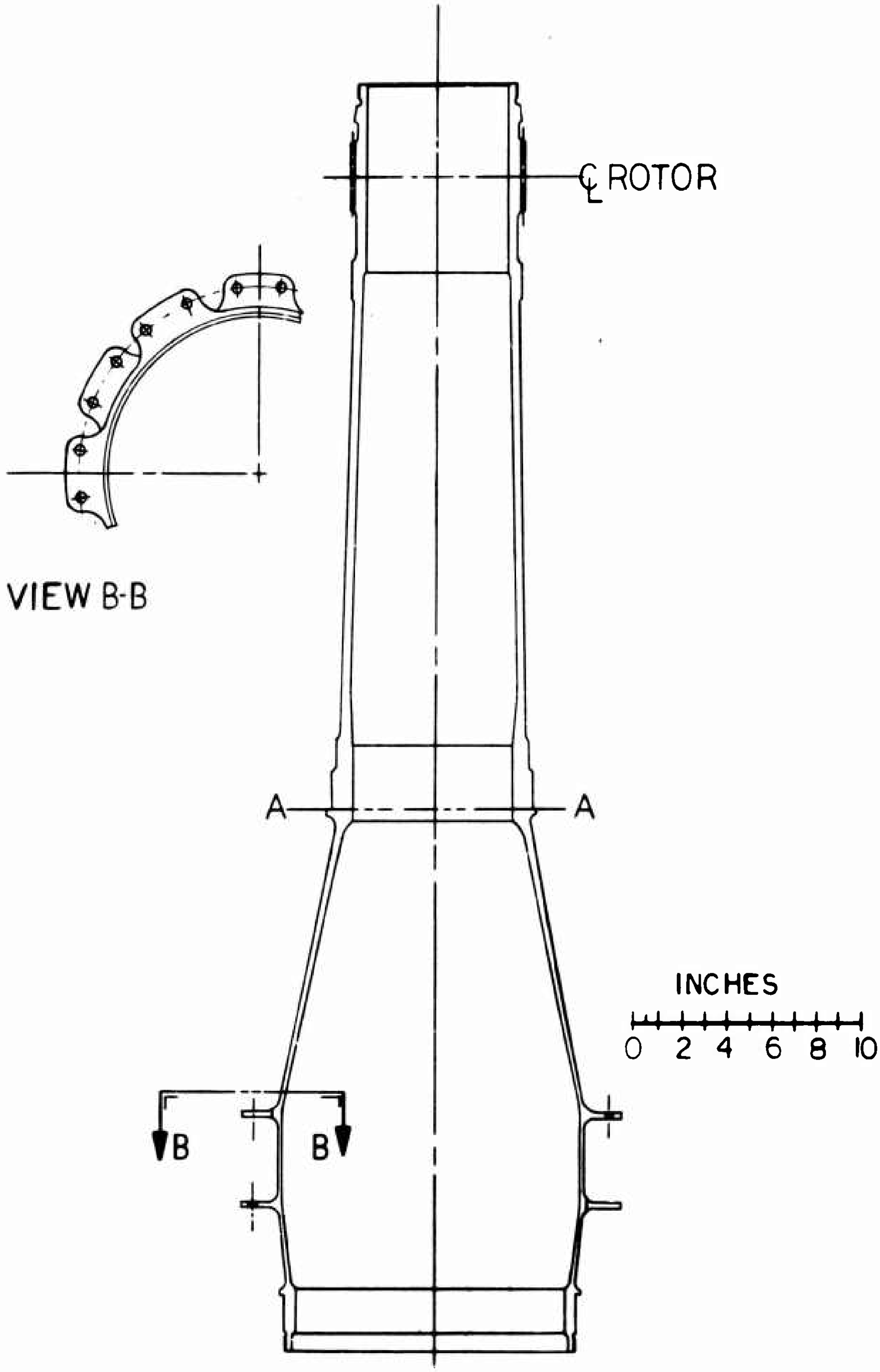


Figure 12. Main Rotor Shaft.

at the upper bearing and is depicted in Figure 12 as Section AA. The bending stress of 9800 psi is within acceptable limits for such locations. Figure 12 also depicts the flange arrangement that allows the rotor shaft and combining spur gear to be assembled. Two rows of bolts are utilized to attach the double flanged spur gear to the rotor shaft. Each of the flanges that is electron beam welded to the spur gear rim is capable of transmitting the total torque to the main rotor shaft.

Epicyclic Gear Reductions

The epicyclic reduction stages used throughout this design study involve separate planet carrier plates connected by clamped spacers and the preload journal of the planet pinion bearings. In these symmetrical double plate configurations, operational experience has indicated that adequate plate thickness is determined by either of two design limitations, maximum allowable bending stress or maximum allowable plate deflections. The torsional stiffness of the planet carrier assembly thus depends on the combined resistance of the plate spacers, the preload journal, the bearing inner races, and the plates themselves. Torsional deflection, arising from the output torque being taken from only one plate, is limited to an amount which results in a tooth misalignment of less than 0.0006 inch per inch. Suitable corrective helix angle modifications, together with crowning and tip relief, are then made in the light of past experience. The above design practices have produced reliable, efficient, and lightweight designs currently in use in several production helicopters.

The epicyclic units which accept engine speed have integral oil transfer passageways drilled in the carrier plates; oil is then fed into the various planetary plate spacers to the planet pinion bearings and the gear tooth meshes. Titanium planet carrier plates and spacers were used in all the transmission systems considered since significant weight savings can be realized from the use of this material.

Freewheel Overrun Speed

The freewheel in the drive train of each engine must transmit the 990 horsepower delivered by the rotating ring gear at a speed of 2713 rpm, as shown in Table IX. On shutdown of an engine, however, the sun gear of the epicyclic unit is brought to rest and, with the main rotor still maintaining its normal speed, the freewheel will experience a speed difference of sS/R rpm across its input and output members as explained in Appendix III. It follows, using $s = 30{,}035$ rpm and $S/R = 25/137$, that the freewheel overrun speed is 5481 rpm.

Efficiency of Transmission

Losses in a transmission system can be estimated from a knowledge of the type of gear mesh, the design horsepower, and the power transmitted by each gear mesh. Past experience, backed by experimental confirmation, has demonstrated that a conservative estimate of tooth mesh losses, together with bearing losses, is 1/2 percent for spur gears, 1/2 percent for bevel gears, and 3/4 percent for planetary units. A further 3/4 percent of power transmitted must then be added to account for the miscellaneous churning losses.

The losses in the transmission are calculated as follows.

Loss Sources and Efficiency of Transmission

Main Rotor Drive - 3400 hp Output

Gear Mesh	Loss Percent	Loss hp
1 epicyclic	0.75	25.5
1 bevel	0.5	17.1
1 spur	0.5	17.2
churning	0.75	25.5
	Main Rotor Drive Losses	85.3

Tail Rotor Drive - 320 hp Output

Gear Mesh	Loss Percent	Loss hp
1 epicyclic	0.75	2.4
2 spur	1.0	3.2
2 bevel	1.0	3.3
churning	0.75	2.4
	Tail Rotor Drive Losses	11.3

Accessory Drive - 123 hp

Gear Mesh	Loss Percent	Loss hp
1 epicyclic	0.75	0.92
2 bevel	1.0	1.23
4 spur	2.0	2.46
churning	0.75	0.92
	Accessory Drive Losses	5.53

Input Power Required

Main Rotor Drive	3400	+	85.3	=	3485.3
Tail Rotor Drive	320	+	11.3	=	331.3
Accessory Drive	123	+	5.53	=	128.53
	3843	+	102.13	=	3945.13

$$\text{Transmission Efficiency} = \left(1 - \frac{102.13}{3945.13}\right) 100 = 97.4 \text{ percent}$$

Heat Generated in Main Gearbox

The above gear tooth losses and bearing losses appear as heat, and the oil supply to the gearbox must be adequate to remove this heat while keeping the oil temperature rise within a specified limit. The total heat generated is given by

$$Q_G = 2545\ F_{hp}$$

$$\text{where } F_{hp} = 102.13 \text{ hp}$$

$$Q_G = 261{,}000 \text{ Btu/hr}$$

With MIL-L-7808 oil, an inlet temperature of 176°F, and an outlet temperature of 230°F, the required oil flow is given by:

$$W_o = \frac{C_e\ (42.4)\ F_{hp}}{0.1337\ C_p\ \rho\ \Delta T}$$

where the factor $C_e = 0.60$ is an experience factor obtained from measurements on production main gearboxes.

$$W_o = \frac{0.60\ (42.4)\ (102.13)}{0.1337\ (.528)(54.4)(54)}$$

$$= 12.5 \text{ gpm}$$

Two oil pumps have been selected for installation in this gearbox, each with a capacity of 12 gpm. These pumps rotate at 3820 rpm, being driven by the spur pinion shaft as shown in Figure 30. Two pumps provide redundancy and adequate cooling even if one pump should experience a malfunction. The oil flow noted above necessitates an oil sump capacity of 8 gallons.

An advantage of the large-diameter rotor shaft is that an oil cooler and an air blower can be mounted inside the rotor shaft. The heat exchanger is then protected from small-arms fire; in contrast to current operational transmissions, the rotor shaft does not restrict the size of the heat exchanger. The heat exchanger is required to cool the gearbox oil from a hot oil temperature of 230°F with a hot day ambient temperature of 104°F. With an effectivity factor of 0.9, the actual temperature drop through the heat exchanger is

$$\Delta T = 0.9\ (230°F - 104°F)$$

$$= 114°F$$

The mass flow rate of air is mass flow rate = $\frac{\text{heat rejection rate}}{C_p \quad \Delta T}$

The heat rejection rate of the heat exchanger is taken to be 85 percent of the total heat generated, Q_G, the remaining 15 percent being conducted through the gearbox housing. It follows that mass flow rate of air

$$W_a = \frac{0.85\ (261{,}000)}{60\ (.241)(114)}$$

$$= 135 \text{ lb/min}$$

The mean air velocity, V_m, then is

$$V_m = \frac{\text{mass flow rate}}{(\text{flow area})\ (\text{density})}$$

With a density of 0.057 lb/ft^3 and a minimum internal diameter in the rotor shaft of 6.2 inches,

$$V_m = \frac{135}{(.21)(.057)(60)}$$

$$= 188 \text{ ft/sec}$$

This transmission design thus permits the oil cooler to be located within the main rotor shaft while keeping air velocities in the rotor shaft at a level which will avoid flow disturbance problems. A schematic of the lubrication system is shown in Figure 13.

Tail Rotor Drive System

Design Consideration

The main and tail rotors must be interconnected by shafting and the associated gearboxes to provide speed synchronization and permit transfer of power that is required during various flight regimes, including autorotation. The shafting must be designed to transmit the required power, accommodate the imposed deflections, provide an appropriate operational critical speed relationship, be lightweight, and yet permit fabrication by available manufacturing technology.

The tail drive system proposed for this aircraft includes an intermediate gearbox at the base of the tail rotor pylon, a tail rotor gearbox, and the

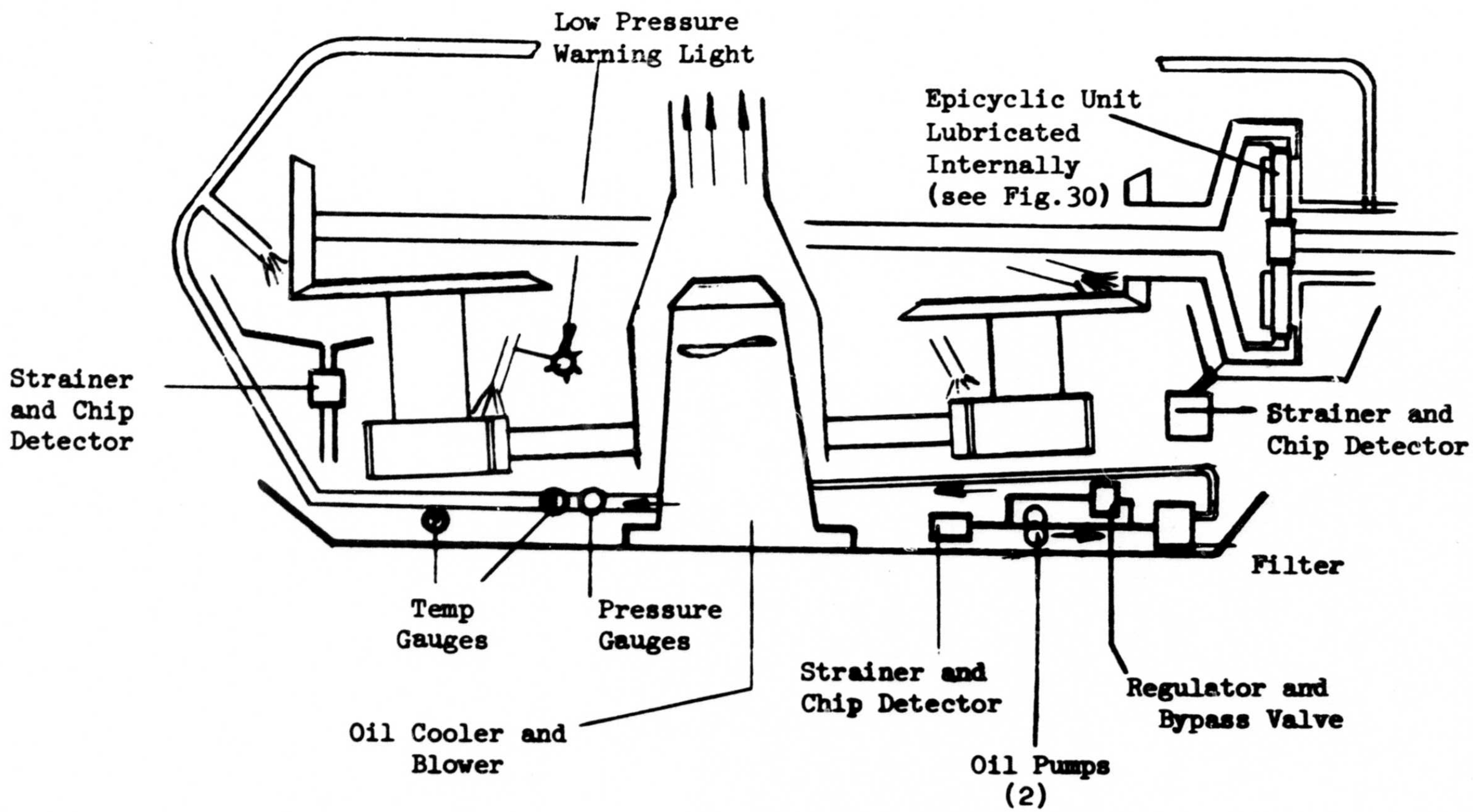


Figure 13. Lubrication System, Main Gearbox.

TABLE XIV
WEIGHT SUMMARY,
MAIN GEARBOX

Item	Assembly Weight (lb)
Input section including epicyclic unit and bevel pinion assembly (incl. 11 lb gimbal)	114
Input section including epicyclic unit and bevel pinion assembly (incl. 11 lb gimbal)	114
Bevel pinion, shaft, and freewheel unit assembly (2 required)	84
Bevel gear spur pinion assembly (2 required)	172
Bevel gear spur pinion assembly (2 required)	178
Main rotor shaft, combining spur gear and bearings	329
Tail takeoff assembly	49
Accessory takeoff assembly	32
Main housing, liner, and stud assembly	238
Sump housing and oil pump assembly	20
	1330 TOTAL

connecting drive shafting. Various operational speeds from 1100 rpm to 5000 rpm were considered in an effort to develop a minimum weight system. Gearing and shafting were first determined for a 3000-rpm system with a small speed reduction taken at the intermediate gearbox and the remainder taken in the tail rotor gearbox.

In comparison, an 1100-rpm drive shaft eliminated the need for any speed reduction but increased the size of the required gearing and gearboxes to accommodate the transmitted torque. At 1800 rpm, the required 1.72 to 1 speed reduction is easily accomplished in the tail rotor gearbox. However, both of the above systems result in weight increases in comparison with the 3000-rpm system, due to the increased torque being transmitted throughout the system. A 5000-rpm shaft installation reduced the size of the intermediate gearbox but resulted in a larger diameter output bevel gear in the tail rotor gearbox than the 3000-rpm system required. Critical speed considerations also dictated the use of a larger diameter drive shaft than the 3000-rpm system. Therefore, the advantages of transmitting a lower torque were negated by the above design restraints. Thus, a 3000-rpm drive shaft installation was selected for this aircraft, as shown in Figure 31.

Redundancy is important in any drive shaft system where the loss of control could result in loss of the aircraft itself. Because a redundant parallel shaft system is not feasible, increased reliability in shafting design is achieved by judicious selection of design allowables. However, coupling design permits consideration of a large number of possible coupling configurations that satisfy the basic operating requirements. In the evaluation of helicopter drive shafting systems Sikorsky Aircraft has investigated and tested various types of couplings including straight and crowned spline, flexible disk, diaphragm, and vane type elastomeric couplings.

The most desirable coupling configuration tested to date has been the laminated flexible disk type coupling; this type coupling has been used in this application. Disk packs of several stainless steel laminations permit high torque capacity while accommodating the imposed angular misalignment. An important feature of the laminated disk type coupling is that the integrity of the drive system is not affected by the fracture of individual laminations. This has been proven by qualification testing and actual service experience which has demonstrated that a fractured lamination does not affect the structural reliability of the coupling. These couplings also permit maintenance-free coupling installations since no lubrication is required.

Description

The main gearbox incorporates a tail takeoff that is driven by the combining spur gear and provides an output speed of 3016 rpm. The proposed tail drive shafting between the main and intermediate gearboxes consists of 5 similar sections of 2024-T3 aluminum alloy tubing with 4340 steel fittings riveted to both ends. This 2.75-O.D. by 2.55-I.D. drive shafting is supported on antifriction ball bearings soft-mounted on viscous damped supports as shown in Figure 31 and is connected by means of flexible disk couplings. Each flexible disk coupling consists of 17 stainless steel laminations, 6.20-O.D. by 4.75-I.D., with disk pack thickness of 0.208 inch.

The intermediate gearbox, shown in Figure 32, accomplishes the required angular change by using a 1.09 to 1 ratio set of spiral bevel gears.

Since the distance between the intermediate and tail gearboxes is 80 inches, one section of shafting is used, eliminating the need for a center support. This 3.3-O.D. by 3.1-I.D. 2024-T3 aluminum shafting has identical flanges riveted to both ends. A flexible disk coupling at either end of the drive shaft completes the installation.

The tail rotor gearbox arrangement, shown in Figure 33, includes a right-angle set of bevel gears to accomplish the required direction change and speed reduction. The tail rotor is driven by the output shaft, and all rotor loads are transferred to the airframe through the gearbox housing.

The efficiency of both the intermediate and tail rotor gearboxes is approximately 99 percent. Based on extensive production experience with similar gearboxes, splash lubrication provides adequate lubrication and cooling for these gearboxes. The intermediate and tail rotor gearboxes will operate satisfactorily with 0.2 and 0.4 gallon of oil, respectively. All bearings have unfactored lives in excess of 3000 hours. Gearing data on these gearboxes are given in Table XXXI. Weights are given in Table XV.

TABLE XV
TAIL ROTOR DRIVE SYSTEM WEIGHT

Component	Weight-lb
Shafting - Main Gearbox to Intermediate Gearbox	63.1
Intermediate Gearbox	41.2
Shafting - Intermediate to Tail Rotor Gearbox	11.3
Tail Rotor Gearbox	94.0
Total	209.6

Accessory Gearbox

Description

The accessory drives required for aircraft operation are listed in Table XLIII. A separate accessory gearbox has been included in the drive train configuration to provide increased component accessibility and to facilitate maintenance. This accessory gearbox is located forward of the main gearbox as shown in Figure 1 and is driven either by an auxiliary power unit (APU) during ground operation or by the drive train during normal aircraft operation.

The arrangement and location of the accessory drives have been developed considering both safety of flight, in the event of accessory gearbox failure, and the power available during APU operation. The accessory gearbox drive train permits one generator, a servo pump, a utility pump, and a lubrication pump to be driven during APU operation. In addition to the preceding accessories, the tachometer and a second generator, both mounted on the accessory gearbox, are driven by the primary drive train during normal aircraft operation. The primarv servo pump is mounted directly on the main gearbox to preclude total loss of servo pressure in the event of failure of the accessory gearbox input drive shaft.

The accessory gearbox is shown in Figure 34, where 3 section views explain the arrangement of the internal gearing. All bearings have unfactored lives in excess of 3,000 hours. The gearing data are given in Table XXXII. Total weight of gearbox and input drive shaft is 60 pounds.

Lubrication and Efficiency Analysis

The lubrication system must provide cooling oil to remove heat generated due to friction losses at gear meshes and bearings and to provide an oil film to support gearing and bearing loads. Experience with similar gearboxes indicates that the losses in the gearbox can be determined by considering 1/2 percent per mesh with an additional 1/2 percent considered for gearbox churning losses. In addition, the no-load gearing losses are approxmiately equal to 70 percent of full load losses. The losses in the gearbox, maximum when the APU is operating, are estimated as follows.

APU Input Mesh	=	(.005)(80)	=	0.40
Servo-Lub Pump Mesh	=	(.005)(56)	=	0.28
Lub Pump-Generator Mesh	=	(.005)(54)	=	0.27
Servo-Input Shaft Mesh	=	(.005)(.7)(67)	=	0.24
Input Shaft-Utility Mesh	=	(.005)(11)	=	0.06
Churning Losses	=	(.005)(80)	=	0.40
	F_{hp}	(friction hp)		1.65

Estimated Efficiency $= (1 - 1.65/80)100 = 98\%$

Total Heat Generated (Q_G)

$$Q_G = (2545)\ 1.65$$
$$= 4200 \text{ Btu/hr}$$

The friction horsepower and generated heat are low in this accessory gearbox, while the surface area is relatively large. Based upon operational experience with similar gearboxes, the friction heat can be dissipated by the normal flow of oil throughout the gearbox as it is circulated via an oil pump. Based upon the use of MIL-L-7808 oil, an inlet temperature of 176°F, and an outlet temperature of 230°F, the required oil flow is determined as follows:

$$W_o = \frac{C_e\ (42.4)\ F_{hp}}{(.1337)\ C_p\ \rho\ \Delta T}\ \text{gpm}$$

$$= \frac{(1)\ (42.4)\ (1.65)}{(.1337)\ (.528)\ (54.4)\ 54}$$

$$= 0.34 \text{ gpm}$$

Based on the above requirement and considering the several gear meshes to be lubricated in this gearbox, a 1-gpm vane-type oil pump operating at 5130 rpm has been selected for this application. Adequate lubrication is obtained by providing 0.3 gallon of oil in the gearbox.

Reliability and Maintainability Analysis

The selected system was analyzed to determine the estimated reliability and maintainability of the various components that comprise the entire drive train. Major components were analyzed considering the effect of each individual part. The reliability of the transmission system components was determined by comparing the drive train components to similar production components with known service experience. Based upon extensive operational experience, the following parameters were predicted:

(1) Removal failure rate of each dynamic component.

(2) Total part failure rate to the subsystem level.

(3) Unscheduled maintenance man-hours per flight hour (MMH/FH) to the subsystem level.

The above data are given in Table XVII.

It should be noted that safety of flight is enhanced by the use of redundant drive paths. In the main gearbox used in this drive train, each engine supplies power to the main rotor via separate drive paths that combine on the final spur gear which is mounted on the main rotor shaft. Thus, single-engine operation will be possible after a failure in the drive path of the other engine. Oil strainers and chip detectors isolate any contamination in various sections of the gearbox and locate the source of any malfunction. The reduced tooth stress levels and the design of the epicyclic units should also contribute to greater reliability than even that predicted by the use of data from current production aircraft, where five or more planet pinions are used in each planetary reduction stage. Thus, the predicted main gearbox failure rates given in Table XVII appear conservative, although no experience on similar components is available upon which to make more accurate estimates. Maintainability is simplified by an increased number of field replaceable components and parts. For example, both freewheel units, all external shaft seals, and both input sections, including the input shafts, the epicyclic gearing and the associated bearings, are field replaceable without affecting the gearbox-rotor head control system installation.

Data on the tail rotor drive system are probably very realistic, since the design conditions are nearly identical to production experience on the H-3 series helicopter.

TABLE XVI
WEIGHT SUMMARY,
PURE HELICOPTER DRIVE TRAIN

Component		Weight - lb
Main Gearbox		1330
Accessory Gearbox		60
Tail Rotor Drive System		209.6
	Total	1599.6

Note: All gearbox weights are dry weights.

TABLE XVII
RELIABILITY AND MAINTAINABILITY DATA,
PURE HELICOPTER DRIVE TRAIN

Component	Dynamic Component Removal Failure Rate (Failures/Flight Hour)		Total Part Failure Rate (Failures/Flight Hour)		Unscheduled MMH/FH	
	Predicted	Experience	Predicted	Experience	Predicted	Experience
Main Gearbox		0.001081(a)	0.00716	0.00650(a)	0.245	0.244 (b)
No. 1 Input Section	0.000174					
No. 2 Input Section	0.000174					
Bevel Spur Section	0.000894					
Accessory Gearbox	0.000344	0.000405(d)	0.00196	0.00231(d)	0.0171	0.0201 (d)
Tail Drive Shafting	N/A	N/A	0.00048	0.00056(d)	0.0019	0.0023 (d)
Intermediate Gearbox	0.000127	0.000150(c)	0.00073	0.00086(c)	0.0120	0.0141 (b)
Tail Rotor Gearbox	0.000067	0.000078(c)	0.00038	0.00045(c)	0.0093	0.0109 (b)
Total System	N/A	-	0.01071	-	0.2853	-

Note: Data Sources:
(a) S61L/N Main Gearbox Failure Survey. Flight hours included = 67,000 hours.
(b) CH-3C Computerized Data (AFM 66-1). Flight hours included = 50,000 hours.
(c) SH-3A/D Gearbox Removal Data, compiled from Field Discrepancy Reports, Unsatisfactory Reports, etc.
(d) CH-53A Computerized Data and Gearbox Removal Data. Flight hours included = 32,000 hours.

COMPOUND HELICOPTER TRANSMISSION SYSTEM

DESIGN GUIDELINES AND CANDIDATE SYSTEMS

The primary design requirements for the compound helicopter center on the development of lightweight and efficient drive trains for both the main rotor and the propeller. As in the case of the pure helicopter, designs are based on engines with output speeds of either 15,000 rpm or 30,000 rpm. The reliability and maintainability of the transmission system were considered throughout the design stages, and all designs proposed are based on current technology.

Design information given in the Basic Data Section indicates that, depending upon the particular mission, the propeller drive transmits a large proportion of the installed engine power for significant time periods. This requirement dictates that the drive trains to the propeller and the main rotor be of comparable efficiency so as not to compromise the performance of the complete transmission system. In addition, it was felt that the independence of the main rotor drive train from the propeller drive train would offer advantages in redundancy, reliability, and maintenance.

A number of compound helicopter drive trains were considered, and six candidate systems were chosen for examination and design evaluation in greater detail. Table XVIII summarizes, for each configuration, the type and number of gear meshes involved in the main rotor and the propeller-tail rotor drive trains.

Configuration Number 11

Initial layouts of transmission systems for the compound helicopter centered on arrangements with engines parallel to the longitudinal axis of the helicopter, as shown in Figure 35. Separation of the engines then dictates that additional gear meshes be introduced if gear pitch line velocities are to be kept within normal limits.

A further problem is that unduly large gears are necessary if any gear mesh carries the total installed engine power. It can be seen from Figure 14 that the engine driving into a spur gear at 14,981 rpm transfer its power, through an intermediate gear and a freewheel unit, to the propeller drive shaft rotating at 6251 rpm. Power from this engine is then divided in the proportions demanded by the main rotor and the propeller-tail rotor drive trains.

A single-stage planetary unit is sufficient to give a propeller speed of 1709 rpm from the 6251 rpm of the propeller drive shaft. It is then convenient to take the drive from the 1709 rpm propeller to the antitorque rotor at 1045 rpm. As is shown in Figures 14 and 35, a two-stage planetary unit is necessary to reduce to a main rotor speed of 245 rpm, the reduction ratios for the first and second stages being 3.62 and 4.08 respectively. The two-stage planetary unit contributes significantly to the transmission weight, and this initial concept indicated the desirability of eliminating one of the output planetary

TABLE XVIII
COMPOUND HELICOPTER
DRIVE TRAIN CONCEPTS

Configuration Number	Engine rpm	Number and Type of Gear Reductions: Main Rotor Drive Train	Number and Type of Gear Reductions: Propeller-Tail Rotor Drive Train
11 (Figures 14 and 35)	15,000	3 spur 1 bevel 2 planetary	2 spur 1 planetary
12 (Figures 15 and 36)	30,000 (15,000 alternate)	1 epicyclic 2 bevel 1 planetary	1 epicyclic 3 bevel
13 (Figures 16 and 37)	30,000 (15,000 alternate)	1 epicyclic 1 bevel 1 spur	1 epicyclic 1 bevel 2 spur
14 (Figures 17 and 38)	30,000 (15,000 alternate)	2 spur 1 bevel 2 planetary	2 spur 1 planetary
15 (Figures 18 and 39)	15,000	2 bevel 1 spur 1 planetary	1 bevel 1 planetary
16 (Figures 19 and 40)	15,000 (30,000 alternate)	3 bevel 1 spur 1 planetary	2 bevel 1 planetary

units.

The drive from the second engine could be arranged in a manner similar to that of the engine input already described; that is, driving into a spur gear at 15,000 rpm and then through an intermediate gear to the propeller drive shaft. But this arrangement would mean that the total power to the main rotor would have to pass through the bevel gear mesh which drives the first-stage sun gear of Figure 14. This difficulty can be avoided by arranging the second engine to drive a bevel gear and a spur gear mesh. Then part of the power from this engine is taken by the main rotor, and part of the power is transferred across the spur gear concentric with the main rotor to the bevel gear mesh and the propeller drive shaft. In this way no gear mesh is subjected to more than the power supplied by a single engine.

This advantage is offset, however, by the weight penalty and the efficiency loss introduced by the intermediate gear meshes. Power to the main rotor must pass through 1 bevel gear mesh and 3 spur gear meshes, while power to the propeller must pass through the equivalent of 1 bevel gear mesh, 3 spur gear meshes, and 1 planetary unit. The weight and efficiency values are given in Table XXV, and data on the gear trains are contained in Table XIX.

Configuration Number 12

Figures 15 and 36 show a compound helicopter drive train with a split power section at each engine input. The purpose of this input section is to allow 30,000-rpm engines to be accepted without the use of high-speed bevel gears and their associated bearings. The input planetary unit splits engine power into two drive paths in a manner identical to that of the pure helicopter Configuration Number 6. Thus, Figure 15 incorporates a counterrotating epicyclic unit at each engine input, the drive paths from each member of the epicyclic unit carrying half engine power to the bevel gear meshes linked to the ring gear and the planet carrier.

The combining bevel gear, which combines the drive paths from the planet carrier and the ring gear, is integral with a second bevel gear rated at full engine power, and not a spur gear as in Configuration Number 6. In this way, as seen from Figure 15, a second bevel gear reduction gives a speed of 900 rpm into the output planetary unit and permits a 1696-rpm drive to the propeller to be taken from the same bevel gear. A judicious choice of reduction ratios allows the transmitted torques to be adjusted so that the bevel gear driving the propeller is the same face width as the bevel pinions rotating at 2450 rpm.

The planetary drive to the main rotor is of conventional design with a reduction ratio of 3.67. As a result of the planetary unit being attached to the rotor shaft at a point of maximum slope, i.e., near the higher bearing of the rotor shaft, mounting arrangements are designed to accommodate the effects of rotor shaft slope by floating the ring gear in a splined sleeve and at the same time allowing the sun gear to float in the planet gears. As a result of having only one output planetary unit, the diameter of the rotor shaft is not limited by a small diameter sun gear and, for this reason, adequate stiffness of the rotor shaft can be obtained.

Disadvantages associated with the transmission include four gear meshes, with their associated losses, in both the drive train to the main rotor and the drive train to the propeller. Also, the drag profile of the transmission housing is wide in the region of the input epicyclic units.

Details of the gear trains are contained in Table XX. From Appendix III it is found that the gear sizes involved result in the ring gear and the planet carrier of the epicyclic units transmitting 987 horsepower and 1013 horsepower respectively.

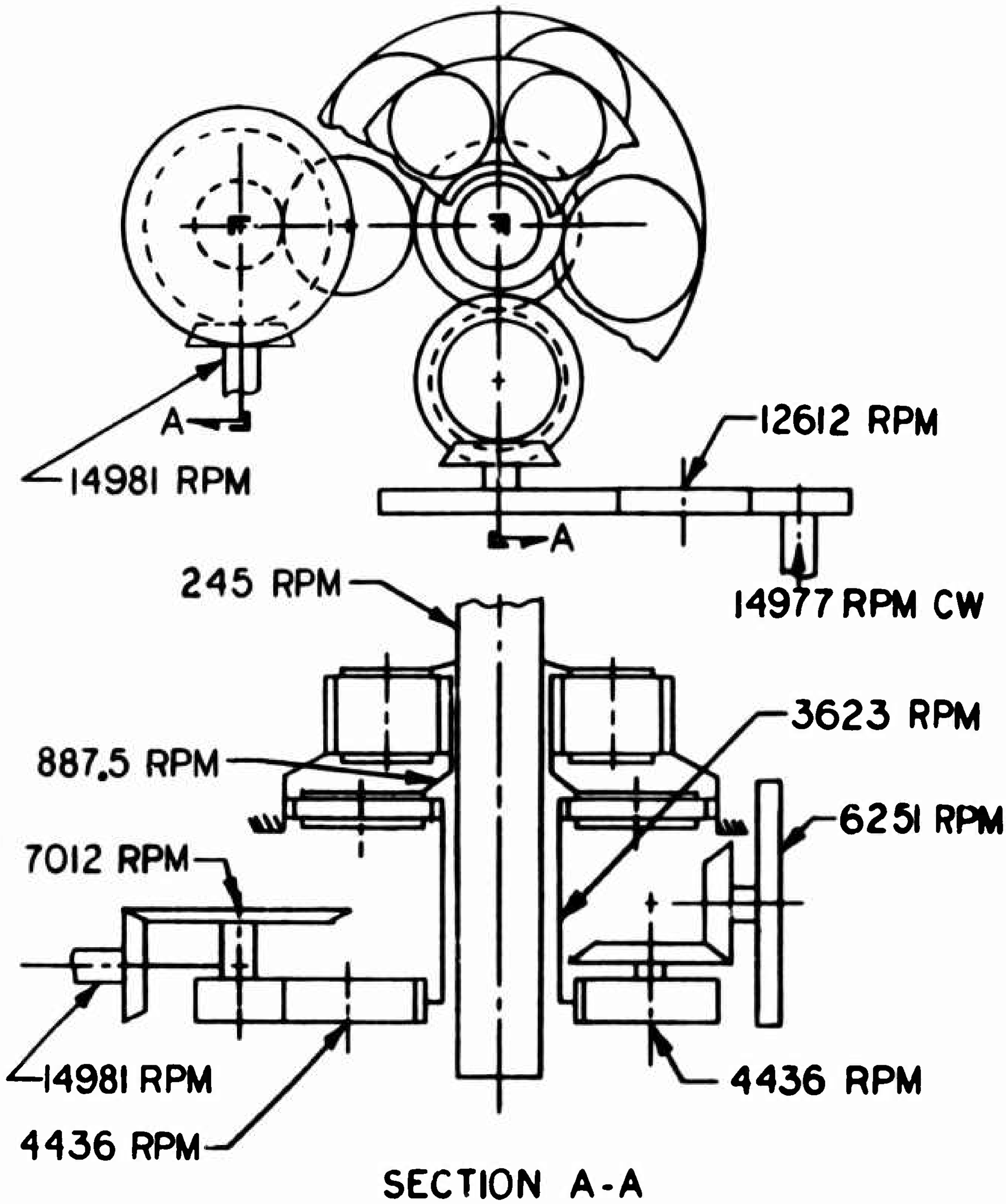


Figure 14. Gearing Schematic of Configuration Number 11.

TABLE XIX
GEAR DATA,
CONFIGURATION NUMBER 11

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	14,981	22	6.600	1.40	25,880
Bevel Gear	7,012	47	14.10	1.40	25,880
Intermediate Spur Gear	7,012	31	5.167	3.40	9,490
Intermediate Spur Gear	4,436	49	8.167	3.40	9,490
Combining Spur Gear	3,623	60	10.00	2.28	9,490
Input Spur Gear	14,977	48	6.000	1.33	23,530
Intermediate Spur Gear	12,612	57	8.125	1.48	23,530
Propeller Spur Gear	6,251	115	14.37	1.33	23,530
Propeller Shaft Bevel Pinion	6,251	22	7.20	1.76	11,780
Bevel Gear	4,436	31	10.15	1.76	11,780
Intermediate Spur Gear	4,436	49	8.167	2.30	9,490
1st Stage Planetary (4.082 RR)					
Sun	3,623	49	8.167	1.13	5,850
Pinion (5 req'd)	2,628	51	8.500	0.98	
Ring	0	151	25.17	0.65	

TABLE XIX-Continued

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
2nd-Stage Planetary (3.623 RR)					
Sun	887.5	53	7.571	4.56	1,270
Pinion (6 req'd)	792	43	6.143	4.31	
Ring	0	139	19.86	2.65	
Carrier	245				
Propeller Planetary (3.659 RR)					
Sun	6,251	41	4.10	1.95	4,880
Pinion (6 req'd)	5,477	34	3.40	1.70	
Ring	0	109	10.90	1.30	
Carrier	1,709				

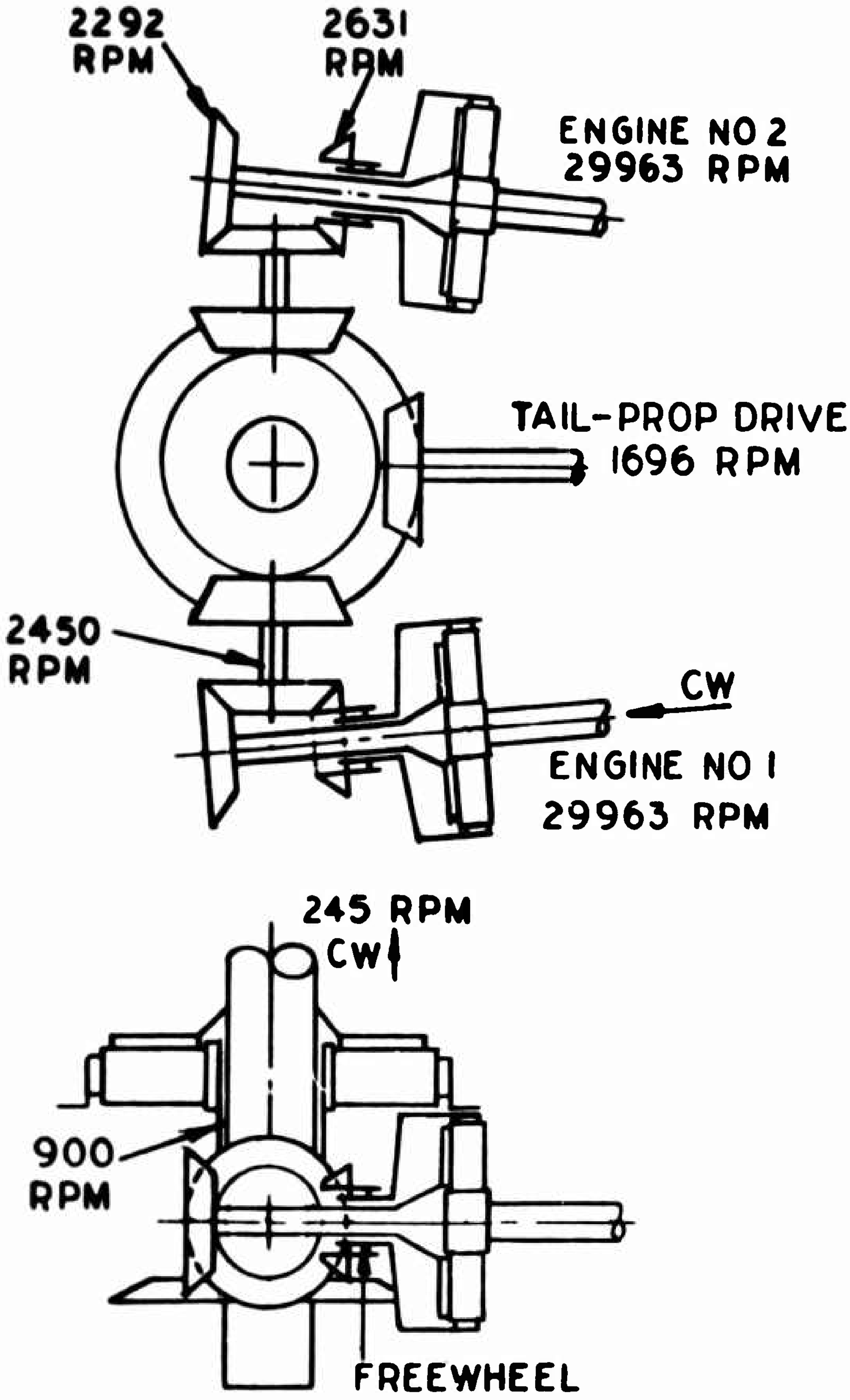


Figure 15. Gearing Schematic of Configuration Number 12.

TABLE XX
GEAR DATA,
CONFIGURATION NUMBER 12

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Width (in.)	Pitch Line Velocity (fpm)
Input Epicyclic Unit					
Sun Gear	29,963	29	2.071	1.70	15,000
Planet Pinion (3 req'd)	-11,977	67	4.786	1.45	
Ring Gear	- 2,631	163	11.643	1.20	
Carrier	2,292				
Bevel Pinion (Driven by Ring Gear)	2,631	27	7.60	1.50	5,230
Bevel Pinion (Driven by Carrier)	2,292	31	8.72	1.50	5,230
Bevel Gear (output)	2,450	29	8.16	1.50	5,230
Bevel Pinion	2,450	18	9.00	3.00	5,770
Combining Bevel	900	49	24.50	3.00	5,770
Propeller Drive Bevel	1,696	26	13.00	3.20	5,770
Main Rotor Planetary (3.673 RR)					
Sun	900	49	8.167	3.59	1,400
Pinion (6 req'd)	783	41	6.833	3.34	
Ring	0	131	21.83	2.55	
Carrier	245				

Configuration Number 13

The transmission system shown in Figures 16 and 37 is an attempt to incorporate the features of the lightest and most efficient pure helicopter transmission system, i.e., Configuration Number 5, in a compound helicopter transmission. The drive train to the main rotor, therefore, is identical to that shown in Figure 27 except that minor changes in gear ratios and torque capacities have been made to accommodate the different main rotor power and rpm required by the compound helicopter. Thus, the main rotor drive train has all the features of the most advantageous pure helicopter drive train, including high efficiency, light weight, spur and bevel gears sized for half engine power, accessible freewheels and drive train redundancy as described for Configuration Number 5.

Figure 16 shows that the propeller draws power from the planet carriers of the counterrotating epicyclic units, and when the propeller demands little power, as in hover conditions, the transmission system behaves primarily as does the pure helicopter drive train. But when the propeller draws the majority of engine power, as in forward flight, there is a change in internal directions of power flow. This change is most easily explained by noting that the ring gear section of the input epicyclic units must always deliver one half engine power (1000 hp) to the main rotor. But the main rotor may be demanding, for instance, only 250 horsepower from each of the pinions driving the rotor. Thus, the 750 excess horsepower from each ring gear passes back from the combining gear through a spur pinion to the counterrotating planet carrier. Each planet carrier then delivers to the propeller its own 1000 horsepower, i.e., half engine power, plus the 750 horsepower recirculated from the ring gear via the main rotor drive gear.

This recirculation of power would be tolerable if it were restricted to a small fraction of the total power transmitted. But the propeller demands a large proportion of the total engine power for significant periods of time, as shown in Appendix II; the result is a reduced efficiency level when compared with Configuration Number 5, the equivalent transmission for the pure helicopter.

Figure 16 shows that the propeller draws its power, at 1698 rpm, from external gears on the outside of the planet carriers. The two gears on the propeller drive shaft are linked by a torque balance unit to ensure that only half the power needed by the propeller is drawn from each epicyclic unit. Data on the gear reductions and the gear sizes are contained in Table XXI.

Configuration Number 14

The drive train configuration shown in Figures 17 and 38 accepts 30,000-rpm engine speeds and, at the same time, avoids the use of high-speed bevel gears or high-speed planetary gears. Thus, the engine input sections of the transmission are simple, and

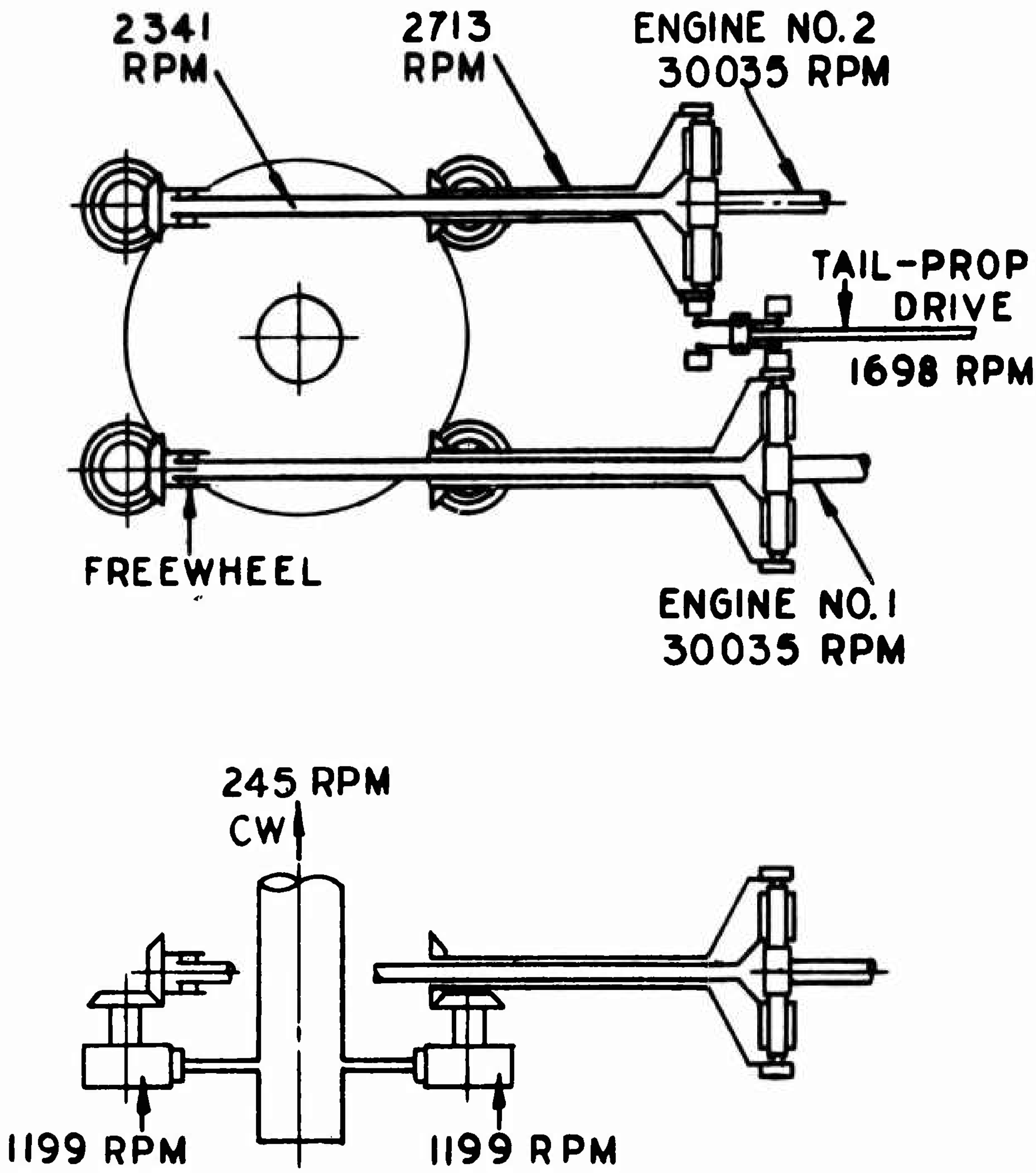


Figure 16. Gearing Schematic of Configuration Number 13.

TABLE XXI
GEAR DATA,
CONFIGURATION NUMBER 13

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Epicyclic Unit					
Sun Gear	30,035	25	2.08	1.80	15,070
Pinion (3 req'd)	-10,022	56	4.67	1.59	
Ring Gear	-2,713	137	11.42	1.20	
Carrier	2,341				
Bevel Pinion (Driven by Ring Gear)	2,713	19	6.50	2.3	4,620
Bevel Gear	1,199	43	14.72	2.3	4,620
Bevel Pinion (Driven by Carrier)	2,341	21	7.40	2.4	4,530
Bevel Gear	1,199	41	14.45	2.4	4,530
Spur Pinion	1,199	26	6.50	3.90	2,040
Combining Spur Gear	245	127	31.75	3.78	2,040
Propeller Drive Spur Finion	2,341	90	15.00	1.14	9,190
Propeller Drive Spur Gear	1,698	124	20.67	1.08	9,190

considerable flexibility in the positioning of engine axes results from the ability to adjust the positions of the input spur gears. A reduction from 30,025 rpm to 11,911 rpm at the first gear mesh allows 12,000-rpm freewheels to be used which are externally accessible for inspection and removal.

A second spur, or helical, gear reduction to 4974 rpm gives a drive shaft which, as shown in Figure 17, combines the power from each engine and supplies main rotor power to a bevel reduction gear and propeller power to a planetary reduction gear. Power to the main rotor passes from the bevel gear reduction into a two-stage planetary unit as shown in Figure 17, the first and second stages having reduction ratios of 3.18 and 3.11 respectively. It follows that power to the main rotor must pass through two spur gear trains, one bevel mesh and two planetary gear trains, while power to the propeller must pass through two spur gear trains and a planetary unit.

This transmission system is simple in concept and, except for the rather high rotational speed of the freewheels, presents no severe problems with regard to bearings, gear mounting, or gear pitch line velocities. Data on gear sizes and gear ratios are contained in Table XXII.

Configuration Number 15

The need for a simple and efficient drive train to the propeller led to the transmission arrangement shown in Figures 18 and 39. It is seen from this figure that the engines are inclined to the longitudinal axis of the helicopter in order to combine the power of the two engines after a single bevel gear mesh. Then the reduction to propeller rpm can be obtained with the addition of only one planetary unit. In this way the propeller drive losses are only those associated with one bevel gear reduction and one planetary reduction.

Power demanded by the main rotor is drawn through a torque dividing mechanism mounted in line with the propeller drive shaft. The torque dividing mechanism ensures that the power to the main rotor is split into two equal parts so that the subsequent bevel and spur reduction meshes to the main rotor can be sized for half rotor power. An important feature is that the torque dividing unit maintains equal torques, and powers, to each bevel mesh despite variations in shaft stiffness and variations in component tolerances.

After the combining spur gear, the final reduction to main rotor rpm is by a single-stage planetary unit which is mounted with a floating ring gear and a floating sun gear to assist in load sharing between the planet pinions.

This transmission configuration is primarily suited to 15,000-rpm engines, since the use of 30,000-rpm engines would mean unduly large bevel gears on the shaft which combines engine power. It will be noted from Figure 18 that this combining shaft is in line with the

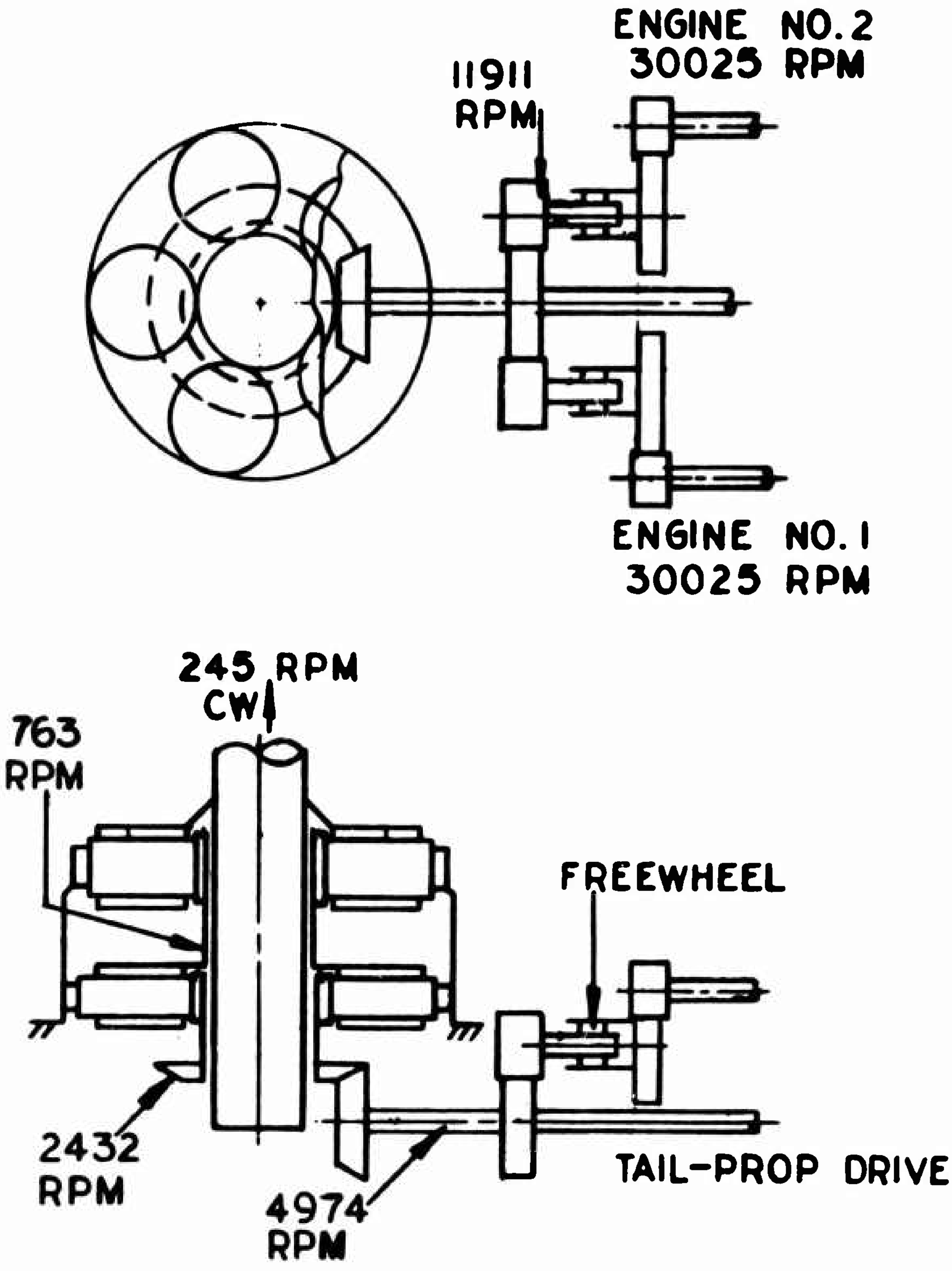


Figure 17. Gearing Schematic of Configuration Number 14.

TABLE XXII
GEAR DATA,
CONFIGURATION NUMBER 14

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Spur Pinion	30,025	48	4.00	1.39	31,440
Input Spur Gear	11,911	121	10.08	1.37	31,440
2nd-Stage Spur Pinion	11,911	38	6.667	1.43	20,790
Spur Combining Gear	4,974	91	15.17	1.23	20,790
Bevel Pinion	4,974	22	8.25	2.90	10,740
Bevel Gear	2,432	45	16 87	2.90	10,740
Main Rotor 1st-Stage Planetary (3.186 RR)					
Sun	2,432	59	9.833	1.95	4,300
Pinion (4 req'd)	2,813	35	5.833	1.70	
Ring	0	129	21.50	1.50	
Main Rotor 2nd-Stage Planetary (3.115 RR)					
Sun	763.1	61	10.17	4.46	1,380
Pinion (5 req'd)	930	34	5.667	4.21	
Ring	0	129	21.50	3.95	
Carrier	245				

TABLE XXII-Continued

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Propeller Drive Planetary (2.921 RR)					
Sun	4,974	63	6.30	2.23	5,400
Pinion (4 req'd)	7,105	29	2.90	1.98	
Ring	0	121	12.10	1.55	
Carrier	1,703				

propeller drive shaft, which allows a transmission housing with a narrow drag profile. Details of the gear size and gear reduction ratios are given in Table XXIII.

Configuration Number 16

The addition of another gear train enables the concepts involved in Configuration Number 15 to be used with a 30,000-rpm engine. Figures 19 and 40 show the engines parallel to the longitudinal axis of the helicopter. The first bevel gear mesh reduces the speed to 10,440 rpm, and the freewheel units can be conveniently placed on this intermediate shaft. The engines then supply power to a single bevel gear that is used to combine the power of the two engines. The remainder of the transmission is then virtually identical with Configuration Number 15.

This configuration allows the engines to be located further forward than in the preceding configuration, but the use of 30,000-rpm engines has, of course, introduced another bevel gear mesh together with the problem of bearings for the 30,000-rpm bevel pinion. Gear data are given in Table XXIV.

DESIGN EVALUATION - COMPOUND HELICOPTER DRIVE TRAIN

The selection of a particular transmission configuration for the compound helicopter involves the factors of weight, efficiency, reliability and maintenance. Pertinent comments on the probability of the lightest weight transmission involving the smallest number of gear meshes have been made on page 40 and these comments are also valid for the compound helicopter transmission system. It is evident from an examination of the candidate systems that a central problem is the requirement that the drive trains to both the main rotor and the propeller must carry a major portion of the installed engine power for significant periods of time.

Reliability and maintainability were also major considerations in designing and evaluating the preceding drive trains. Each configuration was reviewed in detail during layout design, and changes were incorporated to reduce maintenance man-hours per flight hour and accomplish increased reliability in the drive train. A component analysis and a failure mode analysis were conducted to establish estimated removal failure rates and total failure rates of the gearboxes. Removal failure rate is defined as the rate of failure causing gearbox removal, while total failure rate is the rate of all failures, including those requiring gearbox removal plus those not requiring removal. This latter item includes such failures as leakage of field replaceable seals, component wear or corrosion detected during (but not the cause of) overhaul, etc. The removal failure rate can be reduced by providing greater accessibility and more field replaceable components and by including fault detection equipment in various sections of the gearbox, thereby locating as well as indicating the malfunction. The data bank used during the preceding evaluations included that obtained from operational experience with the H-3 and H-53 series production helicopters.

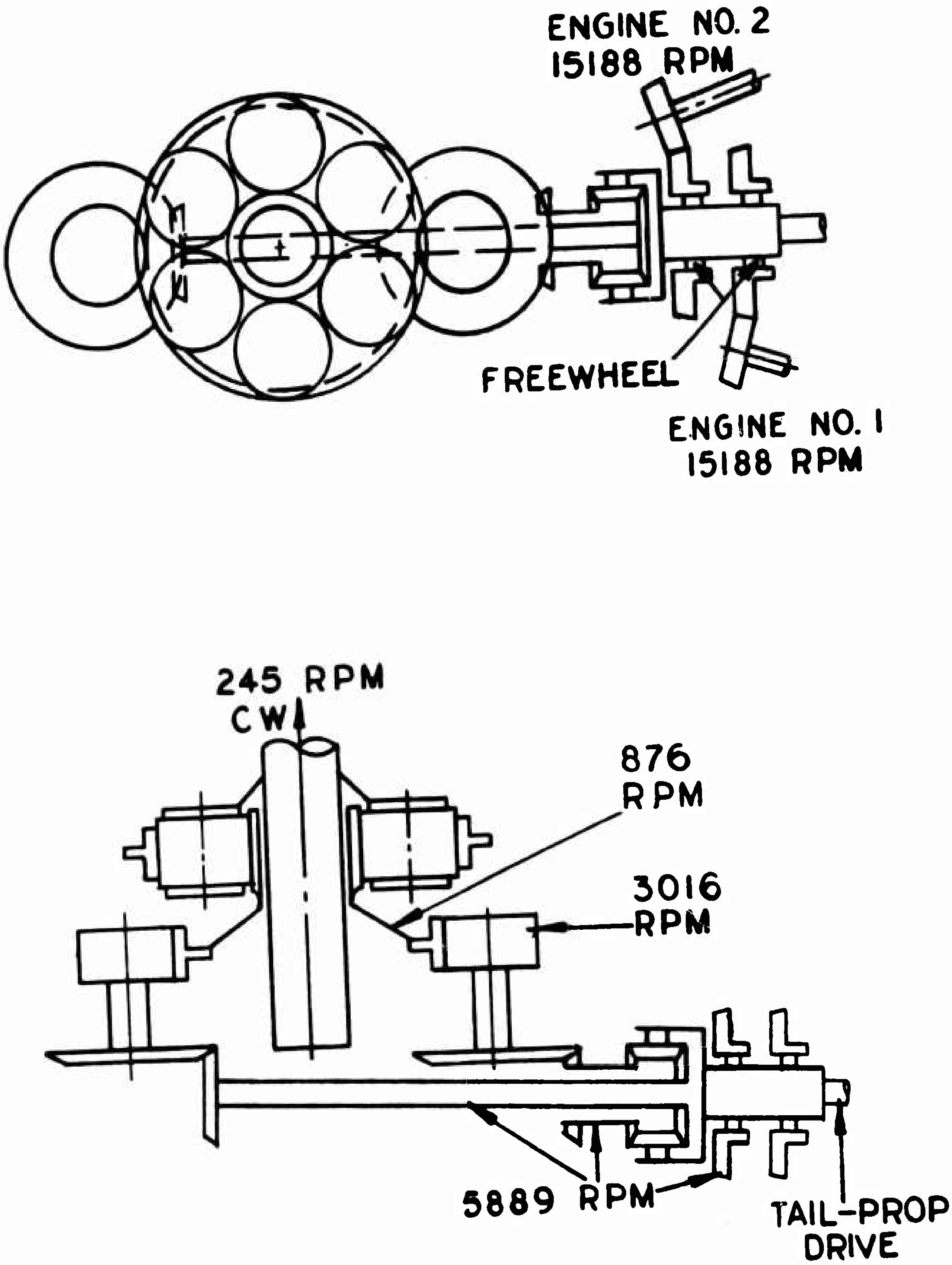


Figure 18. Gearing Schematic of Configuration Number 15.

TABLE XXIII
GEAR DATA,
CONFIGURATION NUMBER 15

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	15,188	19	4.362	1.75	17,350
Combining Bevel Gear	5,889	49	11.25	1.75	17,350
Bevel Pinion From Torque Divider (2 req'd)	5,889	21	6.15	2.125	9,480
Bevel Gear (2 req'd)	3,016	41	12.00	2.125	9,480
Combining Spur Pinion	3,016	36	6.00	3.60	4,740
Combining Spur Gear	876	124	20.67	3.56	4,740
Main Rotor Planetary (3.574 RR)					
Sun	876	47	7.83	4.34	1,290
Pinion (6 req'd)	802	37	6.17	4.09	
Ring	0	121	20.17	3.00	
Carrier	245				
Propeller Planetary (3.469 RR)					
Sun	5,889	71	5.071	1.73	7,830
Pinion (6 req'd)	5,720	52	3.714	1.48	
Ring	0	175	12.50	1.30	
Carrier	1,698				
Torque Divider Pinion		15	2.50	0.625	
Torque Divider Gear		27	4.33	0.625	

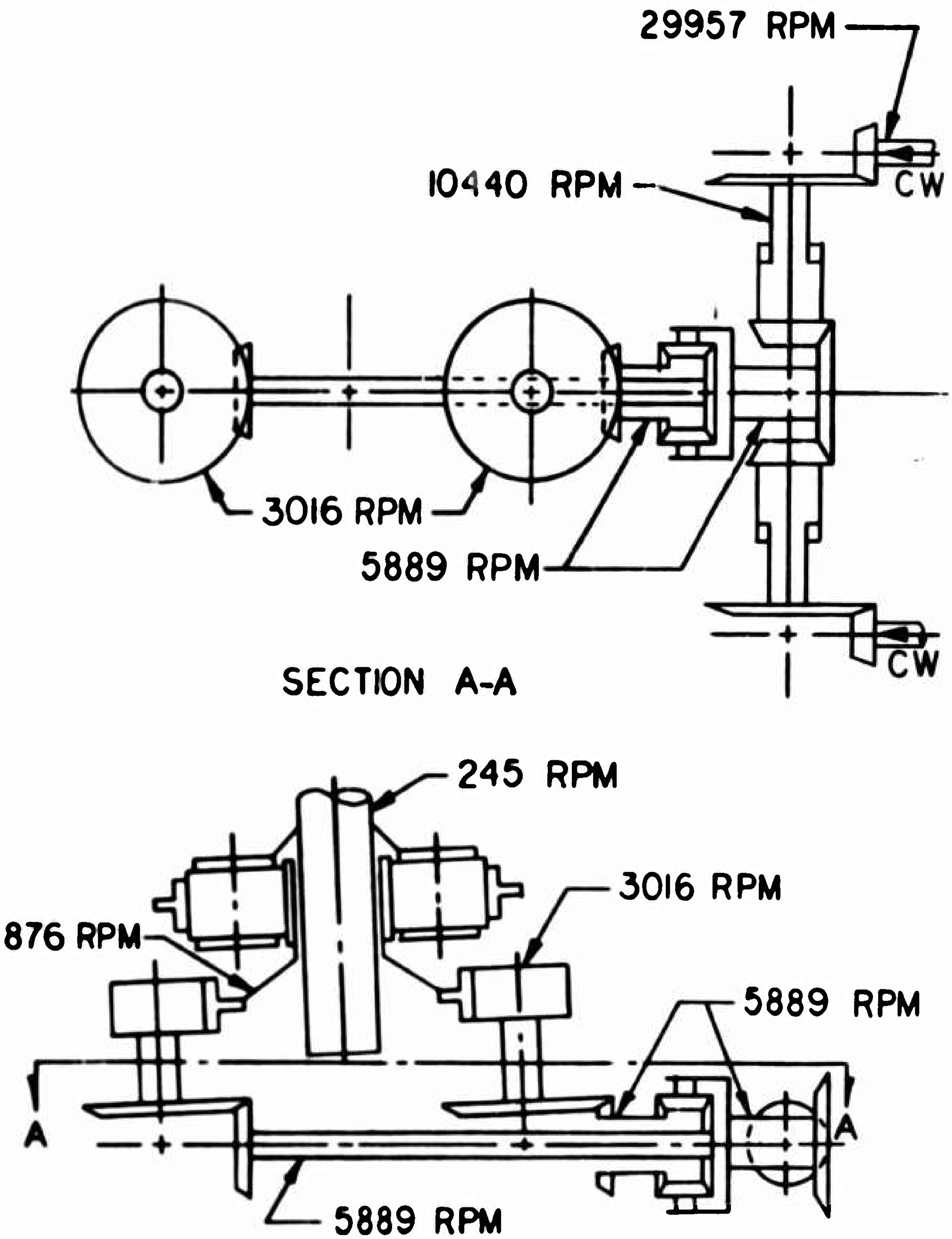


Figure 19. Gearing Schematic of Configuration Number 16.

TABLE XXIV
GEAR DATA,
CONFIGURATION NUMBER 16

Component	rpm	Number of Teeth	Pitch Diameter (in.)	Face Width (in.)	Pitch Line Velocity (fpm)
Input Bevel Pinion	29,957	23	3.82	1.35	29,960
Input Bevel Gear	10,440	66	10.96	1.35	29,960
Combining Bevel Pinion	10,440	22	5.30	1.65	14,490
Combining Bevel Gear	5,889	39	9.39	1.65	14,490
Bevel Pinion From Torque Divider (2 req'd)	5,889	21	6.15	2.125	9,480
Bevel Gear (2 req'd)	3,016	41	12.00	2.125	9,480
Combining Spur Pinion	3,016	36	6.00	3.60	4,740
Combining Spur Gear	876	124	20.67	3.56	4,740
Main Rotor Planetary (3.574 RR)					
Sun	876	47	7.83	4.34	1,290
Pinion (6 req'd)	802	37	6.17	4.09	
Ring	0	121	20.17	3.00	
Carrier	245				
Propeller Planetary (3.469 RR)					
Sun	5,889	49	4.90	1.88	5,380
Pinion (5 req'd)	4,343	36	3.60	1.63	
Ring	0	121	12.10	1.20	
Carrier	1,698				
Torque Divider Pinion		15	2.50	.625	
Torque Divider Gear		27	4.33	.625	

The use of layout designs, and the nature of this phase of the program, indicated that comparative data were more useful than absolute values, since the final system would be developed considering the overall evaluation of the various designs. Thus, the removal failure rates and the total failure rates in Table XXV are given relative to Configuration Number 15.

The advantages and disadvantages of each configuration are summarized in Table XXVI, while Table XXV includes the weight and efficiency of each configuration. The efficiency of the transmission varies according to whether the main rotor or the propeller draws the majority of engine power; for this reason, Table XXV gives the efficiency of the transmission when each of the main rotor and the propeller drive trains transmits its maximum power.

In addition, equivalent system weights have been obtained for the six candidate systems, and these are shown in Figure 20. These equivalent system weights have been determined using an average system efficiency from Table XXV and precedures as outlined on page 41 . However, for the compound helicopter, each horsepower lost by inefficiency represents approximately 4.8 pounds.

From Table XXV and Figure 20, Configurations 13 and 15 exhibit significant weight advantages over the other four configurations. Other factors such as reliability and maintainability clearly enter into the selection of a transmission system, and Table XXV also compares the removal failure and total failure rates of the six compound helicopter drive train configurations. Thus, Configurations 13 and 15 are not deficient on a reliability basis and, on account of their weight advantage, it is evident that these two configurations are the stronger candidates for the compound helicopter transmission.

TABLE XXV
DESIGN DATA,
COMPOUND HELICOPTER TRANSMISSIONS

Config Number	Weight (lb)	lb per hp	Efficiency: Main Rotor (percent)	Efficiency: Propeller Tail Rotor (percent)	Comparative: Removal Failure Rate	Comparative: Total Failure Rate
11	1685	0.42	96.0	96.7	1.19	4.13
12	1605	0.40	96.5	96.7	1.16	4.56
13	1410	0.35	97.2	96.7	1.03	4.78
14	1635	0.41	97.0	97.5	1.05	3.81
15	1400	0.35	97.0	98.0	1.0	4.47
16	1465	0.37	96.5	97.5	1.02	4.62

TABLE XXVI
COMPARATIVE COMMENTS,
COMPOUND HELICOPTER DRIVE TRAIN

Configuration Number	Advantages	Disadvantages
11	11(a) Components similar to those currently in use.	11(a) Idler gears on each engine drive train.
	11(b) Externally accessible freewheel units.	11(b) Two-stage planetary reduction unit fastened to main rotor at point of maximum slope.
12	12(a) Uses single-stage output planetary.	12(a) Large-diameter combining bevel gear.
	12(b) Freewheels rated at half engine power, and can be externally accessible.	12(b) Power to propeller is carried by a single bevel gear.
		12(c) Transmission housing has wide drag profile.
13	13(a) No output planetary required for main rotor.	13(a) Power recirculation from main rotor, back through input epicyclic units, when propeller draws power.
	13(b) Bevel gears and spur gears to main rotor are sized for half engine power.	13(b) Torque balance unit is necessary for propeller drive train.
	13(c) Accessible freewheel sized for half engine power.	13(c) Large-diameter spur gears for propeller drive.
	13(d) Loads on all gear trains are accurately established.	
14	Conventional design, most components similar to those currently in use.	14(a) High-speed spur gearing with 30,000 rpm engines.
		14(b) High-speed freewheel units.

TABLE XXVI-Continued

Configuration Number	Advantages	Disadvantages
		14(c) Two stage-planetary reduction unit fastened to main rotor at point of maximum slope.
15	15(a) Single output planetary unit for main rotor.	15(a) Torque balance unit is necessary.
	15(b) Bevel gears and spur gears to main rotor are sized for half rotor power.	
	15(c) Simple drive to propeller	
	15(d) Transmission housing has narrow drag profile.	
16	16(a) Same as 15(a).	16(a) Same as 15(a)
	16(b) Same as 15(b).	16(b) Additional bevel gear mesh to both main rotor and propeller as compared to Configuration Number 15.
	16(c) Engines can be mounted parallel to fore-aft axis of helicopter, and more forward than with Configuration Number 15.	16(c) With 30,000-rpm engines, special materials must be used to accommodate loads on bearings of input bevel pinion.

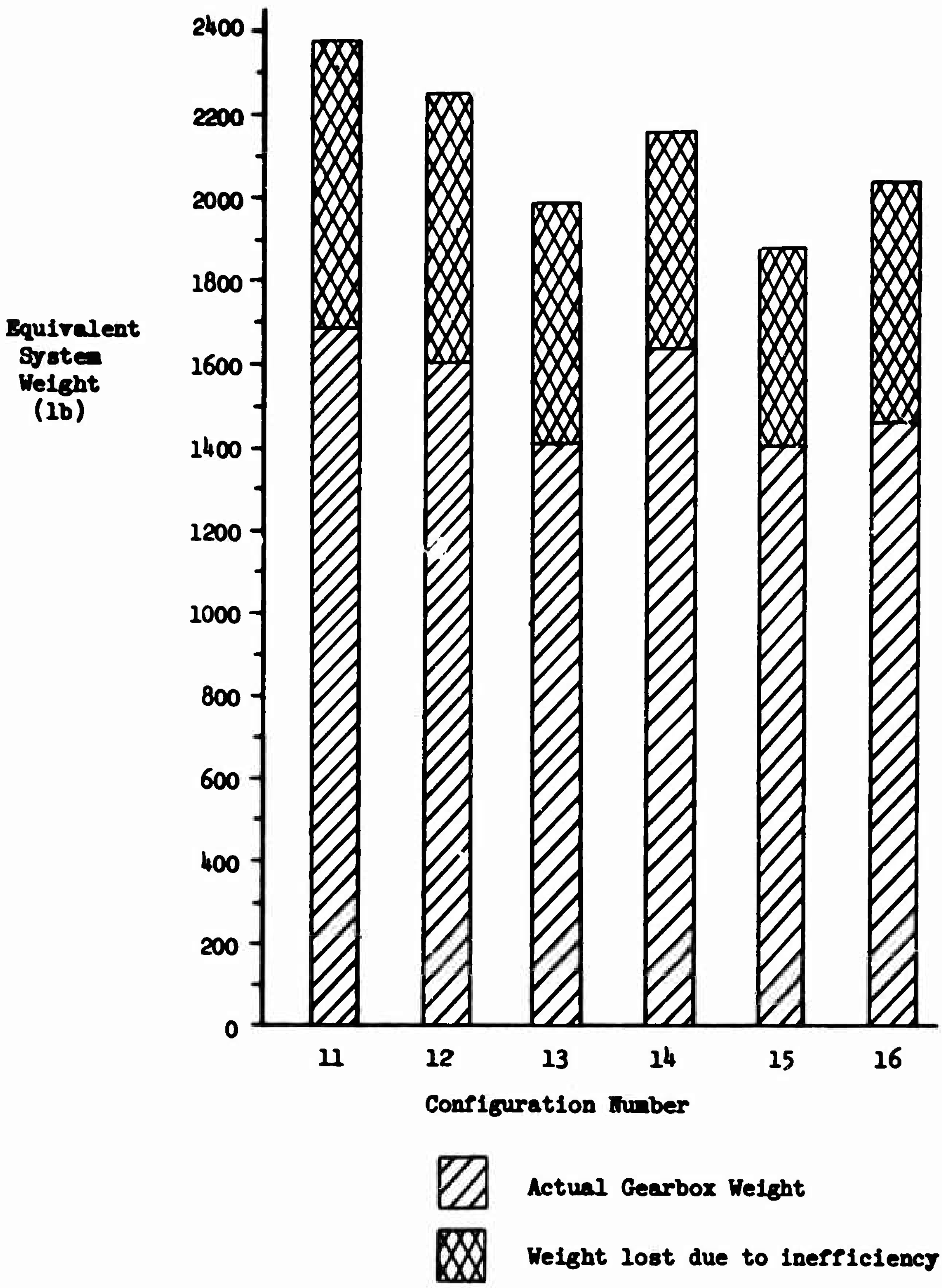


Figure 20. Weight Comparison, Equivalent System Weights.

Configuration Number 13, while having approximately the same weight as Configuration 15, is hampered by the recirculation of power from the main rotor drive train to the propeller drive train, giving a lowered efficiency during transfer of power to the propeller. This recirculation of power, while not uncommon in other transmission systems, is a phenomenon which is unusual enough in the case of helicopter transmissions to resolve the selection of a transmission system in favor of Configuration Number 15.

Although Configuration Number 15 has been developed for a compound helicopter application, it is clearly suitable for a pure helicopter if the propeller drive train is omitted. Minor modifications would be necessary to accommodate the power output and rpm of the pure helicopter main rotor. This alternative transmission cannot match the weight of the system selected for the pure helicopter, nor can it accept 30,000-rpm engines, but it provides a useful alternative to the six pure helicopter configurations. The weights of the compound helicopter drive trains given in Table XXV should not be directly compared with those of the pure helicopter drive trains given in Table XII, since there are significant differences in both the prorated and maximum value of output torque to the main rotors.

DETAILED DESCRIPTION OF SELECTED SYSTEM

Introduction

The comparison of transmission arrangements for the compound helicopter demonstrated that Configuration Number 15 provided the best overall characteristics.

The main features of the final transmission shown in Figures 21, 41, and 42 are as follows:

(1) The main transmission can be fabricated to a weight of 0.36 pound per horsepower using current technology.

(2) At maximum rated power to the main rotor, the transmission system is 96.7 percent efficient; at maximum rated power to the propeller, the transmission system is 97.3 percent efficient.

(3) Equalization of tooth loads on the bevel and spur gears in the main rotor drive is achieved by a torque balancing unit.

(4) Tooth load variations in the planetary drive units to the main rotor and the propeller are minimized by the use of a floating sun gear and a floating ring gear.

(5) The transmission housing has a narrow drag profile as a result of each engine driving directly into a combining drive shaft.

(6) The input gearing, freewheel units, torque balancing unit, and propeller drive trains can be easily removed from the main transmission housing.

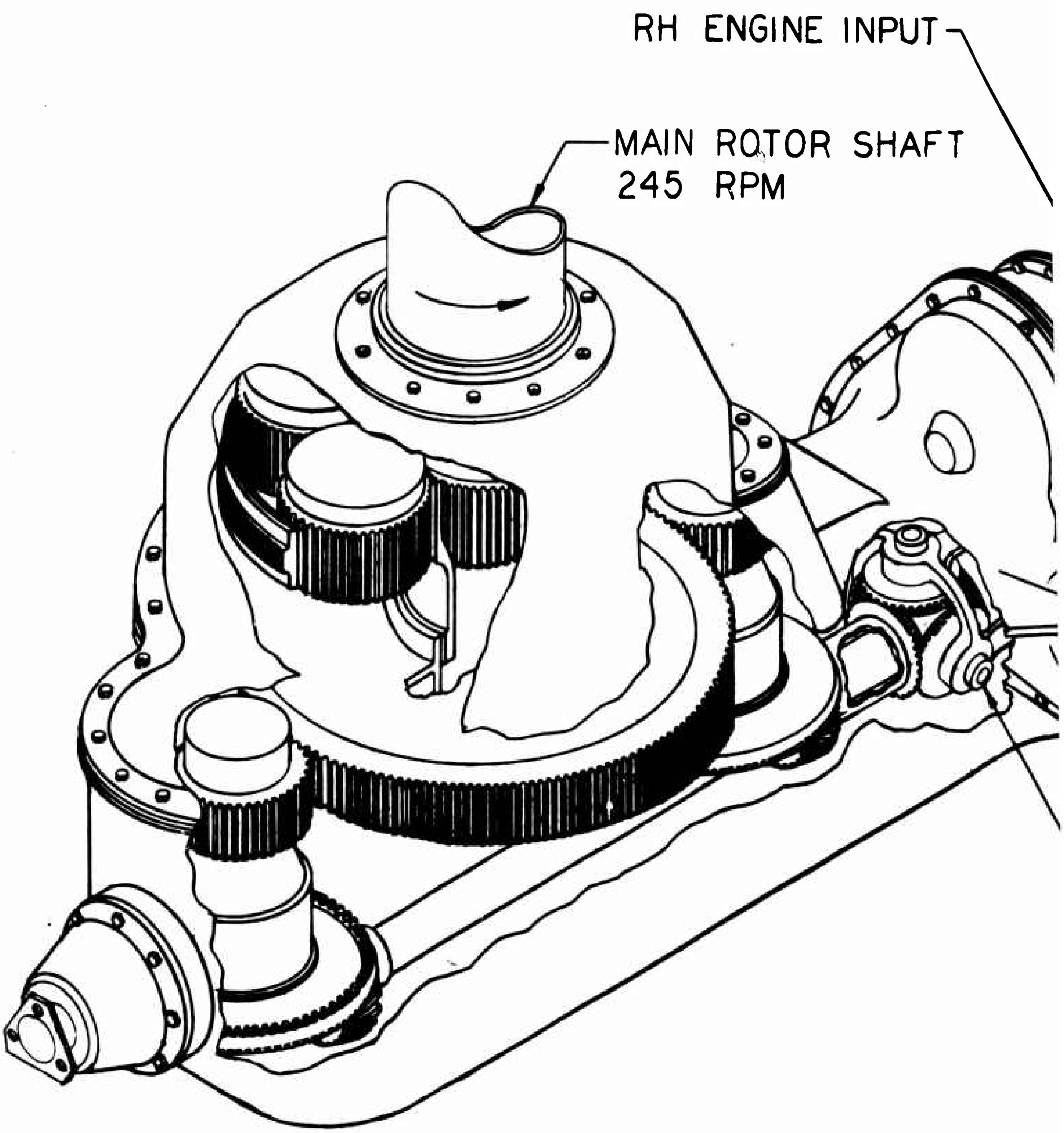
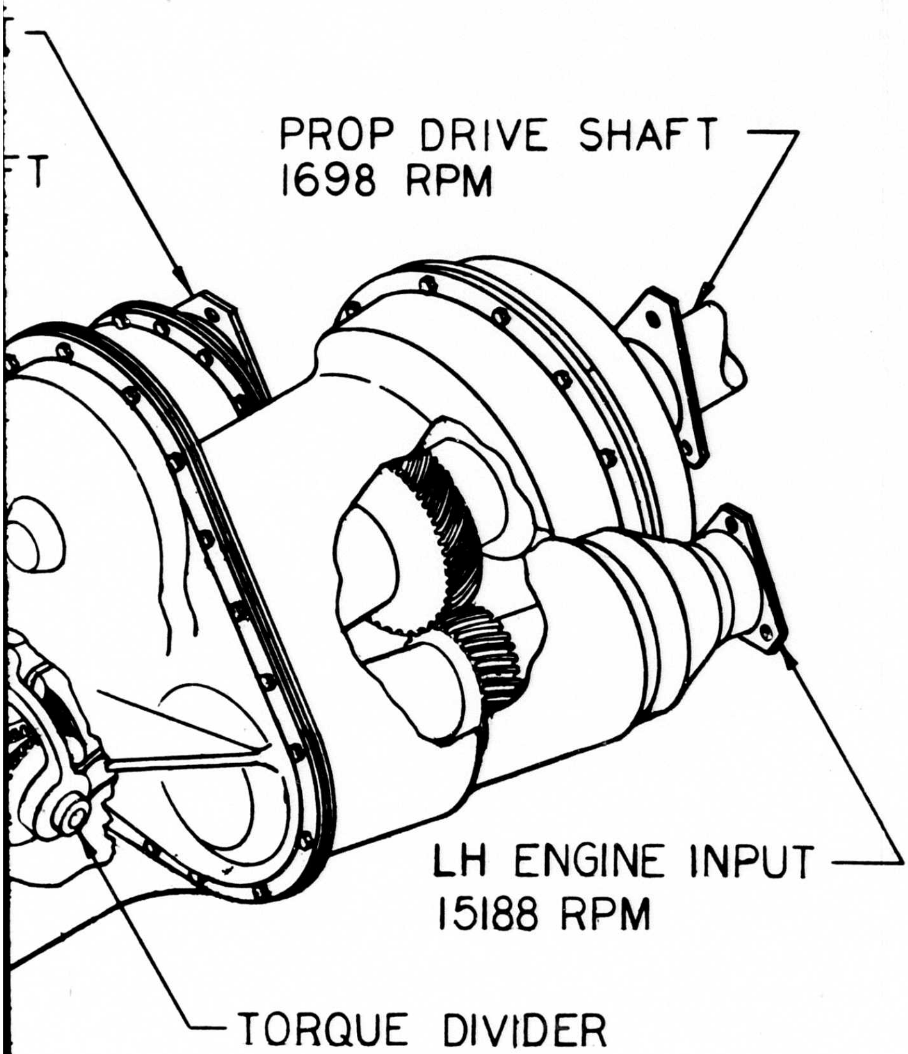


Figure 21. Main Transmission Arrangement, Compound Helicopter.

A


PROP DRIVE SHAFT
1698 RPM
LH ENGINE INPUT
15188 RPM
TORQUE DIVIDER

B

Main Gearbox

Engine Input Section

Each engine is fanned out 15 degrees from the longitudinal centerline of the aircraft while also tilted 2.5 degrees upward relative to a waterline. This arrangement allows a speed of 5889 rpm to be obtained from the initial bevel reduction while limiting pitch line velocities to 17,400 fpm. As shown in Figure 42, the propeller drive speed of 1698 rpm is obtained from a single planetary unit mounted on the engine combining shaft.

Two bevel gears on the combining shaft are necessary as a result of choosing to place roller type freewheel units, rotating at 5889 rpm, on this shaft. A considerable weight savings could be realized, however, by developing 15,000-rpm freewheel units suitable for mounting on the engine input shafts. In this case only one combining bevel gear would be required and the engines could be moved nearer the main transmission housing.

The design of the input sections is such that the engine input shafts, the combining bevel gears, and the freewheel units can be removed as a subassembly from the main transmission housing. The bevel gears driving the freewheels are designed so that their end thrusts are opposed, leaving a relatively small residual thrust load to be carried by the bearing which locates the shaft that combines engine power. This transmission configuration is primarily suited to 15,000-rpm engines since, with this engine speed, 1 bevel gear reduction stage is adequate to provide an acceptable speed for the driven combining shaft which is in line with the propeller drive shaft. Higher engine speeds would necessitate another speed reduction stage.

Main Rotor Drive Train

Figure 42 shows that power from each engine passes from an input bevel gear to a combining shaft which passes below the main rotor shaft. The power demanded by the main rotor is transmitted through a torque dividing mechanism which ensures that the power for the main rotor is split into two equal components so that the subsequent bevel and spur reduction trains to the main rotor can each be sized for half rotor power. In the absence of the torque dividing unit, it would be necessary to take all main rotor power through a single bevel gear mesh and then use two planetary reduction stages to obtain a rotor speed of 245 rpm.

The two bevel pinions driven by the torque dividing unit are in line with the propeller drive shaft, this arrangement providing a transmission housing with a narrow drag profile since the housing need be only slightly larger than the 21-inch-diameter combining gear. A reduction ratio of 1.95 is obtained from the bevel gears linked to the torque dividing unit, and a further reduction of 3.45 gives a sun gear speed of 876 rpm.

Balanced radial loads on the combining gear decrease bearing loads and weights. The weight of the gear itself is kept to a minimum by using thin section gear webs which are electron beam welded to the gear rim. After the spur combining gear, the final reduction to rotor speed is accomplished by a planetary unit in which the sun gear is of sufficient diameter to allow a rigid rotor shaft to pass through. In this way the rotor shaft slope at the upper rotor shaft bearing is kept below 0.002 inch per inch and the resultant radial and angular displacement of the planetary unit is kept within acceptable limits. As a further aid to accommodating shaft deflections, and at the same time assisting in load sharing between the planet pinions, the ring gear is allowed to float in its retaining spline. At the same time the sun gear can pivot on its drive spline and follow the position of the planet gears. Table XXXIV gives details of the gear sizes, stress levels and reduction ratios used in the transmission system.

Propeller-Tail Rotor Drive

The drive to the propeller is seen from Figure 41 to pass from each engine into the initial bevel reduction gear trains and then to the sun gear of a planetary unit with a reduction ratio of 3.46. Positioning the planetary unit near the propeller and transmitting torque to the propeller at 5889 rpm requires an additional lubrication and cooling system and contributes to an undesirable aft center of gravity. For these reasons the planetary unit has been placed adjacent to the freewheel units on the main gearbox.

Torque Dividing Unit

The purpose of the torque dividing unit is to ensure that the power drawn by the main rotor is divided equally between two spiral bevel meshes. At the same time the torque dividing unit will absorb internally the small angular variations between the two drive shafts caused, for instance, by manufacturing tolerances, casing deformations and drive lines of differing stiffness. Since the torque divider rotates as a unit, there is no power loss from the device and the Zerol bevel gear teeth are not subjected to alternating stresses. Torque transmitted to the main rotor is carried by 3 pinions in the torque dividing unit, thus sharing the loads between 6 gear meshes. When in operation the torque dividing unit will experience very small or no relative angular motion; for these reasons the best performance would appear to result from keeping the bearing loads reasonably low so as to minimize any tendency to fretting or brinelling. Care has also been taken in the design to ensure that the bearings of the torque dividing unit are constantly immersed in oil. The gear tooth stresses and bearing loads which can be accommodated must, ultimately, be decided by testing and field experience, but the stress levels given in Table XXXIII are conservative by current industrial standards. Moreover, large reductions in stress levels would involve only a small weight penalty, or an additional pinion, since the complete unit weighs less than 11 pounds.

Efficiency of Transmission

Losses in the transmission are calculated on the same basis as explained on page 55 for the pure helicopter. Full load operation can exist either when the main rotor is demanding its maximum power, or when the propeller is demanding its maximum power. The distribution of engine power in these two cases is given in Table XL.

Loss sources and efficiency of transmission when maximum power is being transmitted to main rotor

Main Rotor Drive - 3300 hp Output

Gear Mesh	Loss, Percent	Loss, hp
1 input bevel	0.5	16.79
1 bevel	0.5	16.71
1 combining spur	0.5	16.62
1 planetary	0.75	24.75
churning loss	0.75	24.75
	Main Rotor Drive Losses	99.62

Propeller Drive - 100 hp Output

Gear Mesh	Loss, Percent	Loss, hp
input bevel	0.5	0.5
1 planetary	0.75	0.75
churning loss	0.75 x 2708/100	20.30
	Propeller Drive Losses	21.55

Tail Rotor Drive - 320 hp

Gear mesh	Loss, Percent	Loss, hp
input bevel	0.5	1.62
1 planetary	0.75	2.41
1 bevel	0.5	1.60
churning loss	0.75	2.41
	Tail Rotor Drive Losses	8.04

Accessory Drive - 123 hp

Gear Mesh	Loss, Percent	Loss, hp
3 bevel	1.5	1.85
churning loss	0.75	.92
	Accessory Drive Losses	2.77

Input Power Required

Main Rotor Drive	3300 +	99.62	=	3399.62
Propeller Drive	100 +	21.55	=	121.55
Tail Rotor Drive	320 +	8.04	=	328.04
Accessory Drive	123 +	2.77	=	125.77
Total hp req'd	3843 +	131.98	=	3974.98

Loss sources and efficiency of transmission when maximum power is being transmitted to propeller

Main Rotor Drive - 766 hp Output

Gear Mesh	Loss, Percent	Loss, hp
input bevel	0.5	3.90
1 bevel	0.5	3.88
1 combining spur	0.5	3.86
1 planetary	0.75	5.75
churning loss	0.75 x 3300/100	24.75
	Main Rotor Drive Losses	42.14

Propeller Drive - 2633 hp Output

Gear Mesh	Loss, Percent	Loss, hp
input bevel	0.5	13.25
planetary	0.75	19.75
churning loss	0.75 x 2633/100	19.75
	Propeller Drive Losses	52.75

Tail Rotor Drive - 75 hp

Gear Mesh	Loss, Percent	Loss, hp
input bevel	0.5	0.38
planetary	0.75	0.56
bevel	0.5	0.38
churning loss	0.75 x 75/100	0.56
	Tail Rotor Drive Losses	1.88

Accessory Drive - 123 hp

Gear Mesh	Loss, Percent	Loss, hp
3 bevel	1.5	1.85
churning loss	.75	.92
		2.77

Input Power Required

Main Rotor Drive	766	+	42.14	=	808.14
Propeller Drive	2633	+	52.75	=	2685.75
Tail Rotor Drive	75	+	1.88	=	76.88
Accessory Drive	123	+	2.77	=	125.77
Total hp req'd	3597	+	99.54	=	3696.54

$$\text{Efficiency} = \left(1 - \frac{\text{losses}}{\text{input hp}}\right) 100 = 96.7 \text{ percent}$$

Heat Generated in Main Gearbox

The above gear tooth losses and bearing losses appear as heat, and the gearbox lubrication system must be adequate to remove this heat while keeping the oil temperature rise within a specified limit. The total heat generated is given by

$$Q_G = 2545\ F_{hp}$$

where $F_{hp} = 131.98$

$$Q_G = 336{,}000 \text{ Btu/hr}$$

With MIL-L-7808 oil, an inlet temperature of 176°F, and an outlet temperature of 230°F, the required oil flow is given by:

$$W_o = \frac{C_e \ (42.4) \ F_{hp}}{0.1337 \ C_p \ (\rho) \ \Delta T}$$

where the factor C_e = 0.60 is an experience factor obtained from measurements on production main gearboxes.

$$W_o = \frac{0.60 \ (42.4) \ (131.98)}{0.1337(0.528)(54.4)(54)}$$

$$= 16.2 \text{ gpm}$$

One oil pump has been selected for installation in this gearbox. This pump rotates at 3016 rpm, is driven as shown in Figure 42, and has an oil capacity of 20 gpm. To provide redundancy and adequate lubrication even if one pump should fail, a second oil pump could be driven by the oil cooler spur gear train. The oil flow noted above necessitates an oil sump capacity of 10 gallons. A schematic of the lubrication system is shown in Figure 22.

TABLE XXVII
WEIGHT SUMMARY,
MAIN GEARBOX

Item	Assembly Weight (lb)
Input assembly, including housing, bevel gearing, freewheel unit, and combining shaft	315
Torque divider, including support shaft and bearings	14
Bevel gear assemblies, including spur pinions	120
Output planetary	340
Main rotor shaft, bearings, and combining spur gear	260
Prop-tail rotor drive planetary, including output shaft	58
Main housing, liner, and stud assembly, including planetary attachments	290
Accessory drives	43
Sump housing and oil pump assembly	24
TOTAL	1464

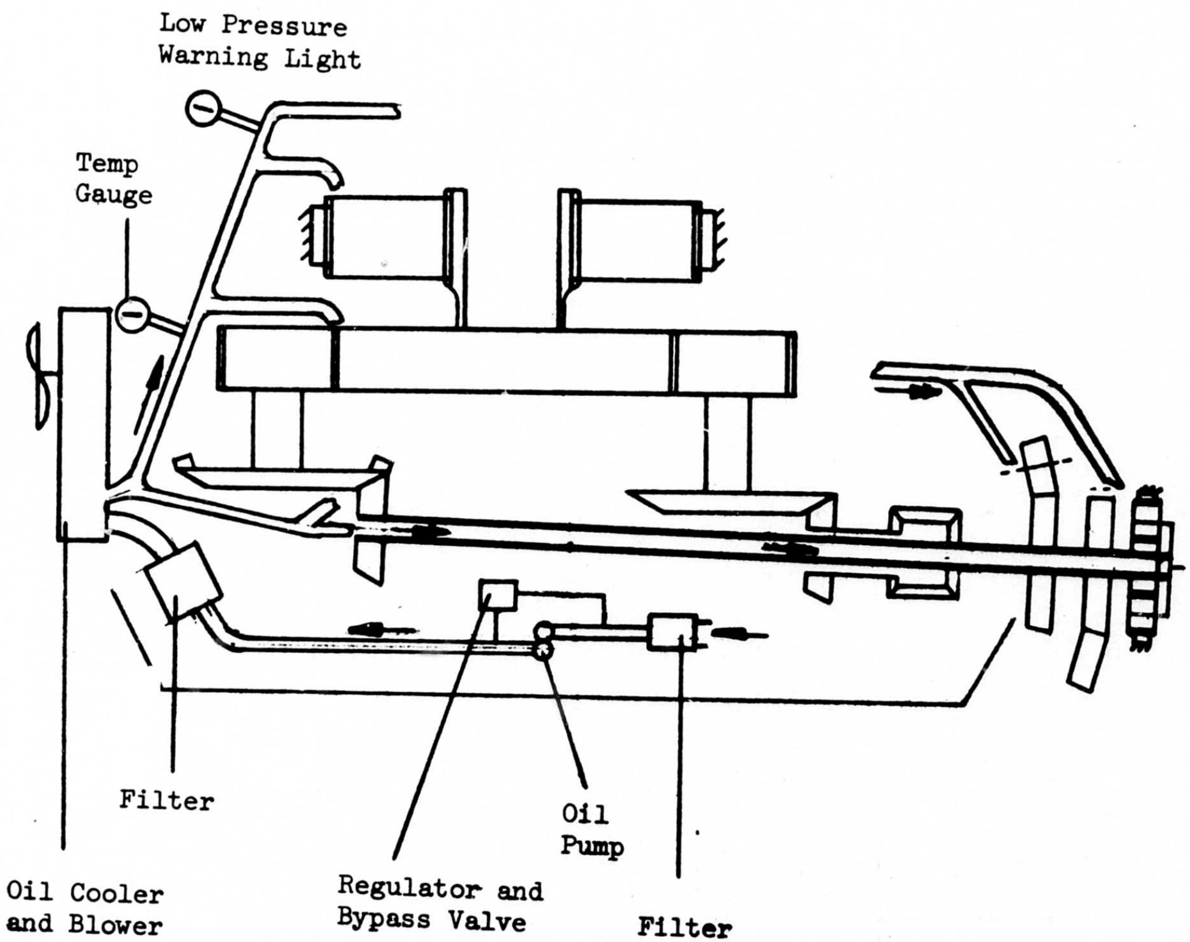


Figure 22. Lubrication System, Main Gearbox.

Propeller-Tail Rotor Drive System

Design Consideration

The main and tail rotors must be interconnected by shafting and the associated gearboxes to provide speed synchronization and permit transfer of power that is required during various flight regimes, including autorotation. The shafting must be designed to transmit the required power, accommodate the imposed deflections, provide an appropriate operational critical speed relationship, be lightweight, and yet permit fabrication by available manufacturing technology. The general shafting design considerations outlined in the description of the pure helicopter tail rotor drive system apply to this installation as well.

Description

The main gearbox incorporates a planetary on the combining gear shaft that provides an output speed of 1698 rpm. Although a 6000-rpm shafting installation would weigh approximately 35 pounds less than a 1700-rpm installation, such shafting would require a large gearbox adjacent to the propeller.

The location of this gearbox incorporating the planetary reduction would then displace the aircraft center of gravity further aft, as well as require a separate lubrication system. These two considerations caused the epicyclic unit to be placed in the main gearbox, providing a 1698-rpm drive shaft installation. The proposed tail drive shafting between the main and intermediate gearboxes consists of 5 similar sections of 2024-T3 aluminum alloy tubing with a 4340 steel fitting riveted to each end. This 6.0-O.D. by 5.7-I.D. drive shafting is supported on antifriction ball bearings soft-mounted on viscous damped supports as shown in Figure 43 and is connected by means of flexible disk couplings. Each flexible disk coupling consists of 25 stainless steel laminations, 8.5-O.D. by 6.0-I.D., with a disk pack thickness of 0.306 inch. The propeller-tail rotor gearbox, shown in Figure 44, accomplishes the required speed reduction for the tail rotor and provides the support for the propeller. The tail rotor, located on butt line 78, is supported by a double pillow-block bearing arrangement and is connected to the propeller-tail rotor gearbox by a splined shaft.

The efficiency of the propeller-tail rotor gearbox is approximately 99 percent. Based on extensive production experience with similar gearboxes, splash lubrication provides adequate lubrication and cooling for this gearbox and the tail rotor support bearing housing. The propeller-tail rotor gearbox will operate satisfactorily with 0.4 gallon of oil, while the tail rotor support bearings require 1 pint of oil. All bearings have unfactored lives in excess of 3,000 hours. Gearing data on the gearbox are given in Table XXXIII. Weights are given in Table XXVIII.

Accessory Gearbox

The accessory gearbox for the compound helicopter is identical to that developed for the pure helicopter. The gearbox arrangement is shown in

TABLE XXVIII PROPELLER-TAIL ROTOR DRIVE SYSTEM WEIGHT		
Component		Weight - lb
Shafting - Main Gearbox to Propeller - Tail Rotor Gearbox		175
Propeller - Tail Rotor Gearbox (minus the rotor hub & controls)		86
Shafting - Propeller - Tail Rotor Gearbox to Tail Rotor		23
Tail Rotor Support Bearings and Housing		31
	Total	315

Figure 34.

Reliability and Maintainability Analysis

The selected system was analyzed to determine the estimated reliability and maintainability of the various components that comprise the entire drive train. Major components were analyzed considering the effect of each individual part. The reliability of the transmission system components was determined by comparing the drive train components to similar production components with known service experience. Based upon extensive operational experience, the following parameters were predicted:

(1) Removal failure rate of each dynamic component

(2) Total part failure rate to the subsystem level

(3) Unscheduled maintenance man-hours per flight hour (MMH/FH) to the subsystem level

The above data are given in Table XXIX.

TABLE XXIX
RELIABILITY AND MAINTAINABILITY DATA,
COMPOUND HELICOPTER DRIVE TRAIN

Component	Dynamic Component Removal Failure Rate (Failures/Flight Hour)		Total Failure Rate (Failures/Flight Hour)		Unscheduled MMH/FH	
	Predicted	Experienced	Predicted	Experienced	Predicted	Experienced
Main Gearbox		0.001081 (a)	0.00619	0.00650 (a)	0.2400	0.2440 (b)
Input Section	0.000172					
T.T.O. Section	0.000391					
Main Drive Section	0.000912					
Accessory Gearbox	0.000344	0.000405 (c)	0.00196	0.00231 (c)	0.0171	0.0201 (c)
Tail Drive Shaft Installation	N/A	N/A	0.00133	0.00056 (a)	0.0055	0.0023 (c)
Propeller - Tail Rotor Gearbox	0.000109	- (a)	0.00137	- (a)	0.0285	- (b)
Total System	N/A	-	0.01085	-	0.2911	-

Note: Data Sources (a) S61L/N Main Gearbox Failure Survey. Flight hours included = 67,000 hours.

(b) CH-3C Computerized Data (AFM 66-1). Flight hours included = 50,000 hours.

(c) CH-53A Computerized Data and Gearbox Removal Data. Flight hours included = 32,000 hours.

TABLE XXX
WEIGHT SUMMARY,
COMPOUND HELICOPTER
DRIVE TRAIN

Components		Weight-lb
Main Gearbox		1464
Accessory Gearbox		60
Propeller-Tail Rotor Drive System		315
	Total	1839

Note: All gearbox weights are dry weights.

CONCLUSIONS

As a result of this drive train configuration study, it is concluded that:

1. The increased speed reduction necessitated by higher rpm engines can be more conveniently accommodated in the helicopter transmission system than in an additional engine gearbox. However, the engine mounting interface requirements and output drive shaft configurations should be reevaluated in order to simplify the requirements imposed on gearbox design.

2. Power splitting within the main gearbox can offer design flexibility as a result of:

   (a) Coaxial output shafts from a high reduction ratio counterrotating epicyclic unit.

   (b) Providing redundant load paths.

   (c) In general, increased component accessibility for inspection and maintenance.

   (d) Predictable tooth loads in the transmission system as a result of an inherent torque balancing effect.

   Therefore, an analytical and experimental evaluation of various torque dividing mechanisms should be conducted, since such devices can provide design flexibility and may assume even greater importance as larger turbine engines are utilized in helicopter applications.

3. The transmission system selected for the pure helicopter demonstrates both a significant weight reduction and an increase in efficiency when compared to current transmission designs. These improvements are largely the result of the equivalent of only one epicyclic unit, one bevel gear mesh, and one spur gear mesh transmitting total input power.

4. The selected compound helicopter drive train allows high power to be transmitted to either the rotor or the propeller through separate drive trains without any gear mesh carrying more than single-engine power.

5. The "free-planet" planetary system reduction offers a significant weight savings over conventional gear reduction systems and appears practicable for helicopter applications, providing the load sharing concept involved performs as anticipated. A further detailed design investigation is recommended in order to evaluate the reliability and the potential weight savings of using the free-planet concept in the drive train of a production helicopter.

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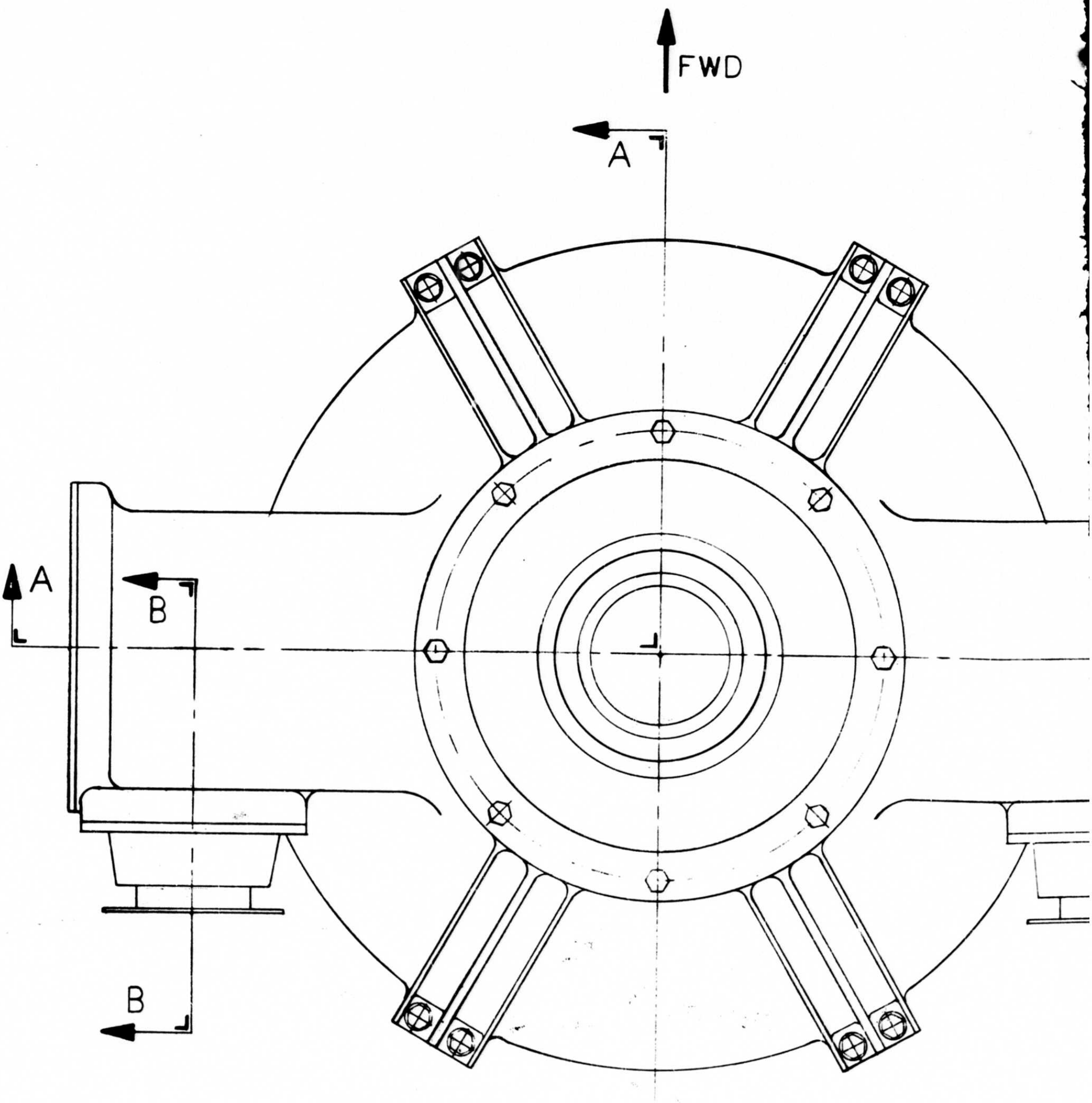
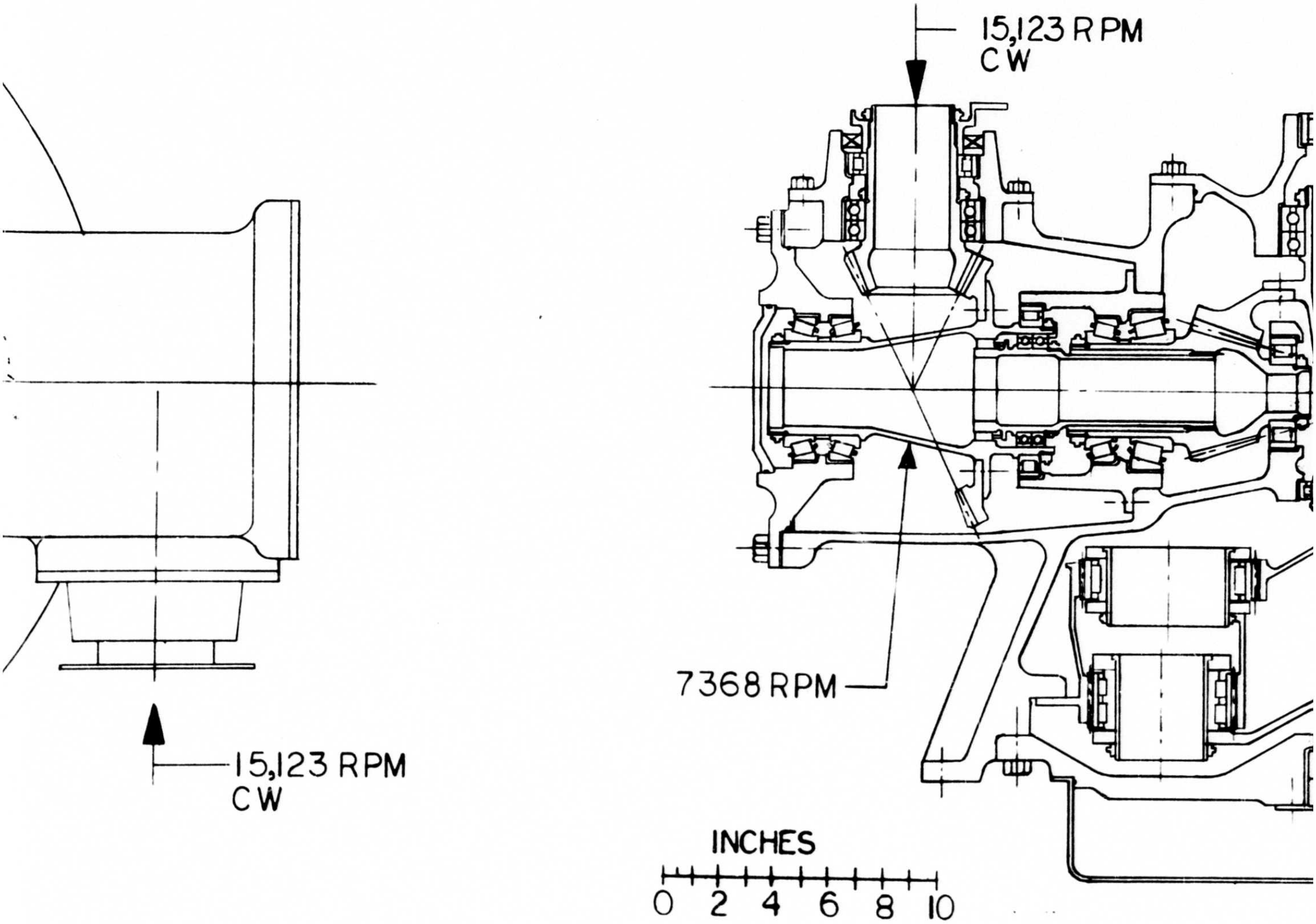


Figure 23. Configuration Number 1.

A



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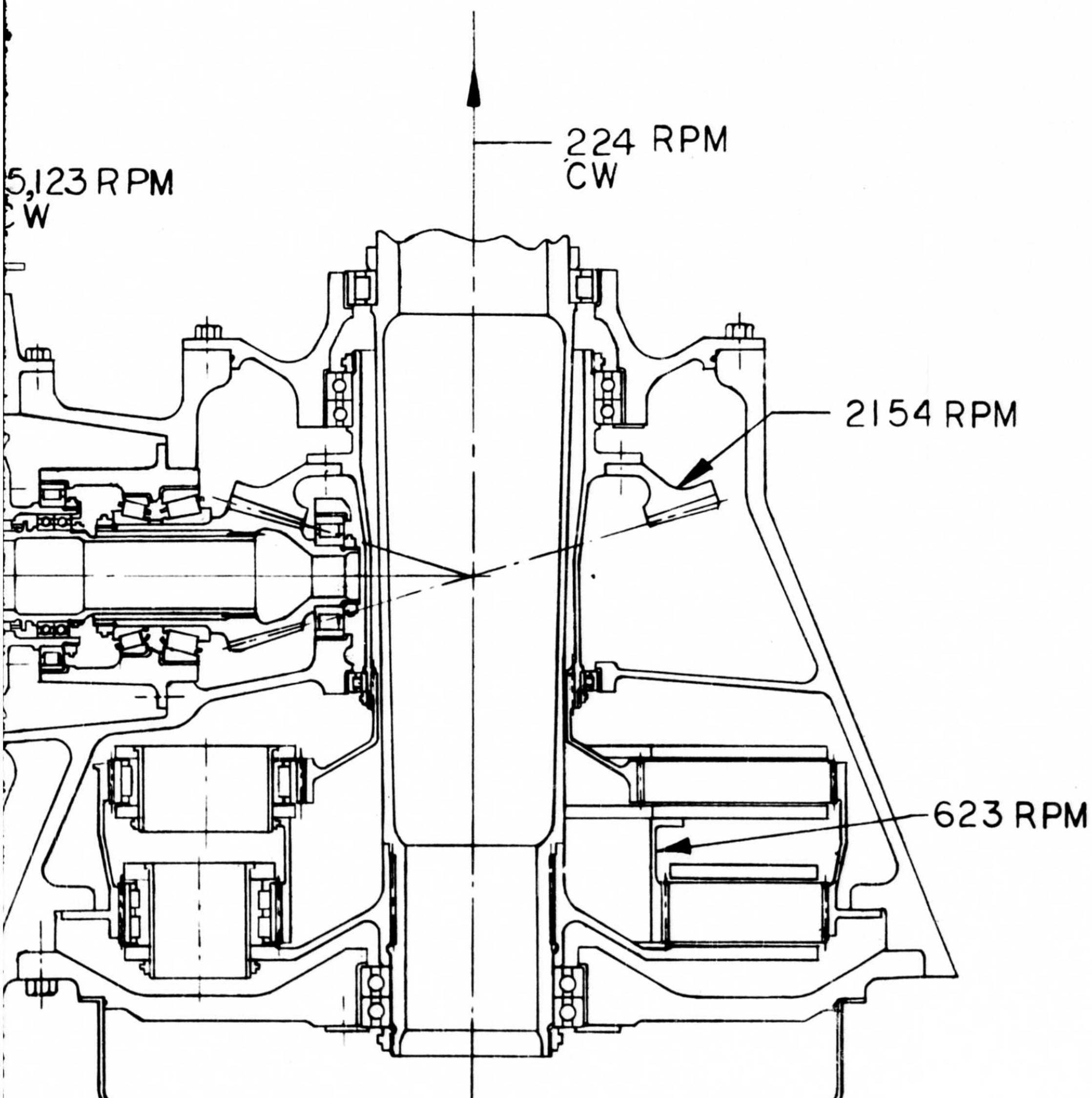
SECTION B B ROTATED 90° UP

B



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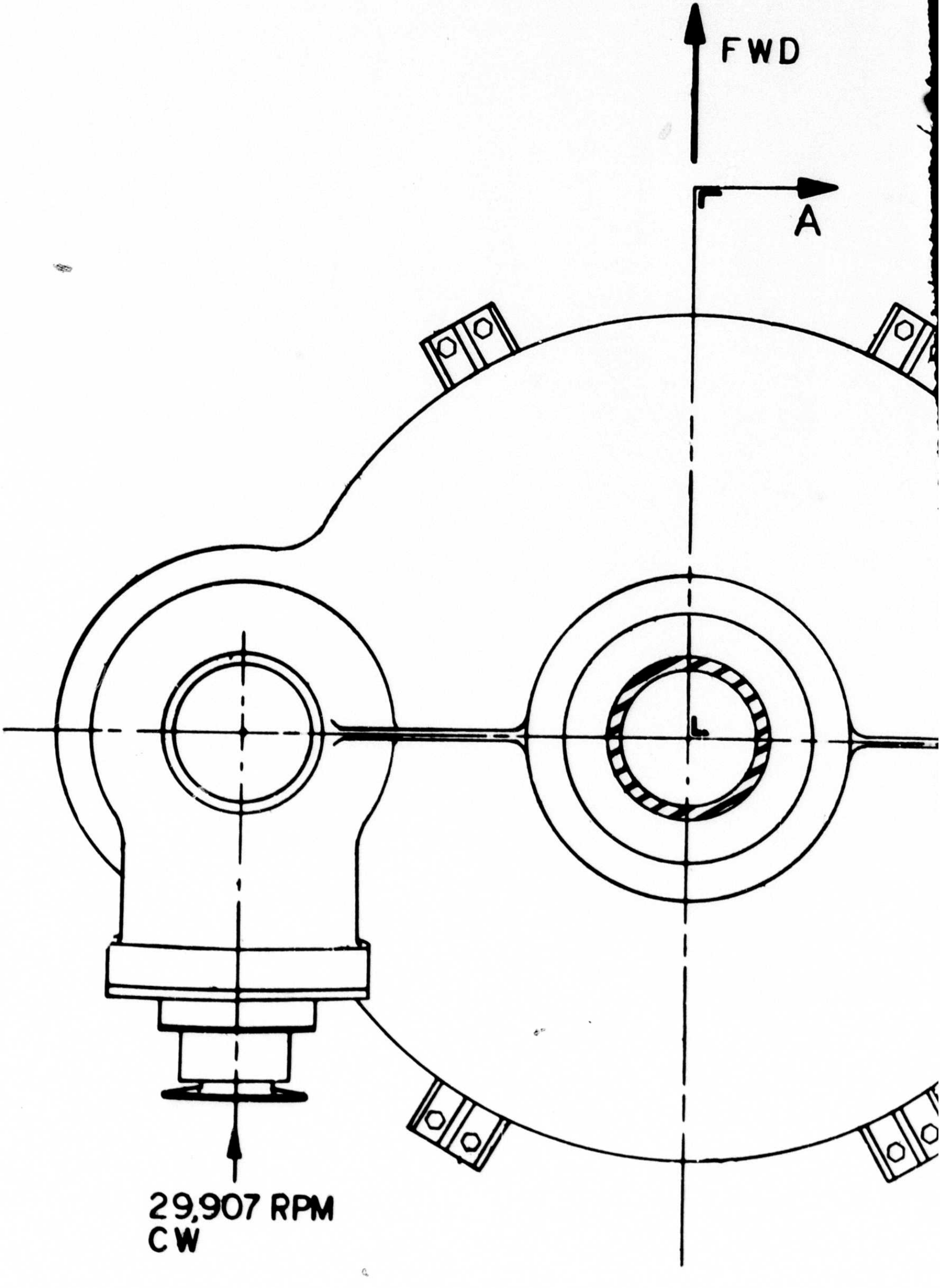
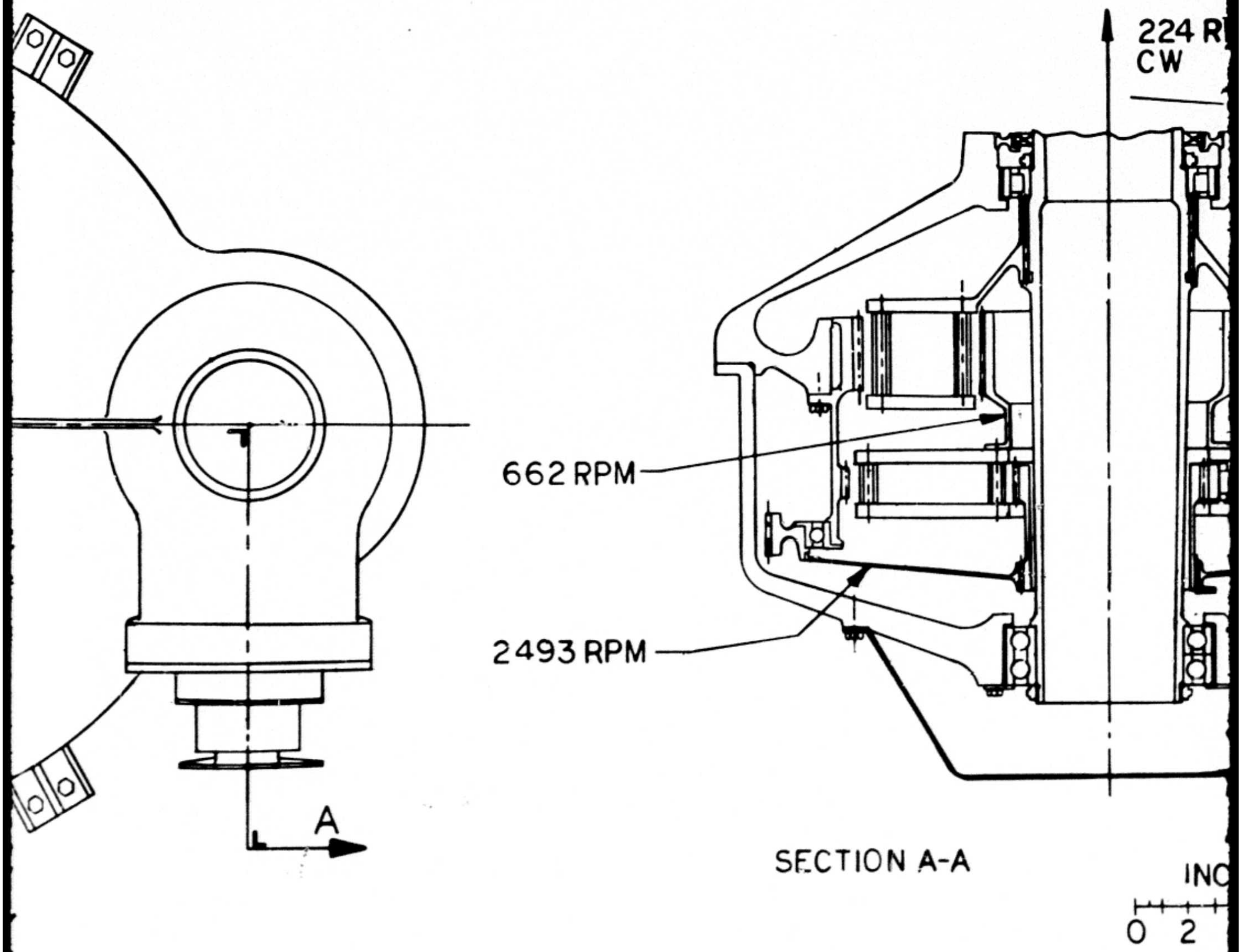


Figure 24. Configuration Number 2.

A



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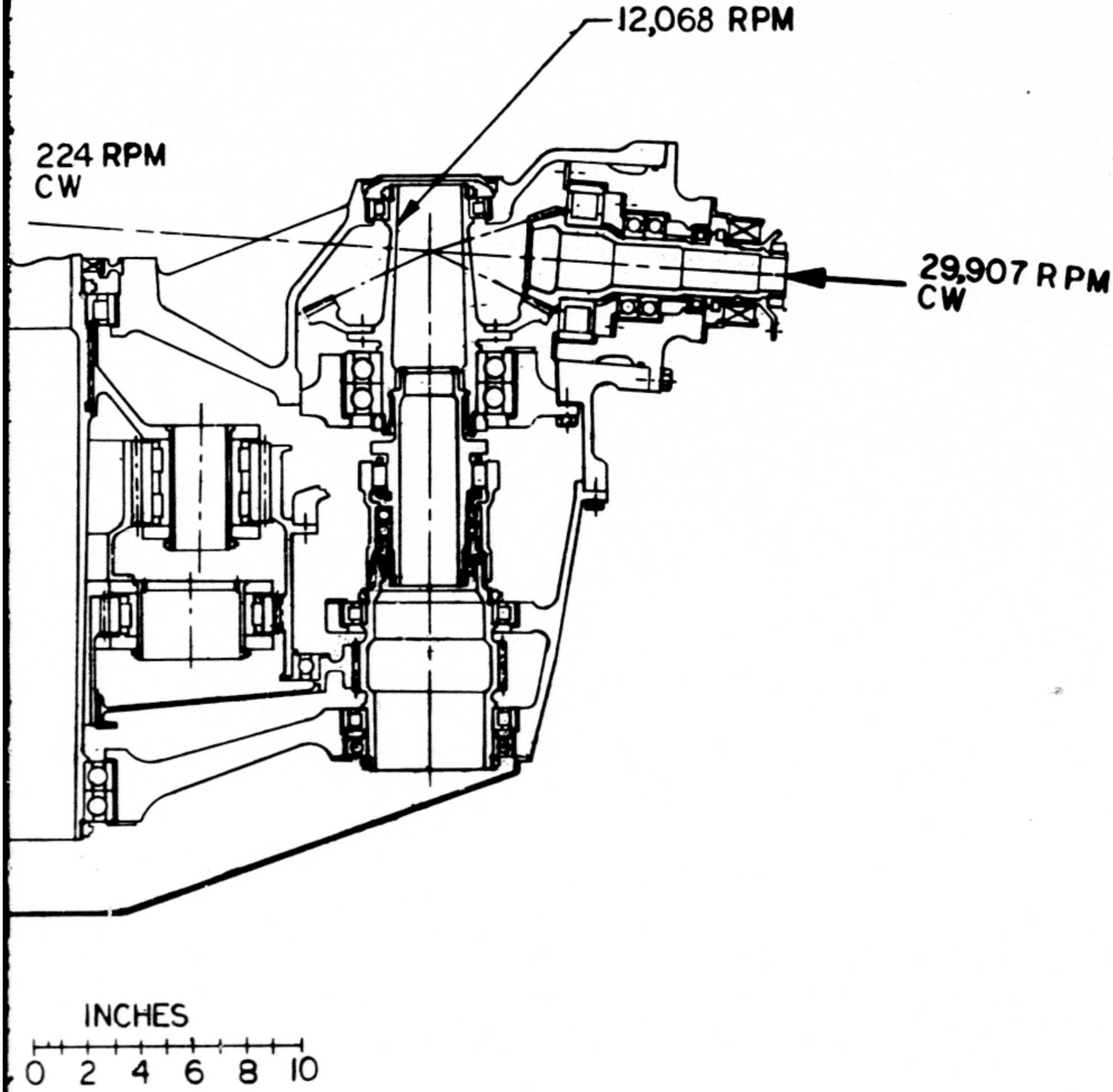


C

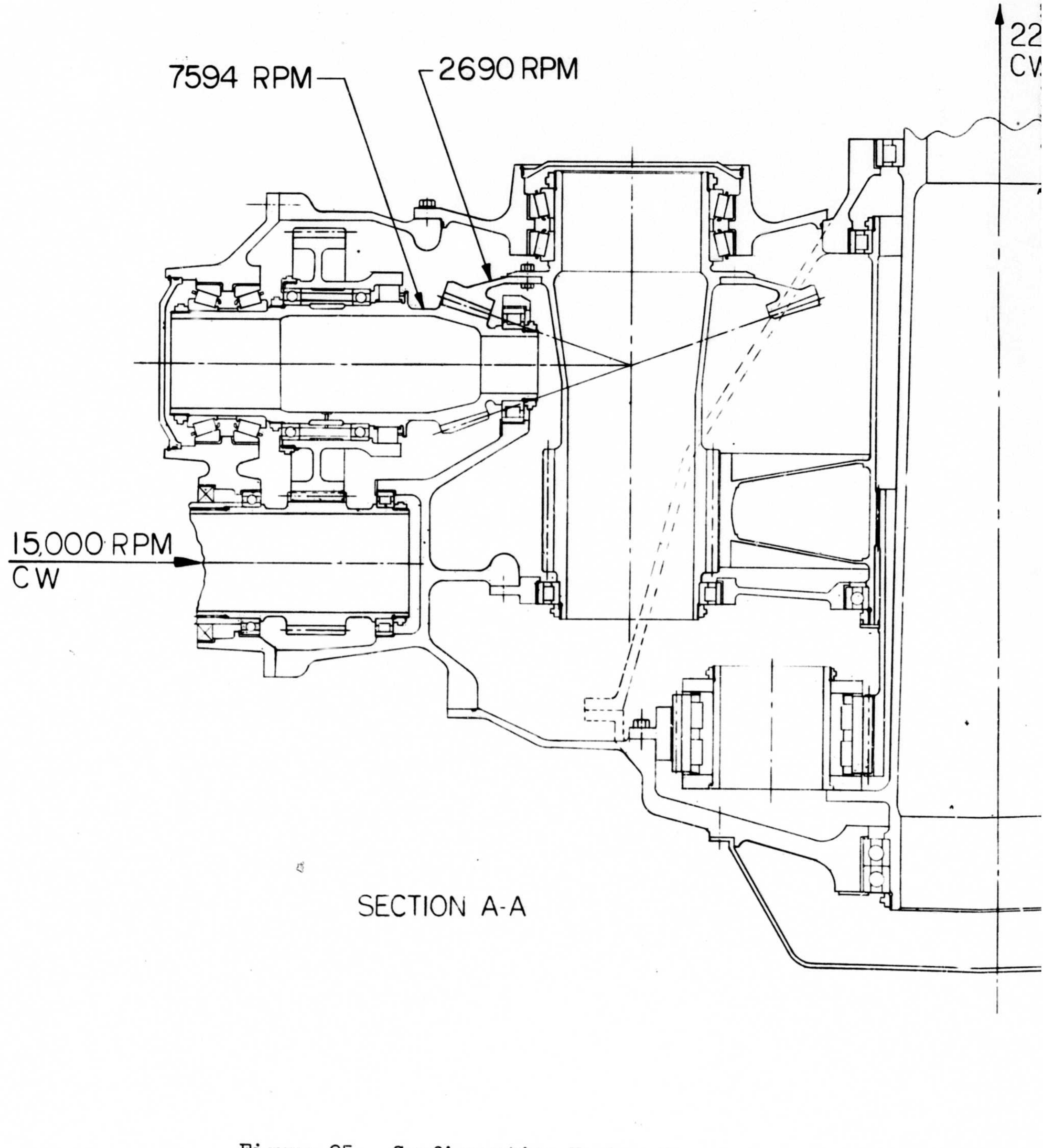
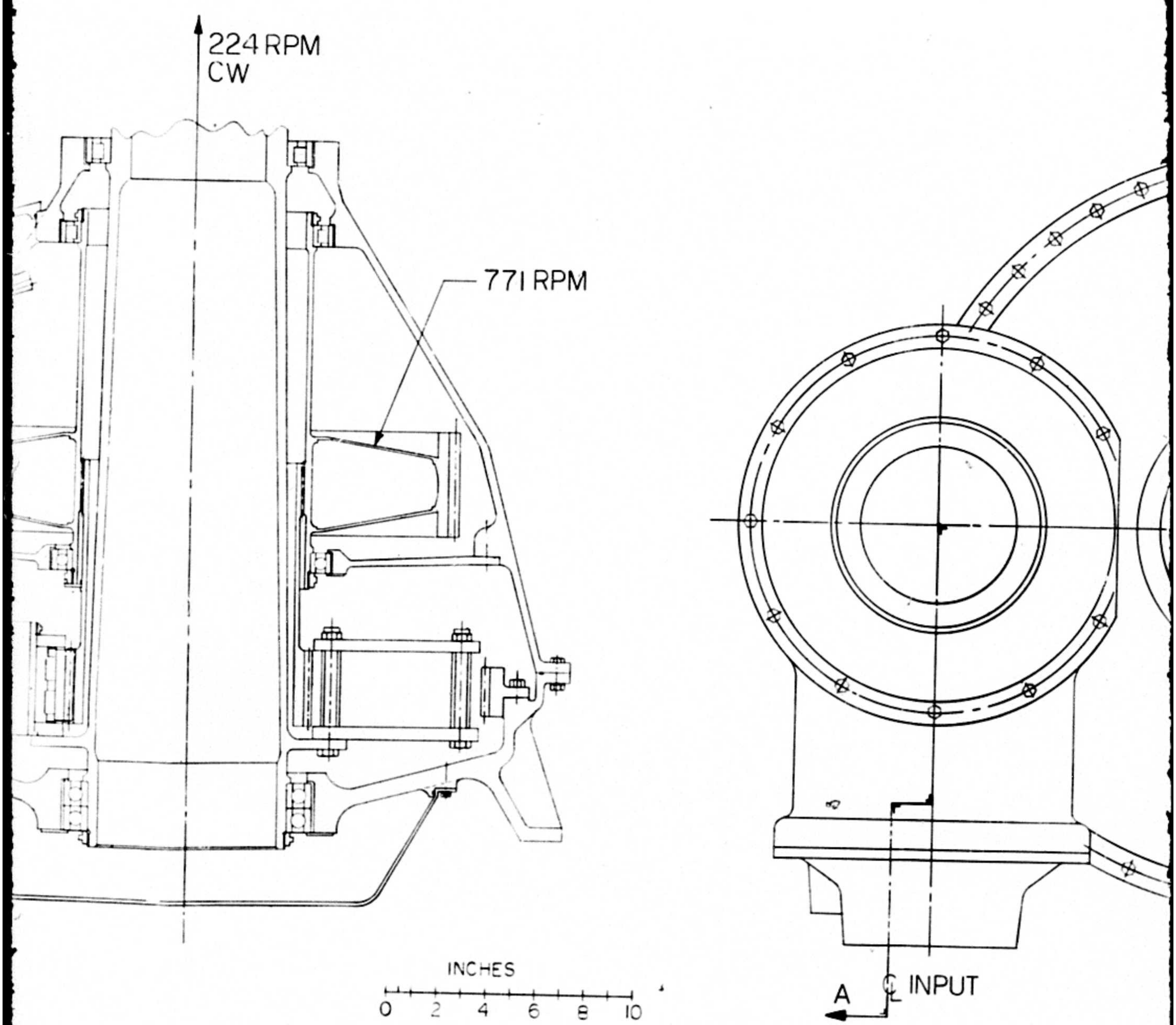
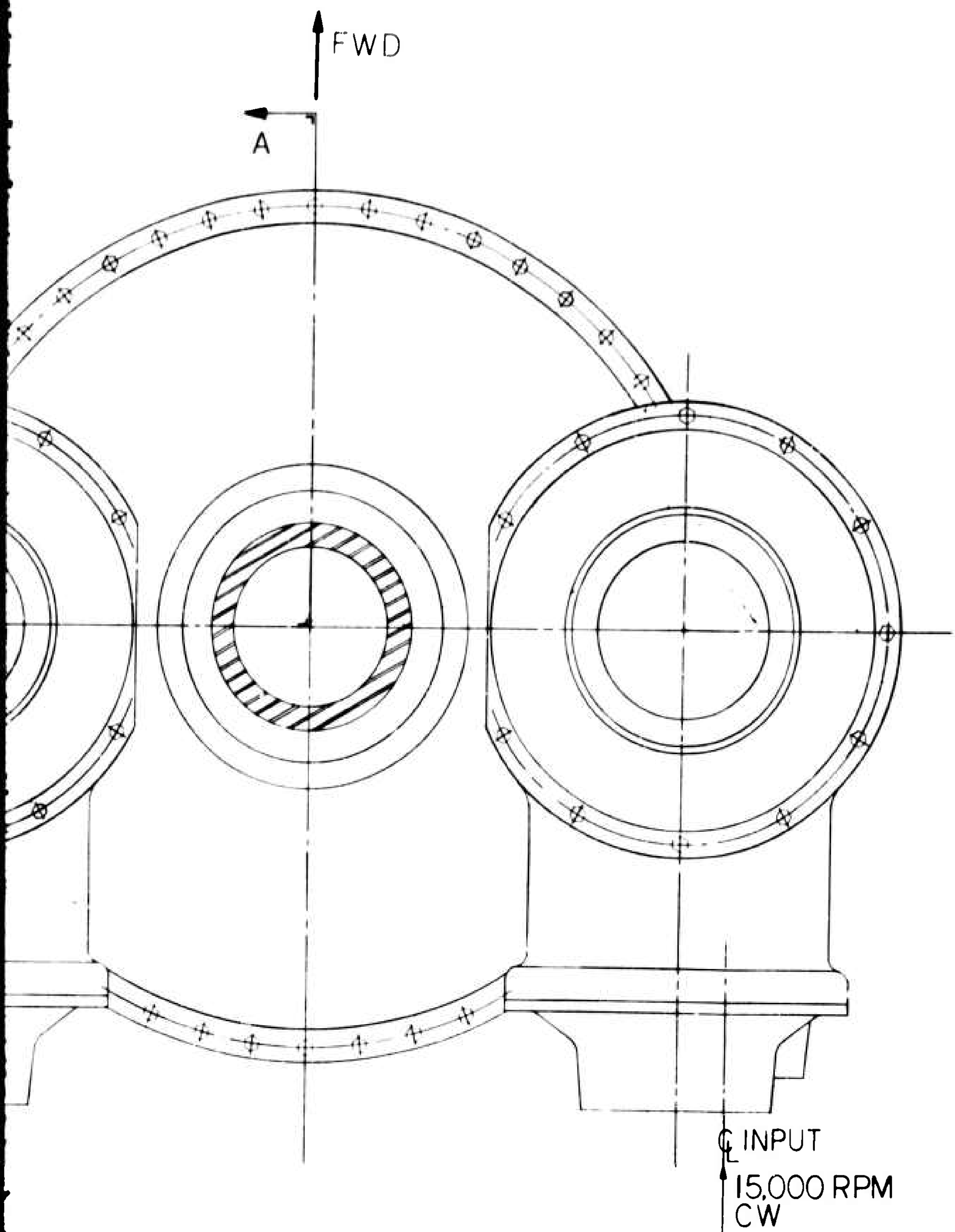


Figure 25. Configuration Number 3.

A



B



FWD
A
℄ INPUT
15,000 RPM
CW

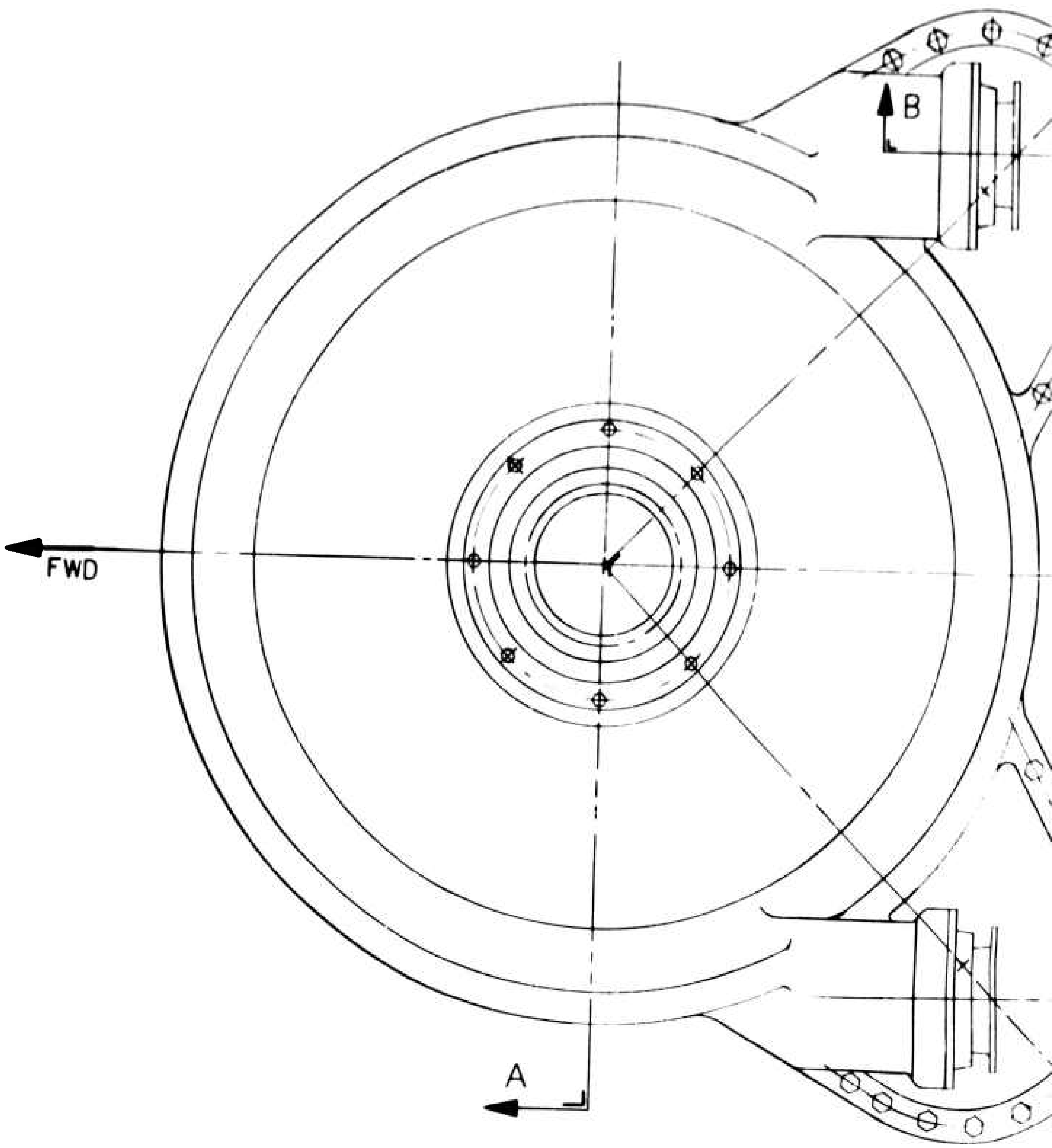
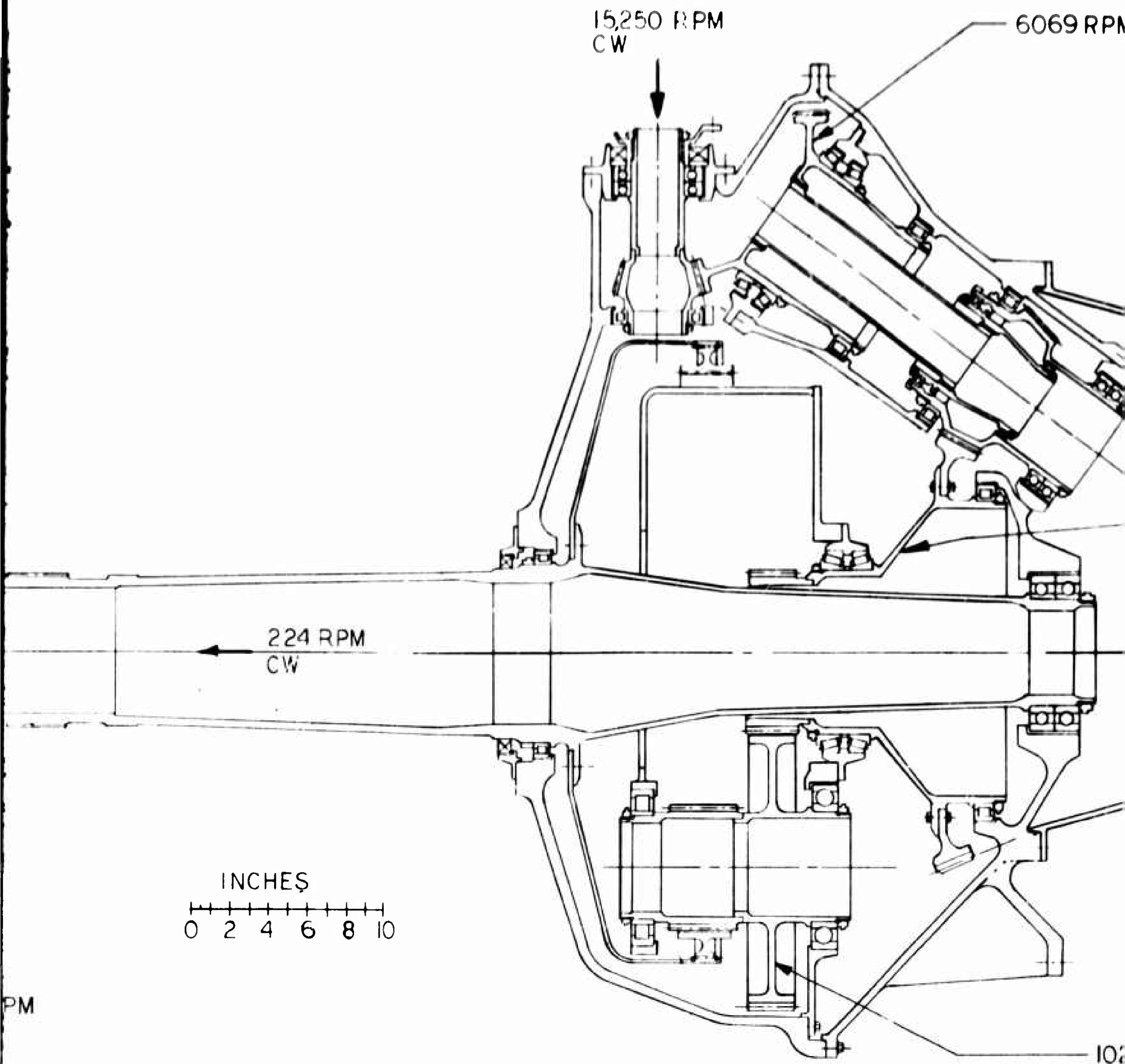
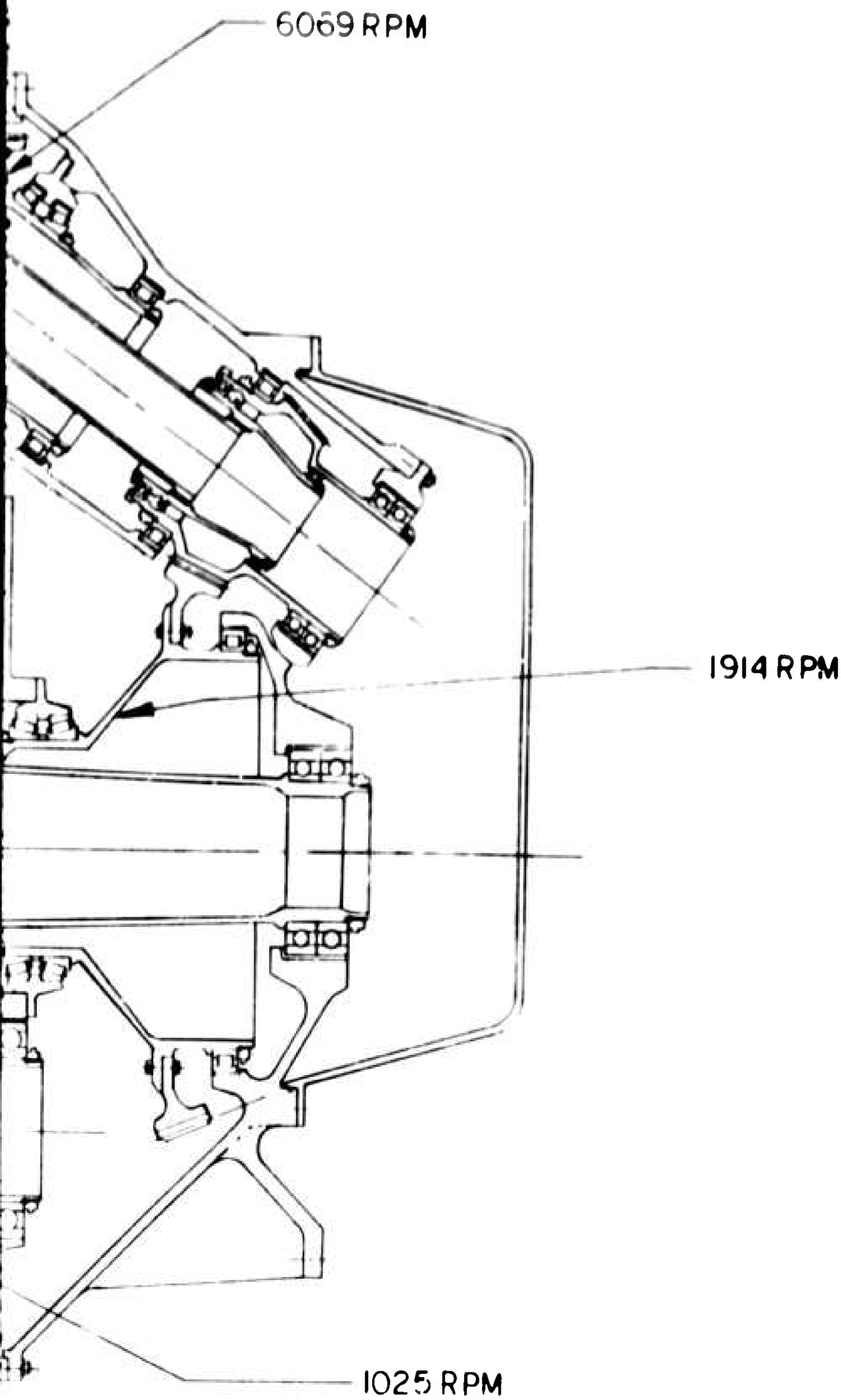


Figure 26. Configuration Number 4.



SECTION A-A & SECTION B-B
(SECTION B B ROTATED 43° CCW)

B



C

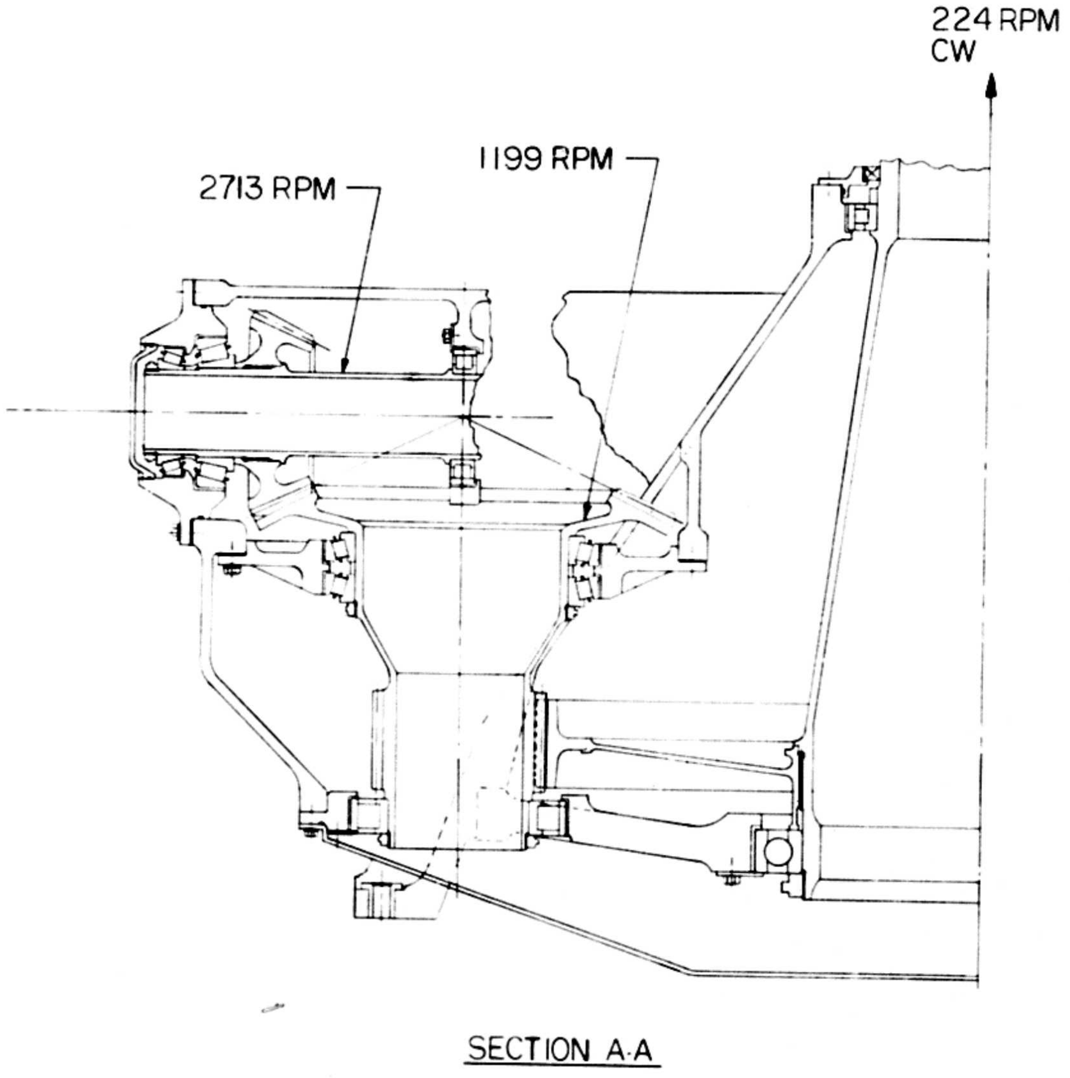
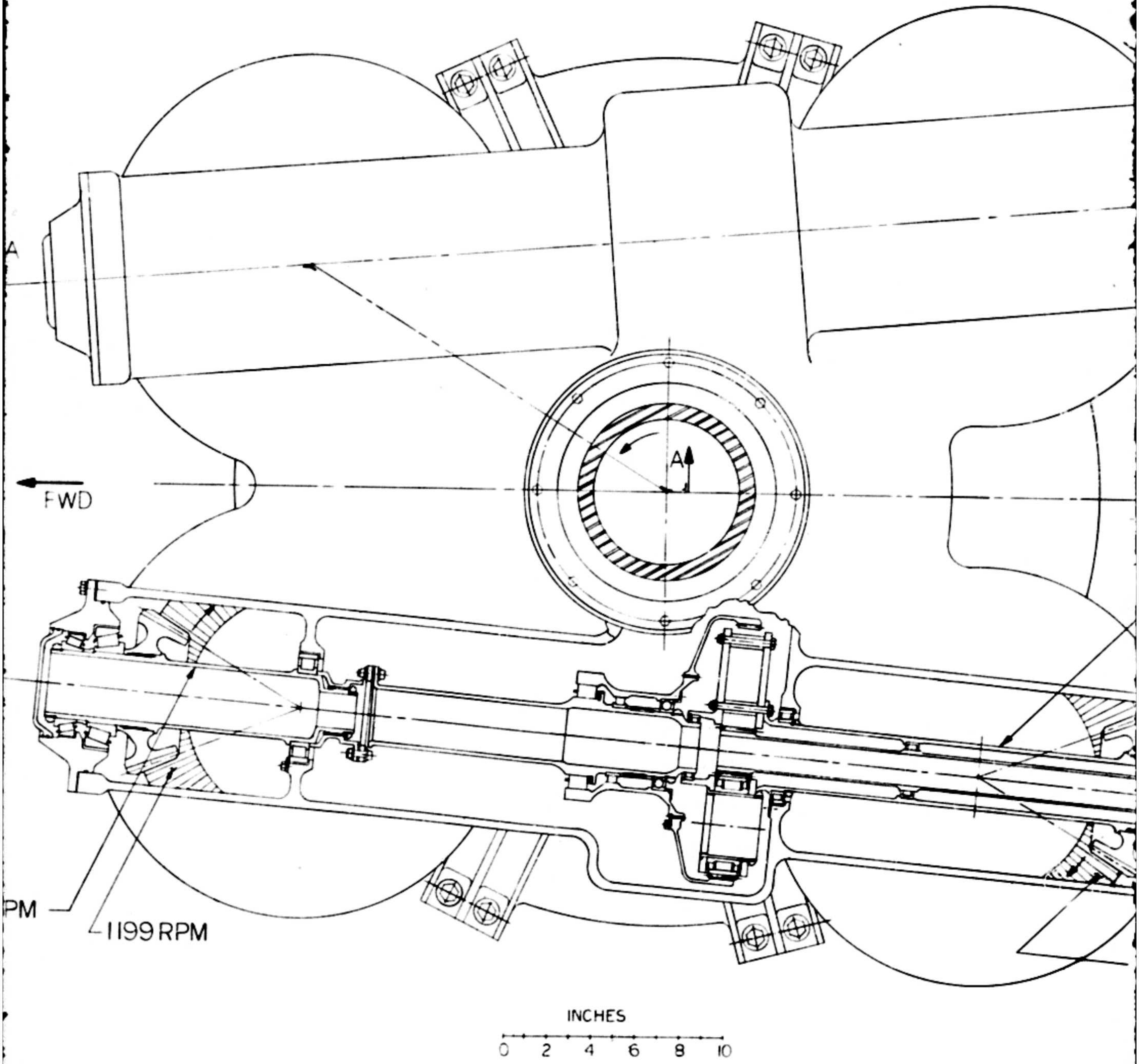
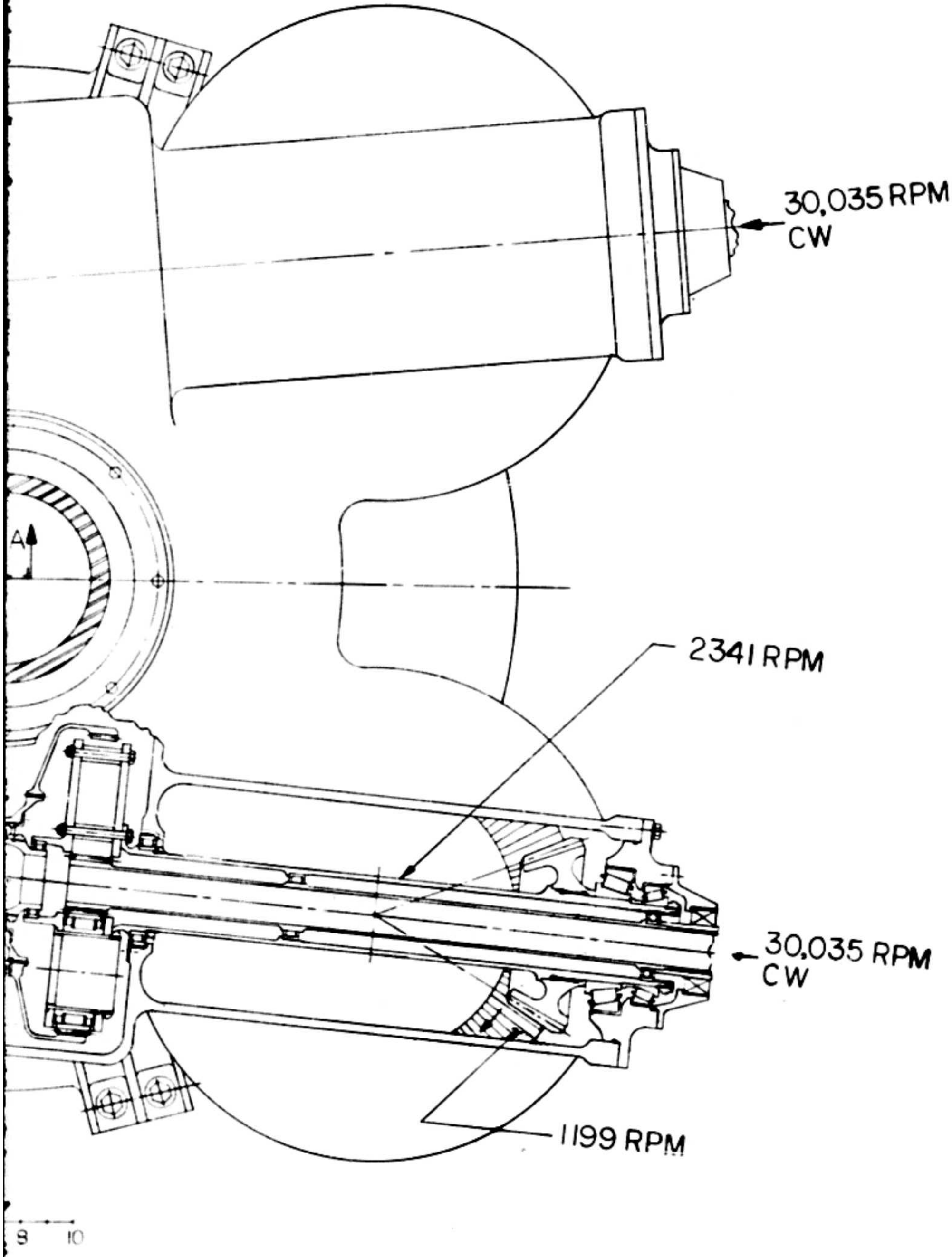


Figure 27. Configuration Number 5.

A



B


30,035 RPM
CW
A
2341 RPM
30,035 RPM
CW
1199 RPM
8
10

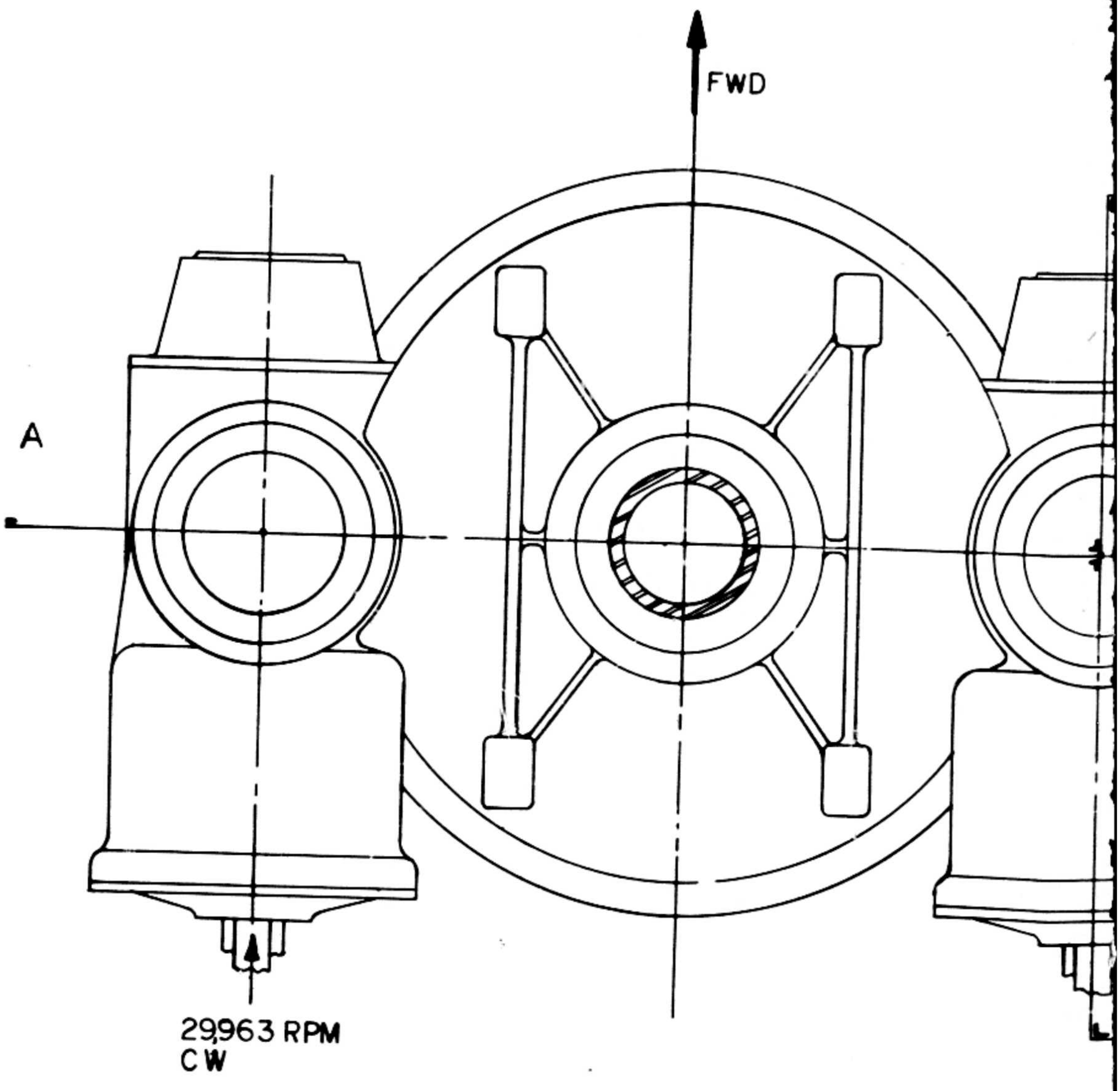
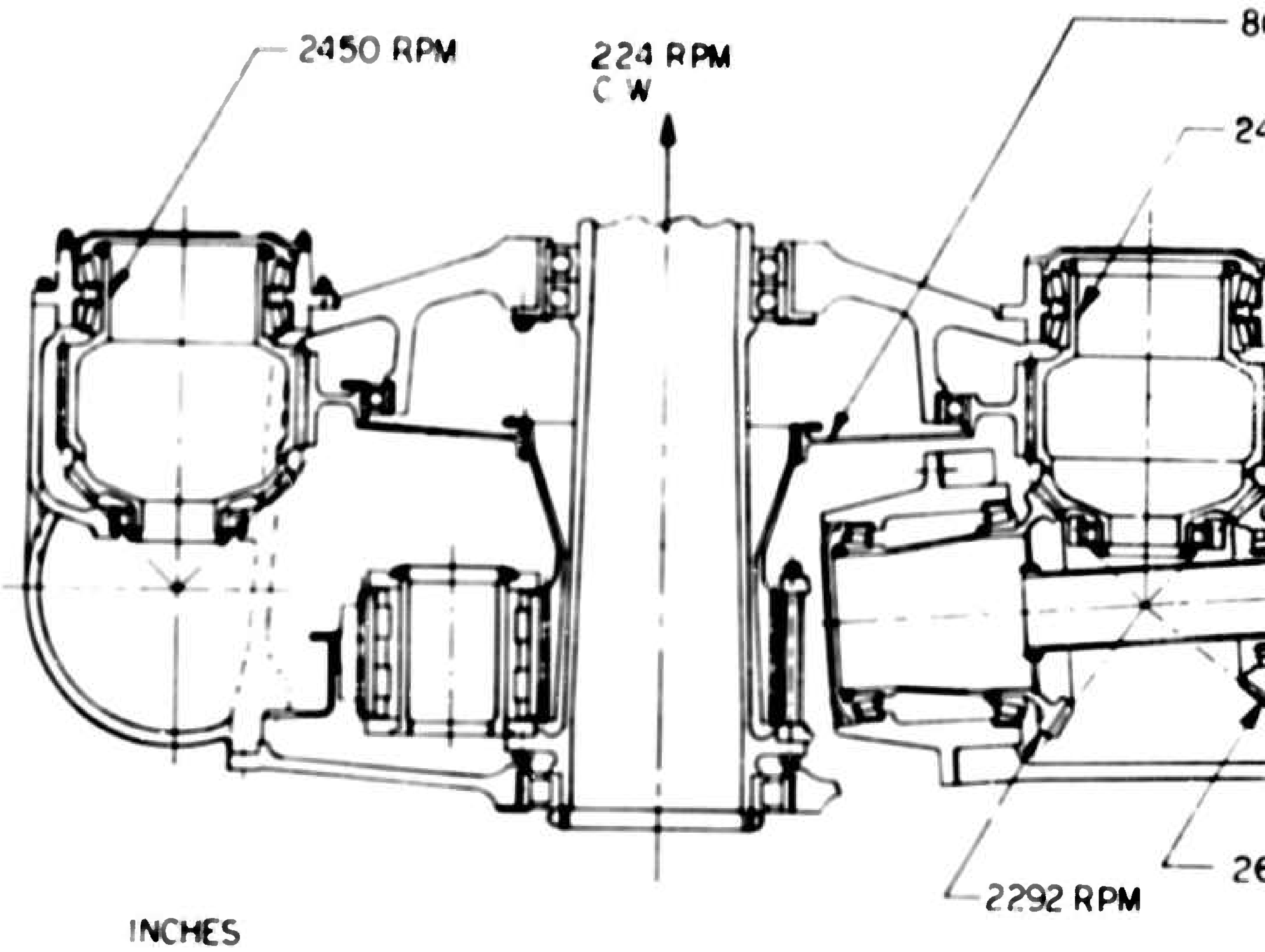


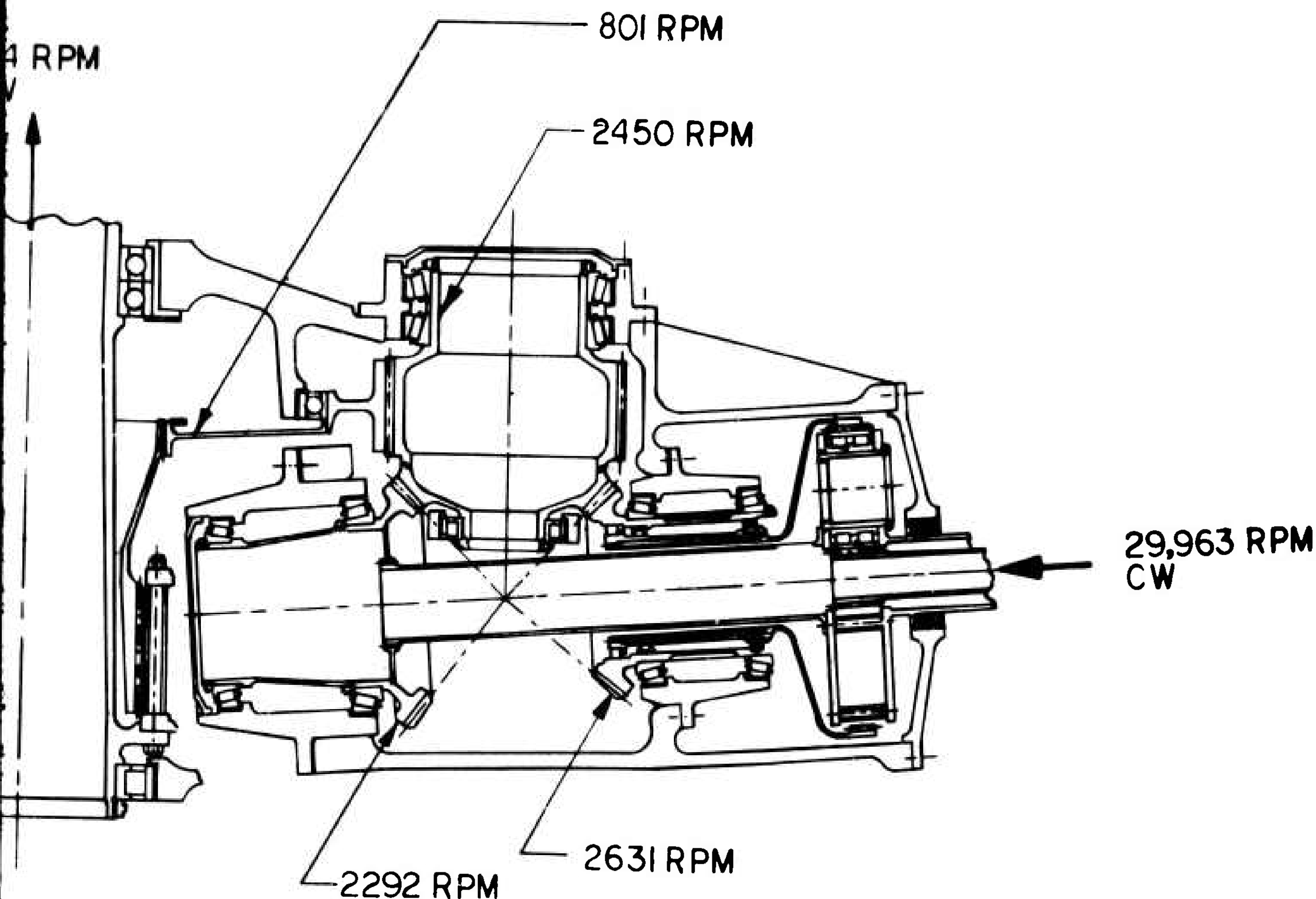
Figure 28. Configuration Number 6.

A



SECTION A-A & SECTION B-B
(SECTION B-B ROTATED 90° CCW)

B



SECTION A-A & SECTION B-B
(SECTION B-B ROTATED 90° CCW)

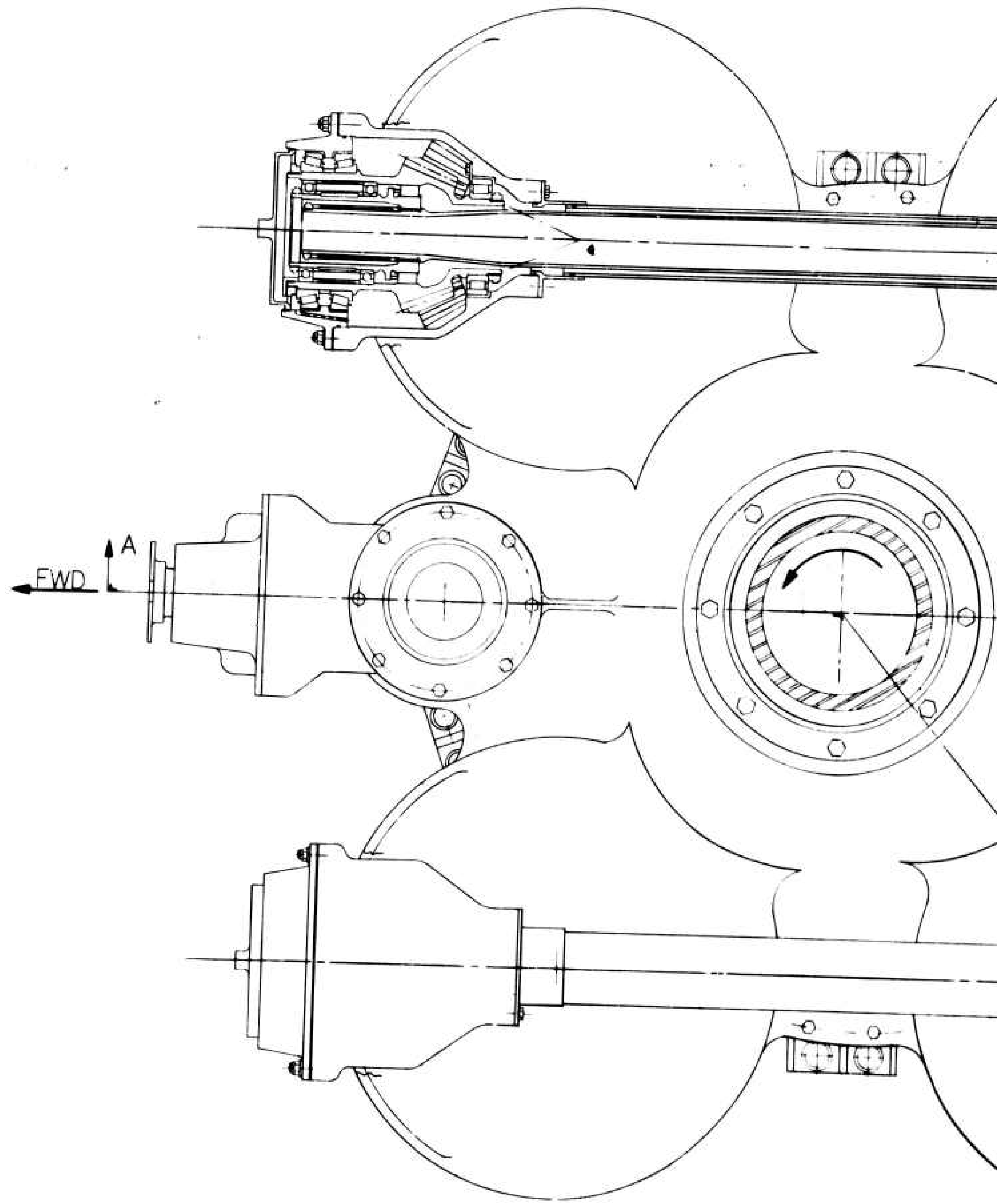
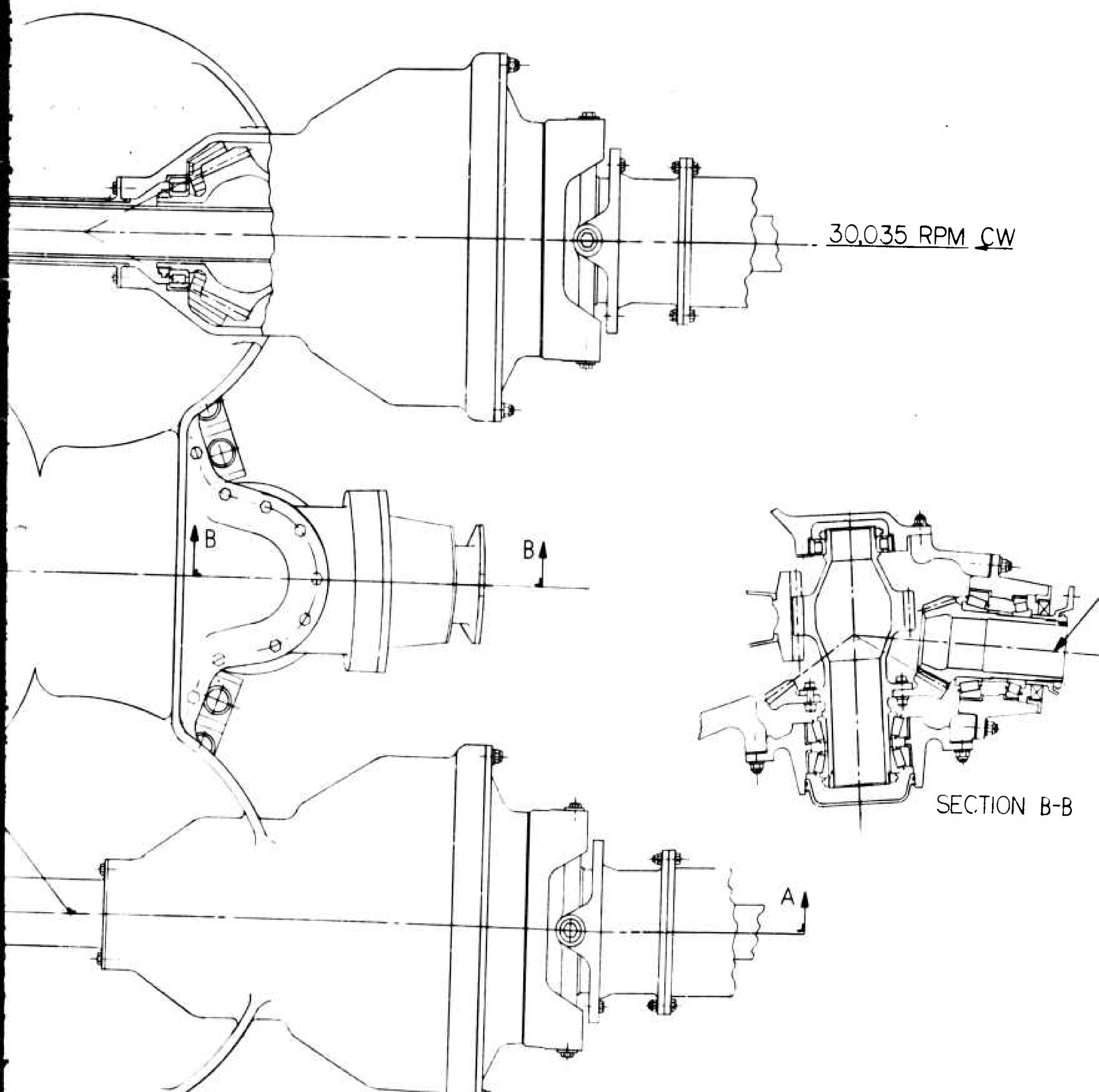
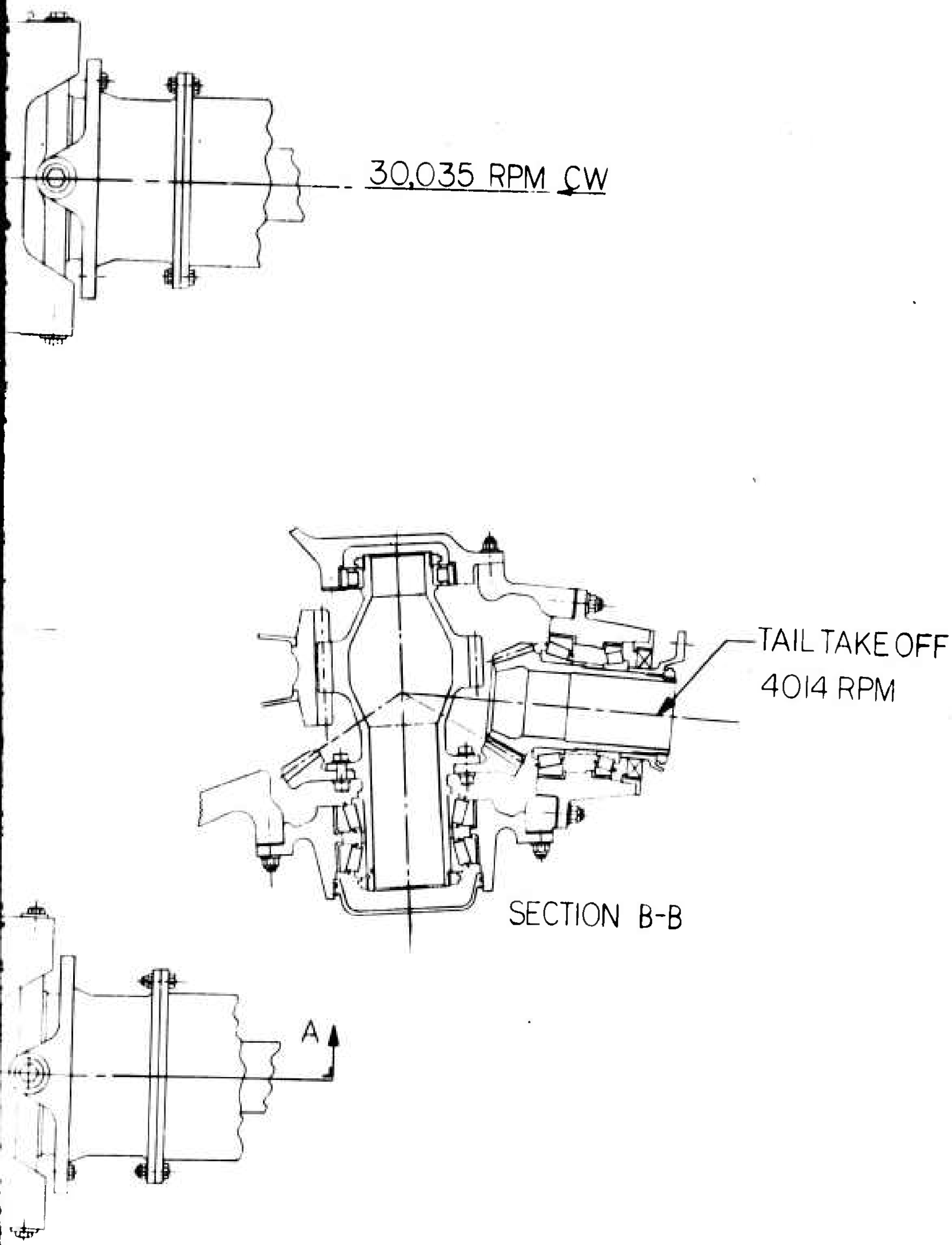


Figure 29. Main Gearbox, Pure Helicopter.

A



13



e

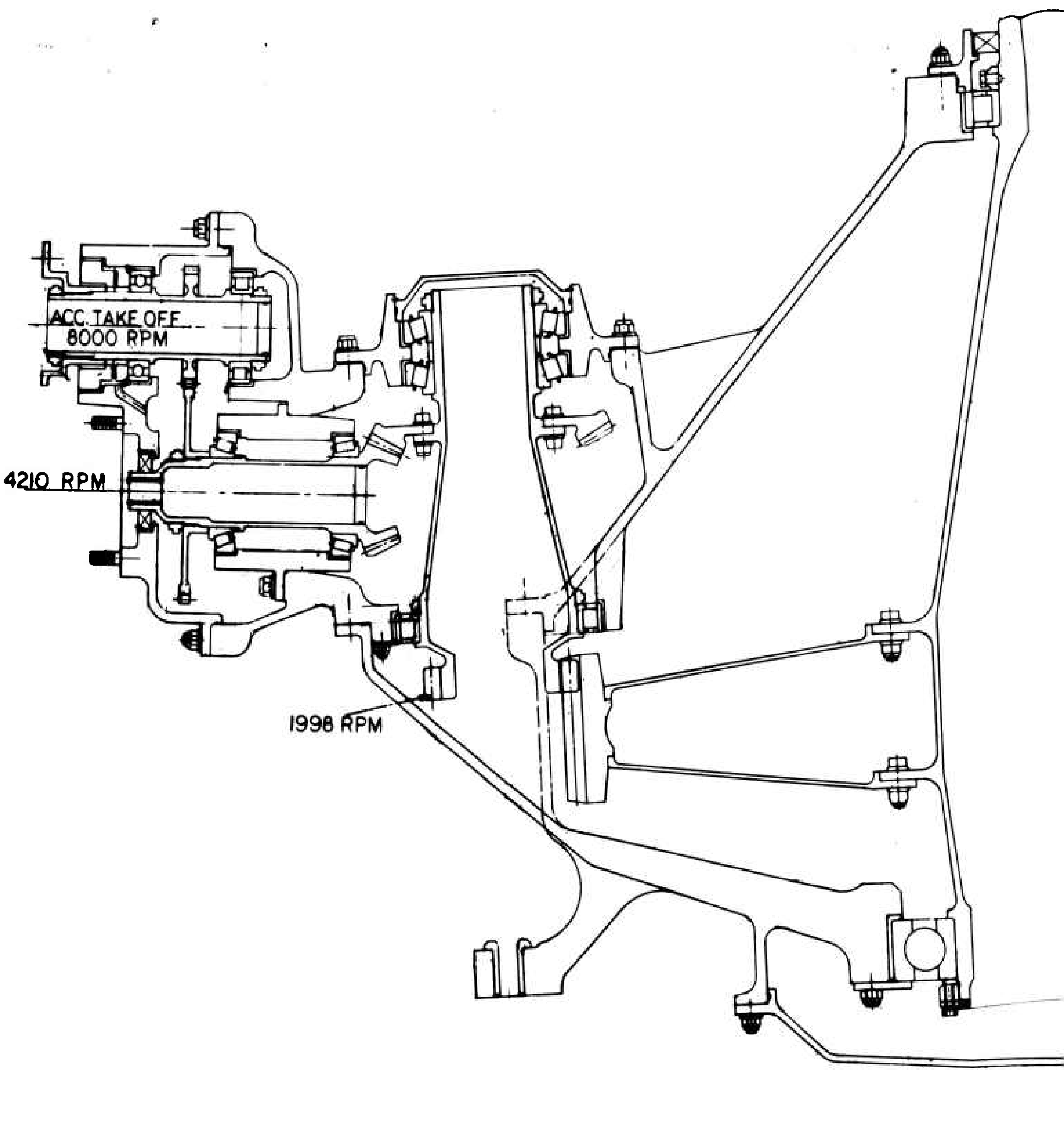
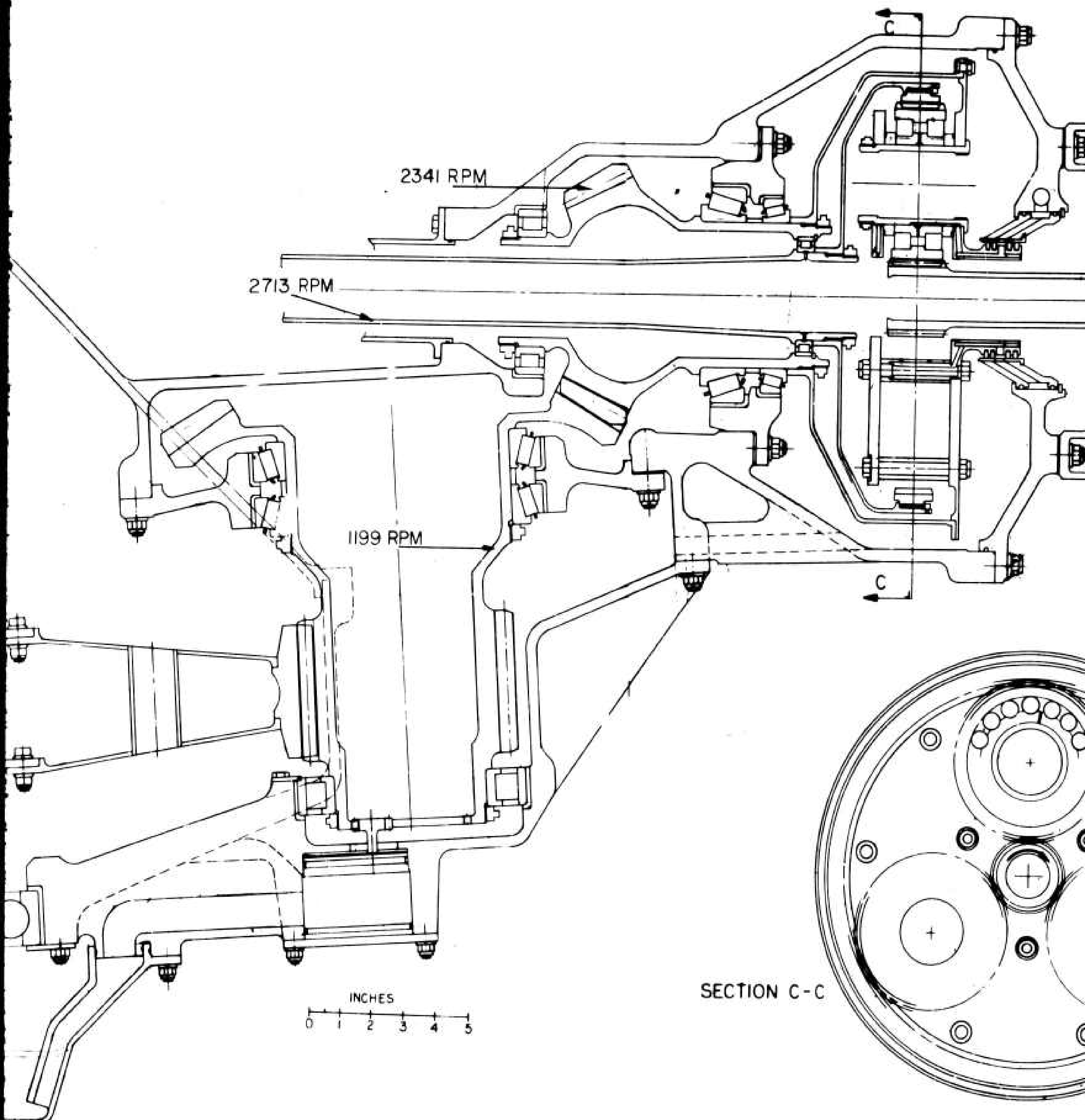
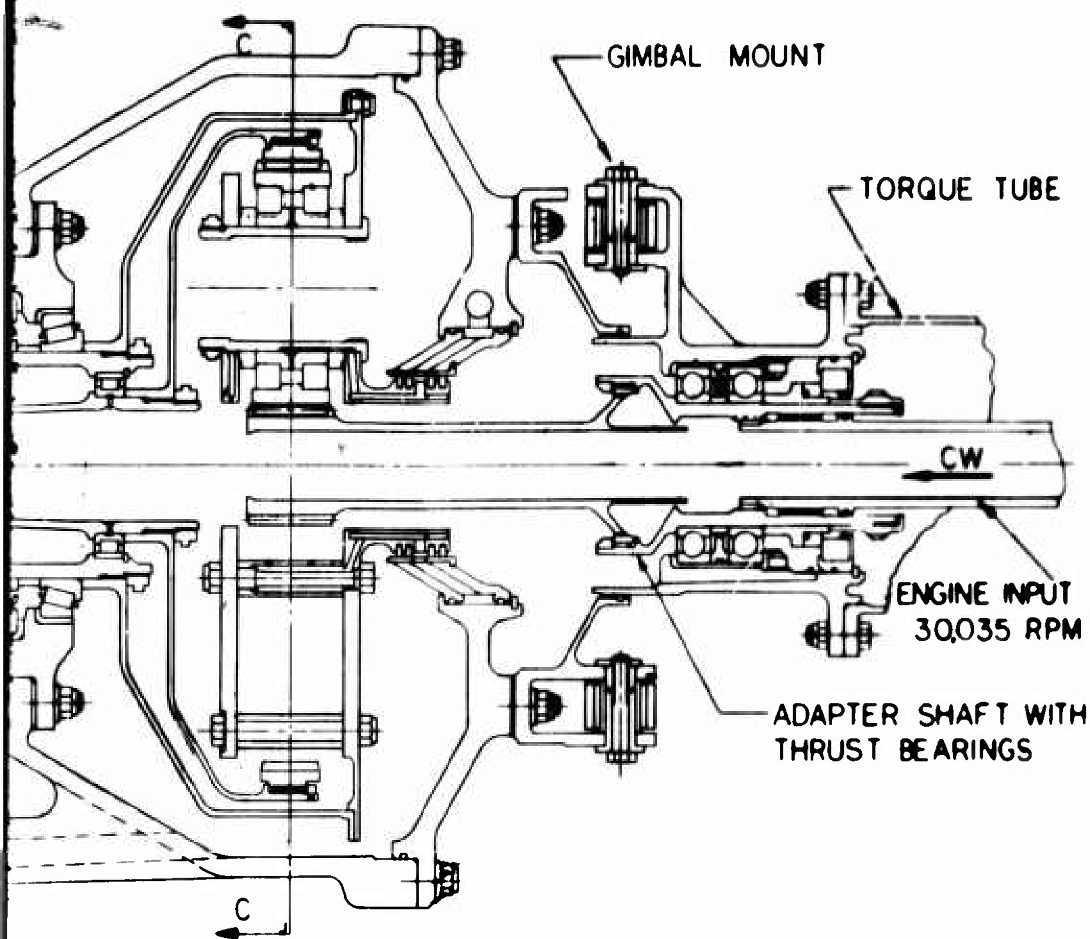


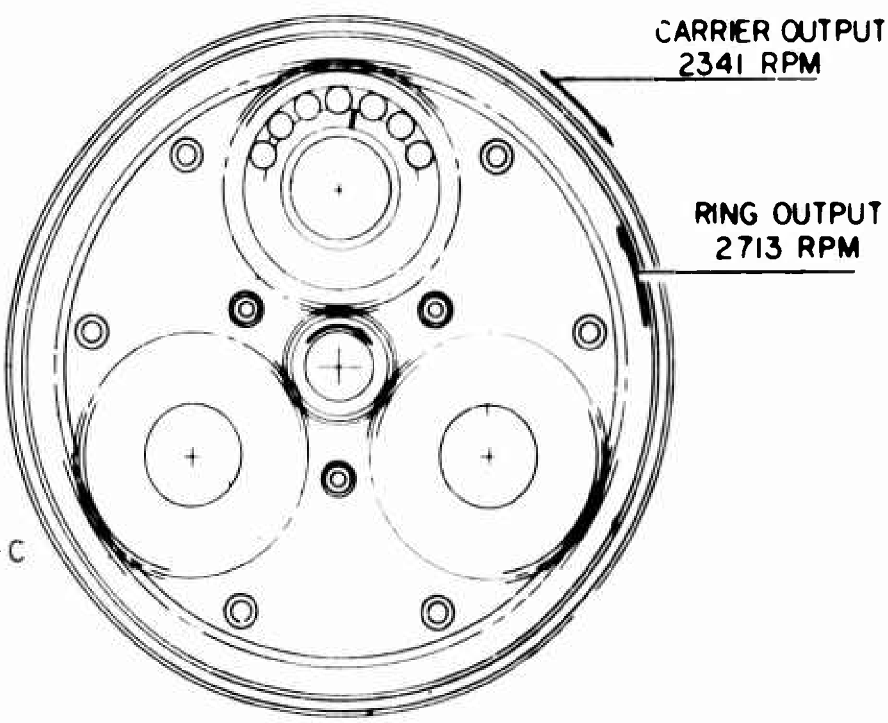
Figure 30. Section A-A, Main Gearbox, Pure Helicopter.

A



B


C
GIMBAL MOUNT
TORQUE TUBE
CW
ENGINE INPUT
30,035 RPM
ADAPTER SHAFT WITH
THRUST BEARINGS
C


CARRIER OUTPUT
2341 RPM
RING OUTPUT
2713 RPM

CTION C-C

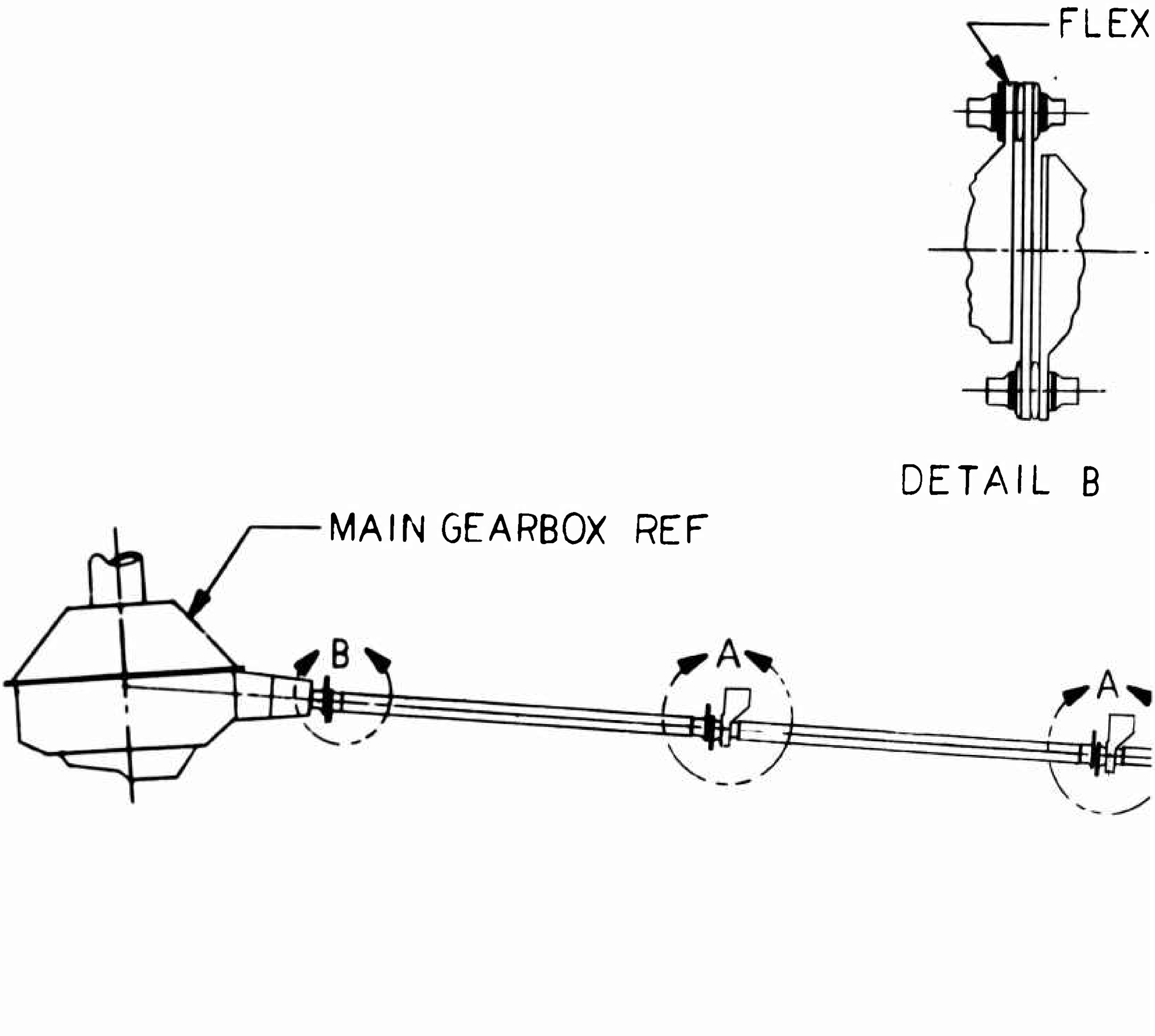
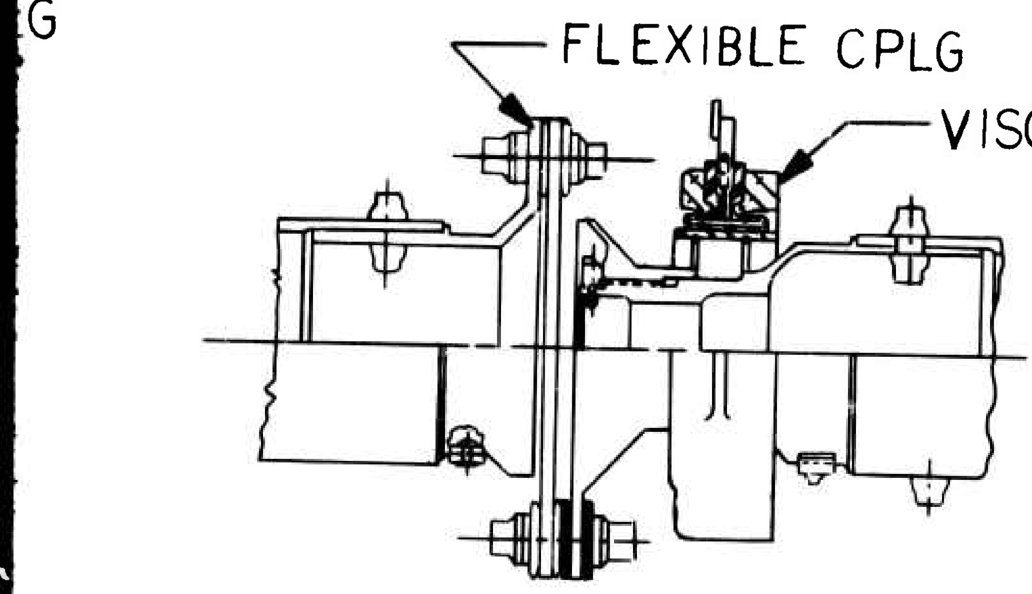
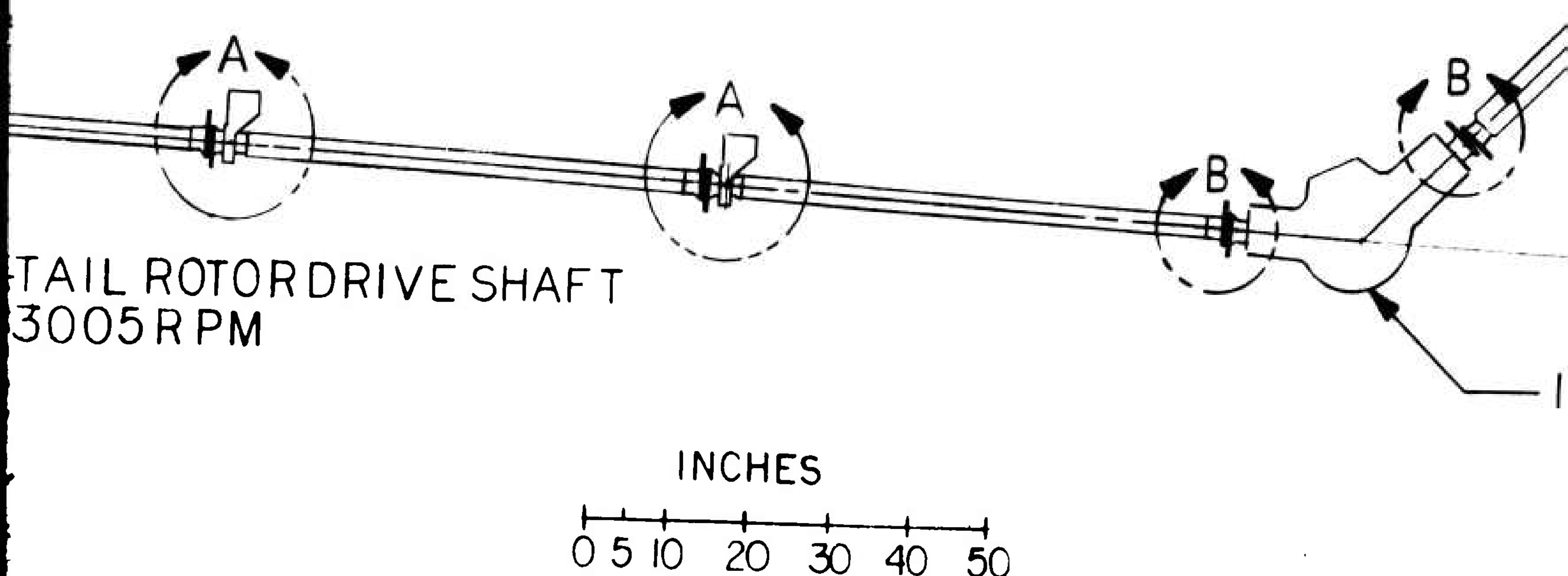


Figure 31. Tail Rotor Drive System, Pure Helicopter.



G
FLEXIBLE CPLG
VISCOUS DAMPED BEARING SU

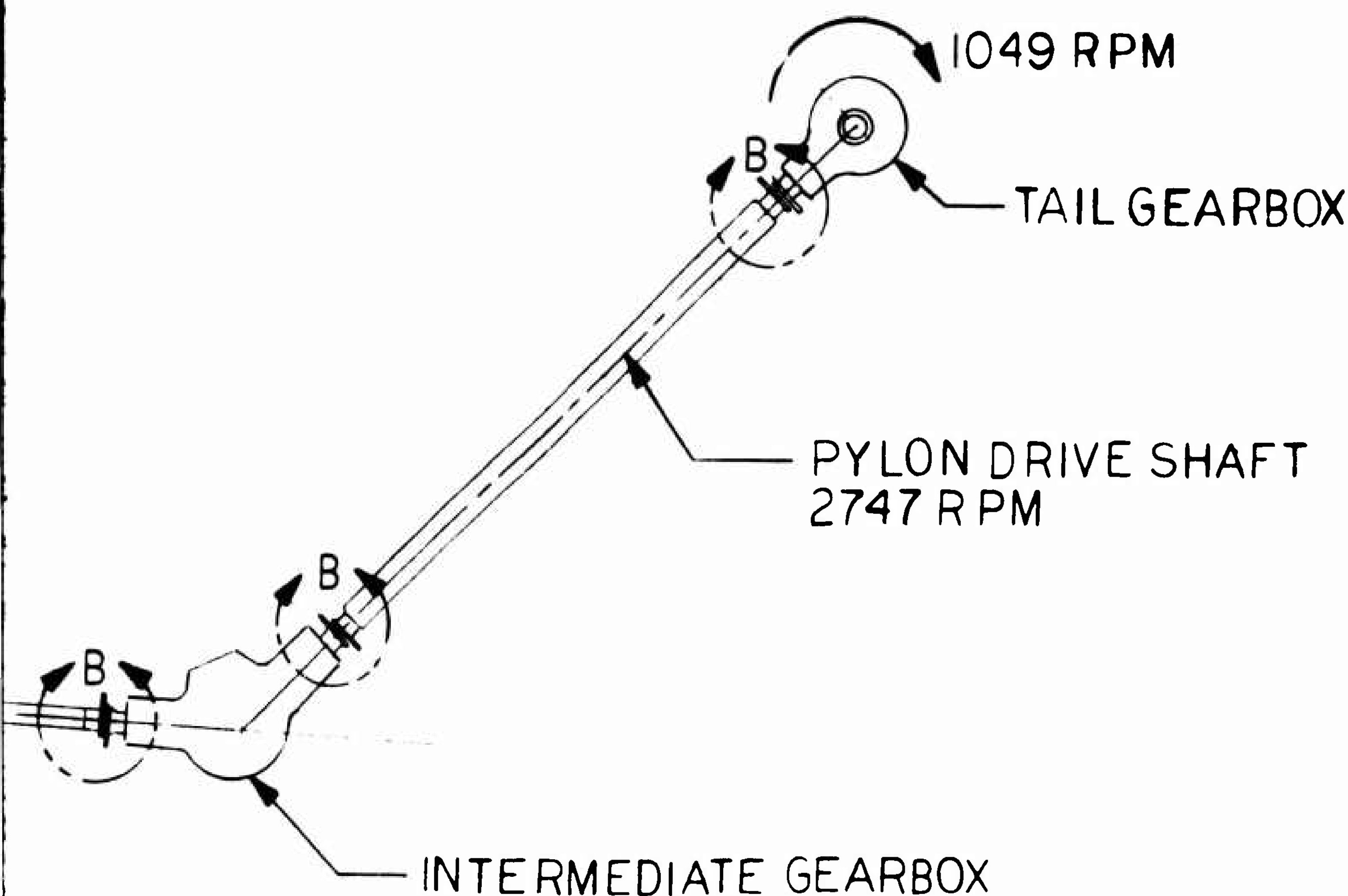
DETAIL A



A
A
B
B
TAIL ROTOR DRIVE SHAFT
3005 RPM
INCHES
0 5 10 20 30 40 50

ℬ

AMPED BEARING SUPPORT



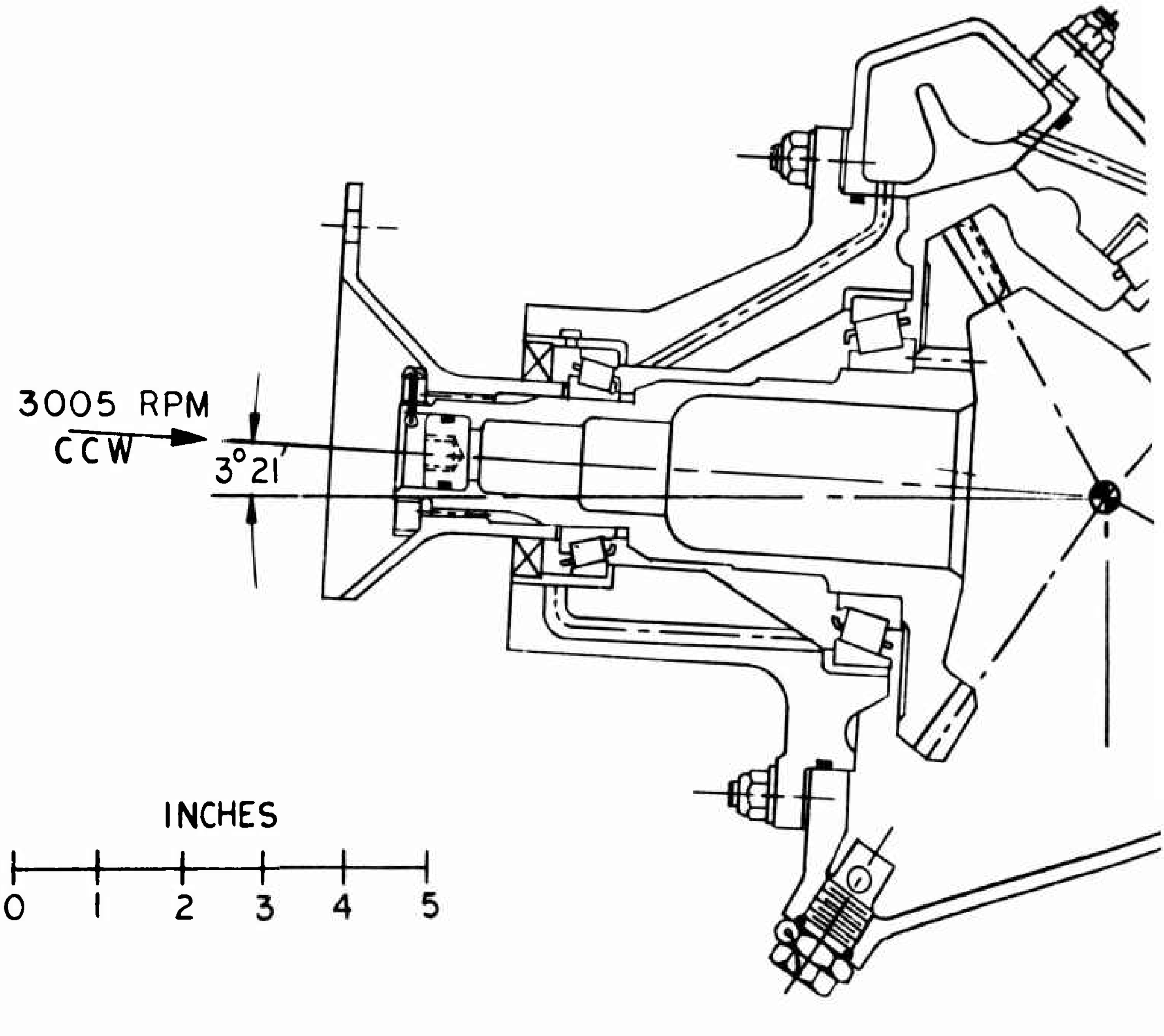
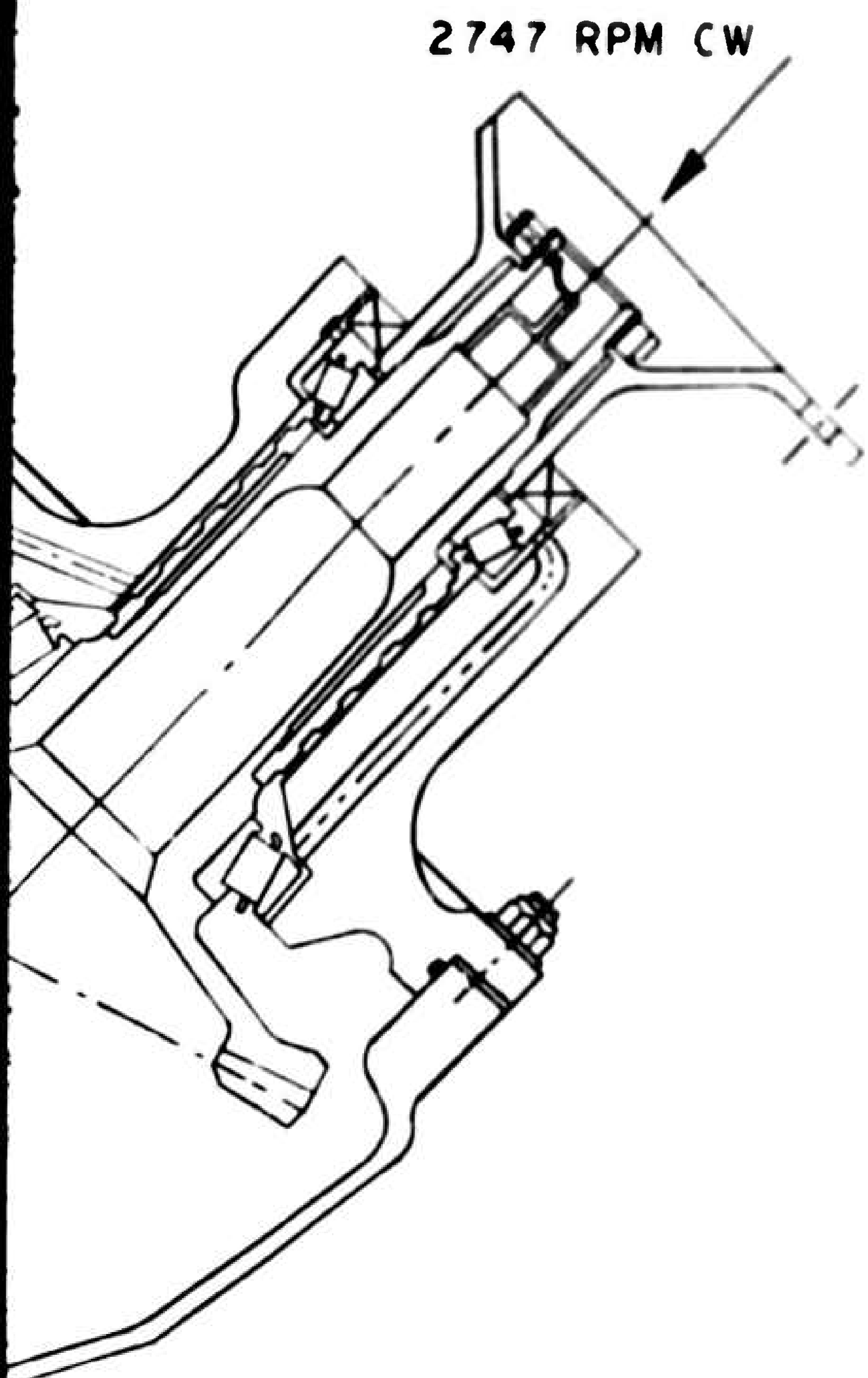


Figure 32. Intermediate Gearbox, Pure Helicopter.

A



B

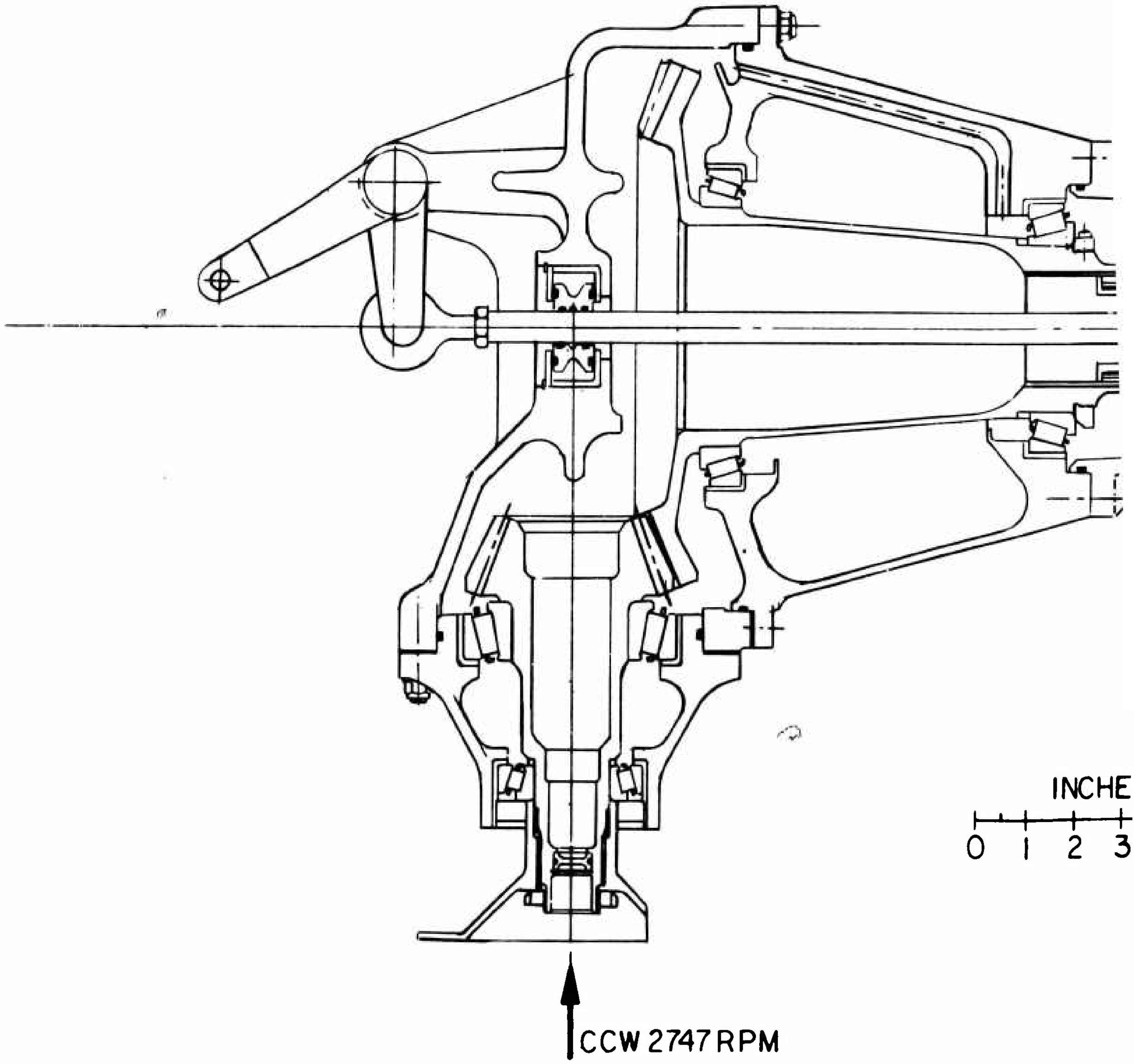
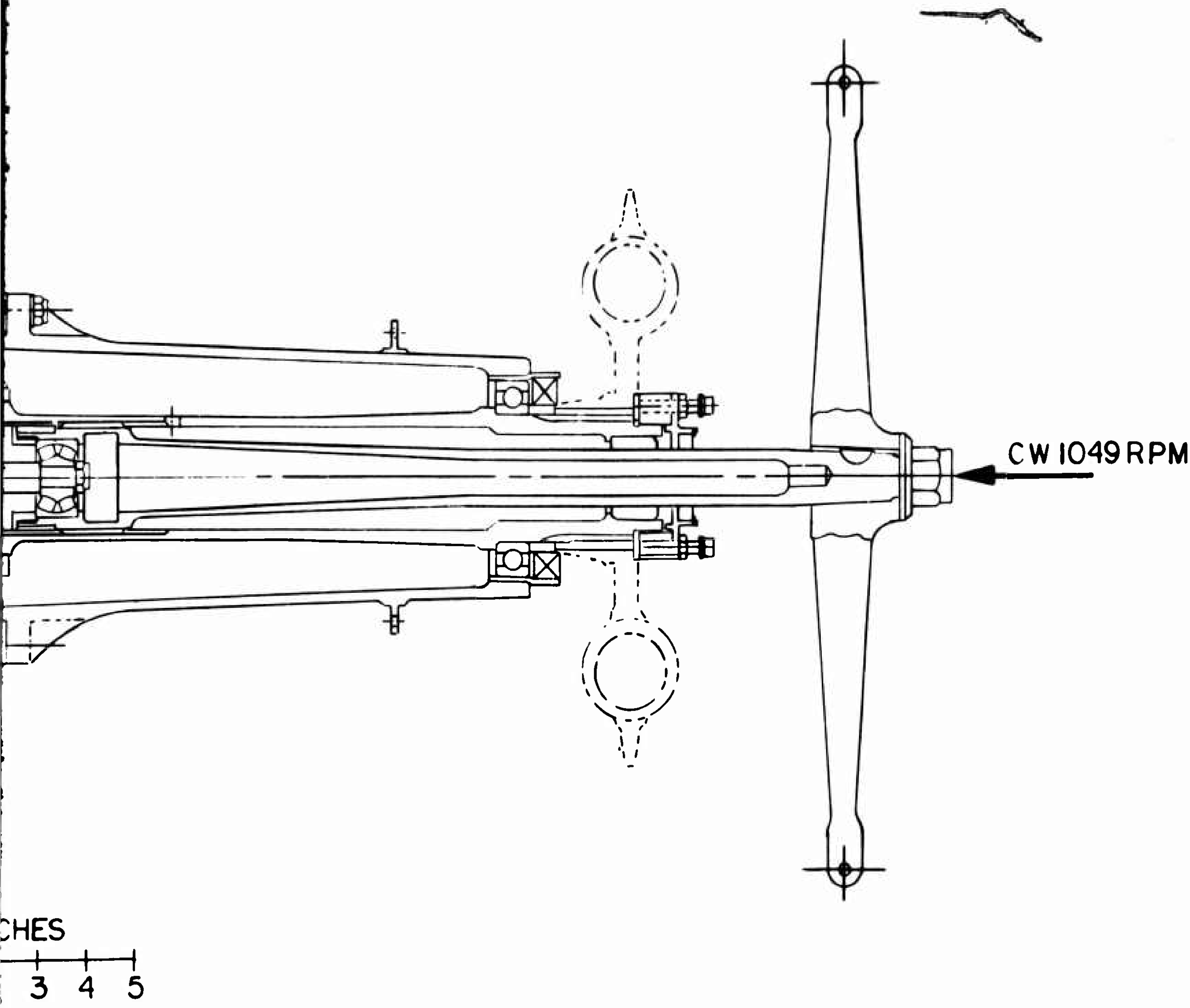


Figure 33. Tail Rotor Gearbox, Pure Helicopter.

A



B

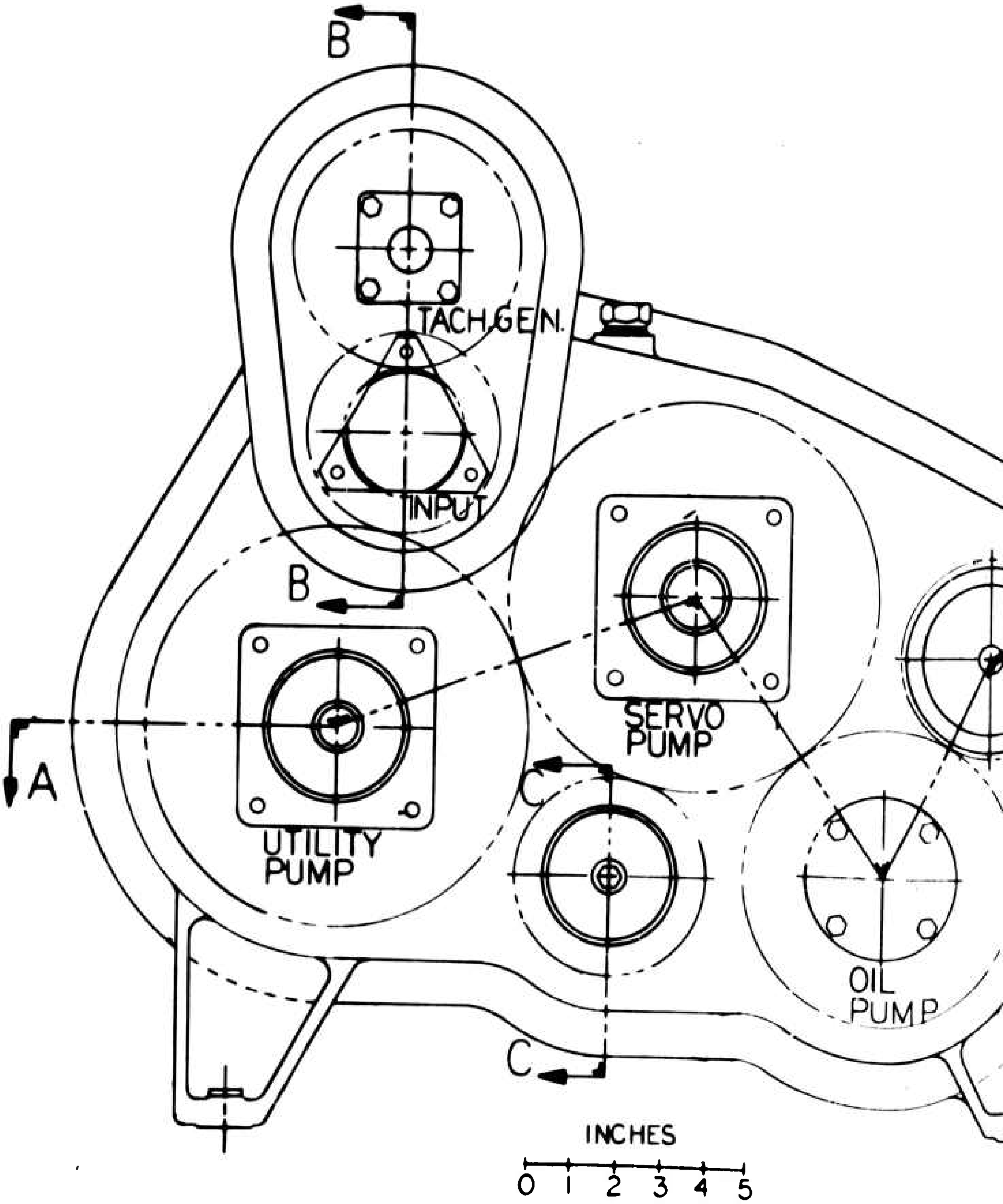
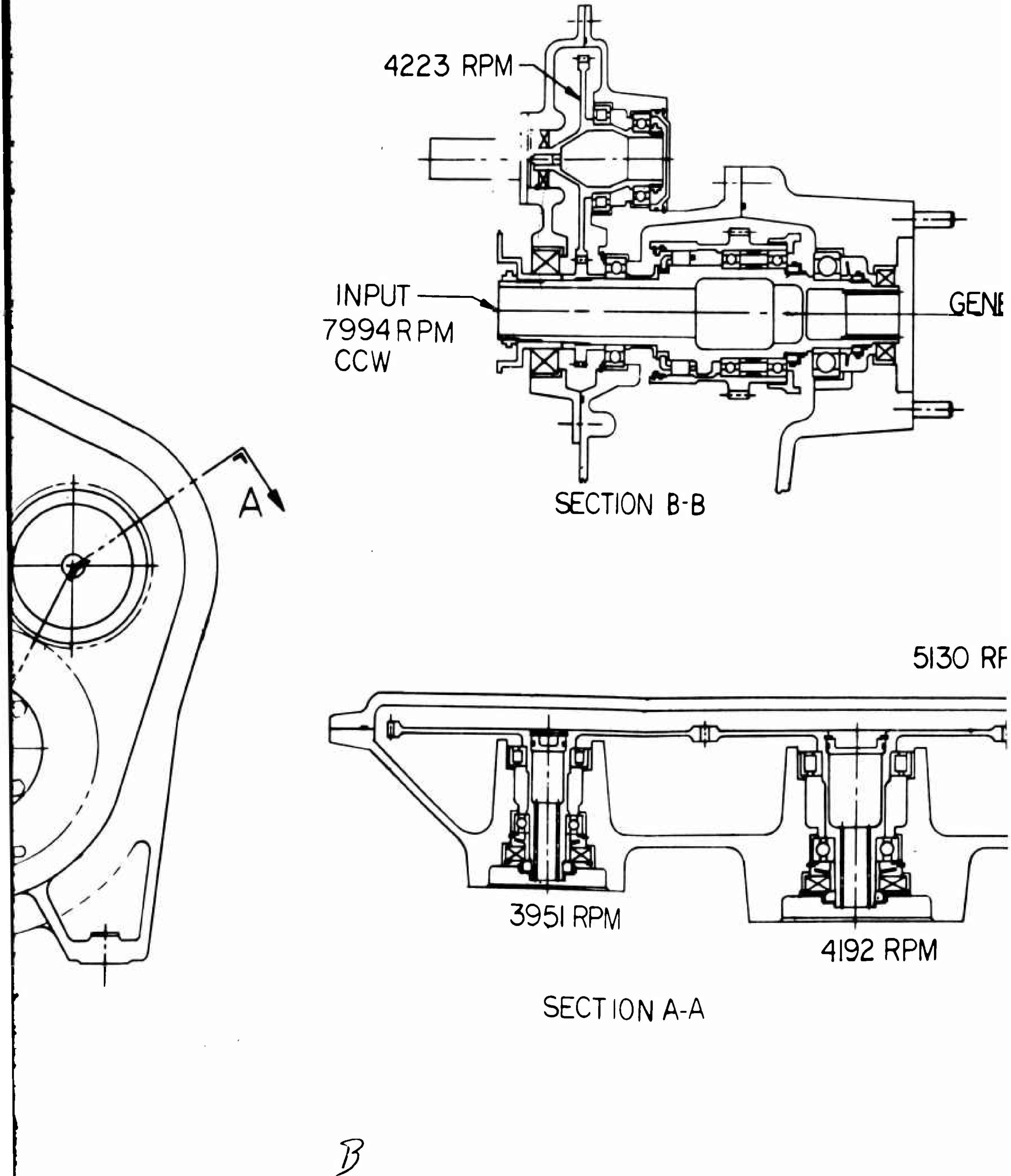
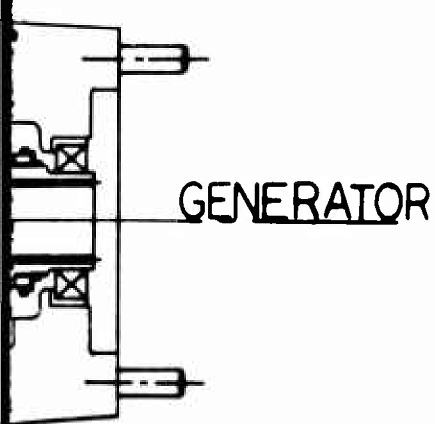


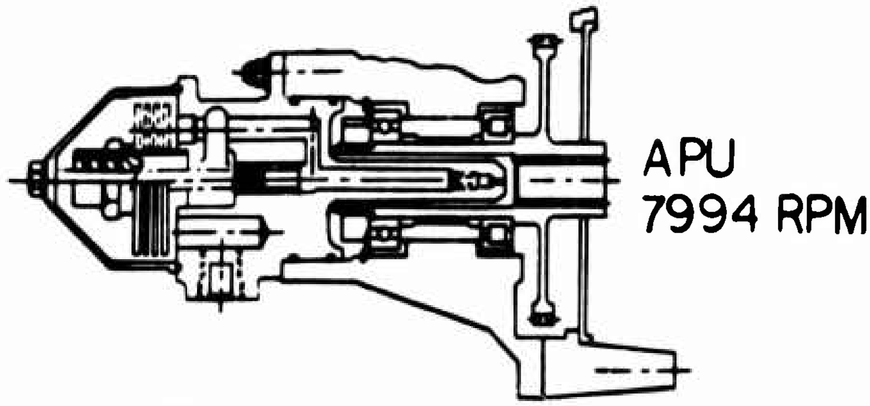
Figure 34. Accessory Gearbox, Pure Helicopter.

A

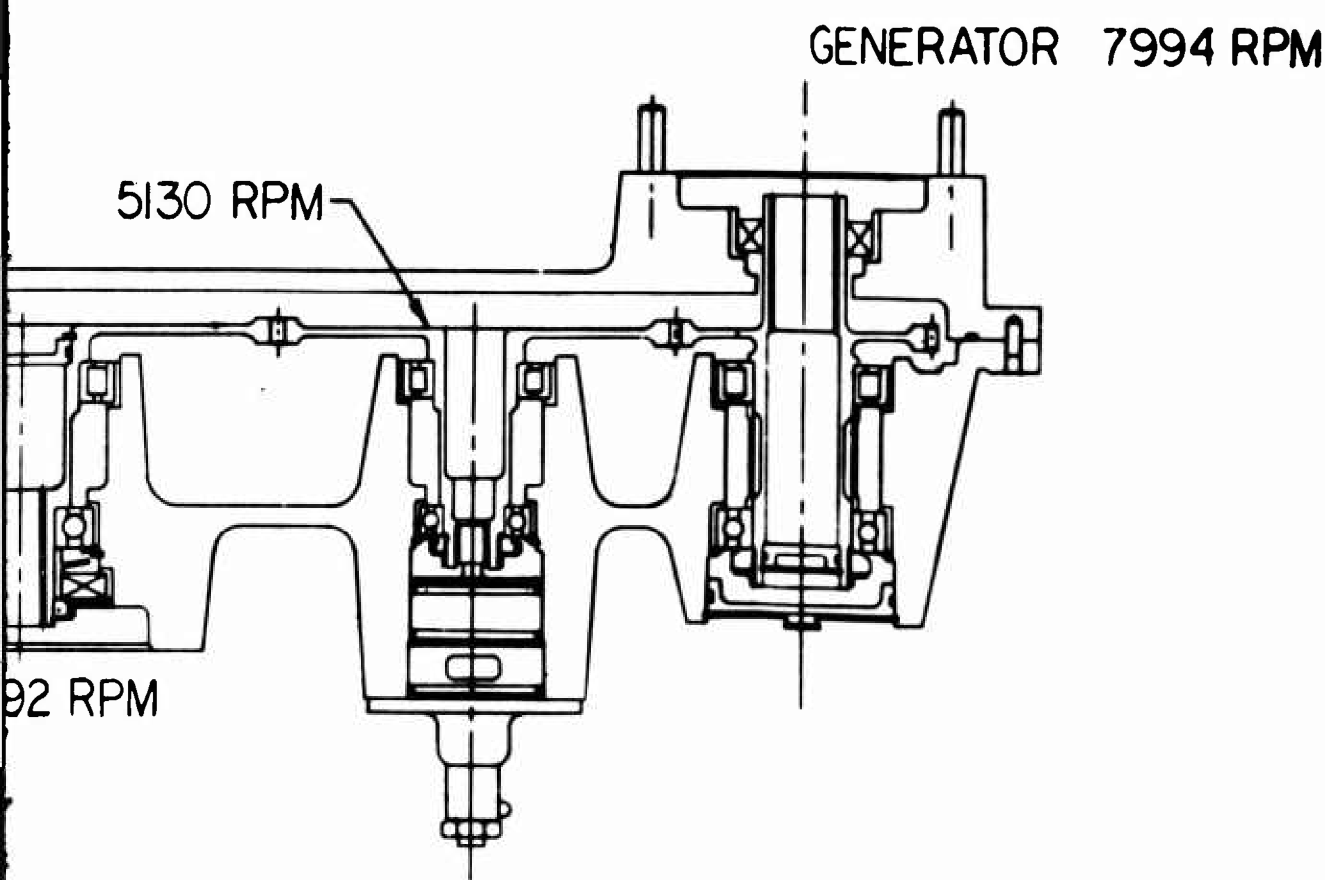


4223 RPM
INPUT
7994 RPM
CCW
SECTION B-B
A
5130 RF
3951 RPM
4192 RPM
SECTION A-A
B





SECTION C-C



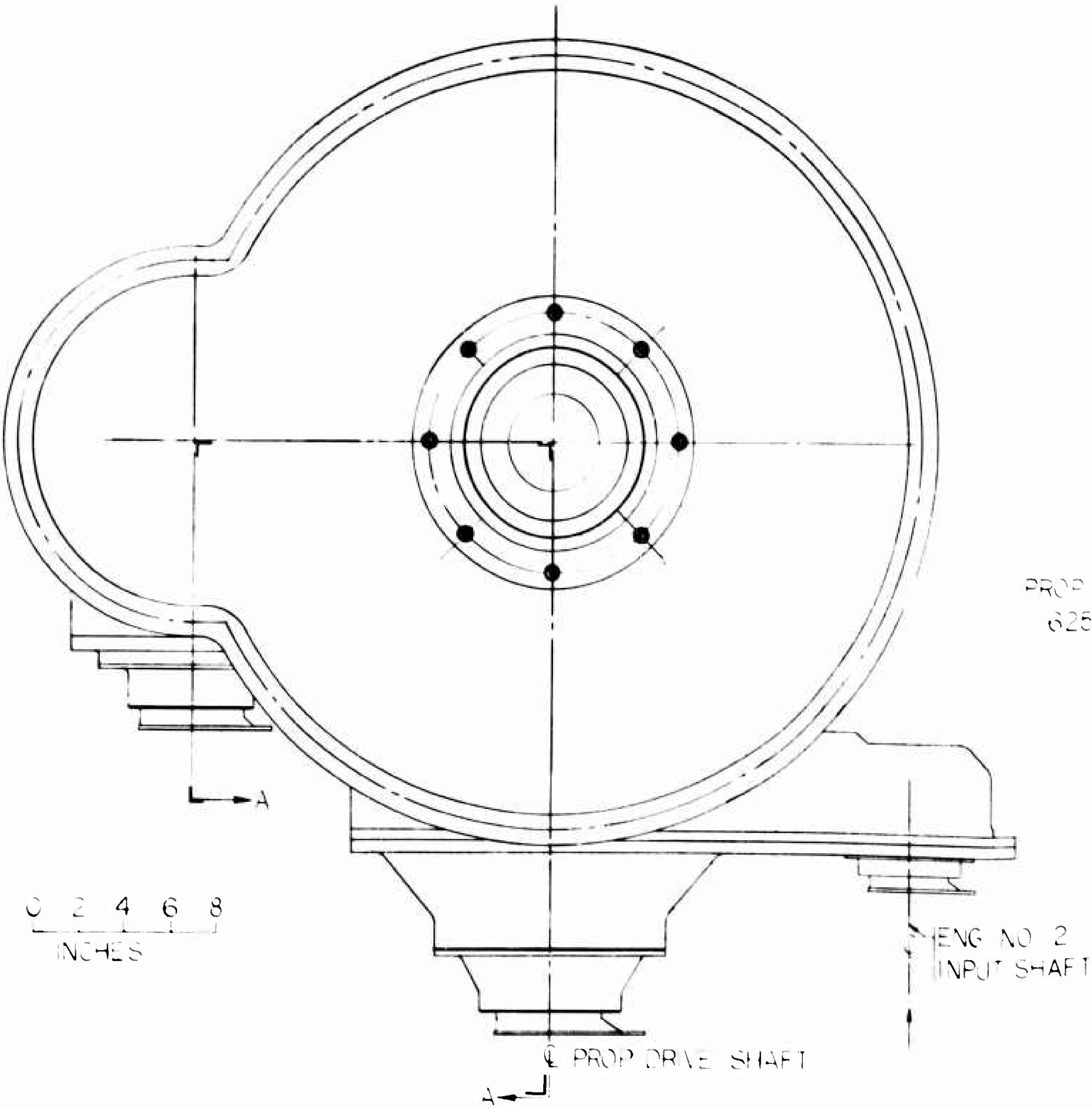
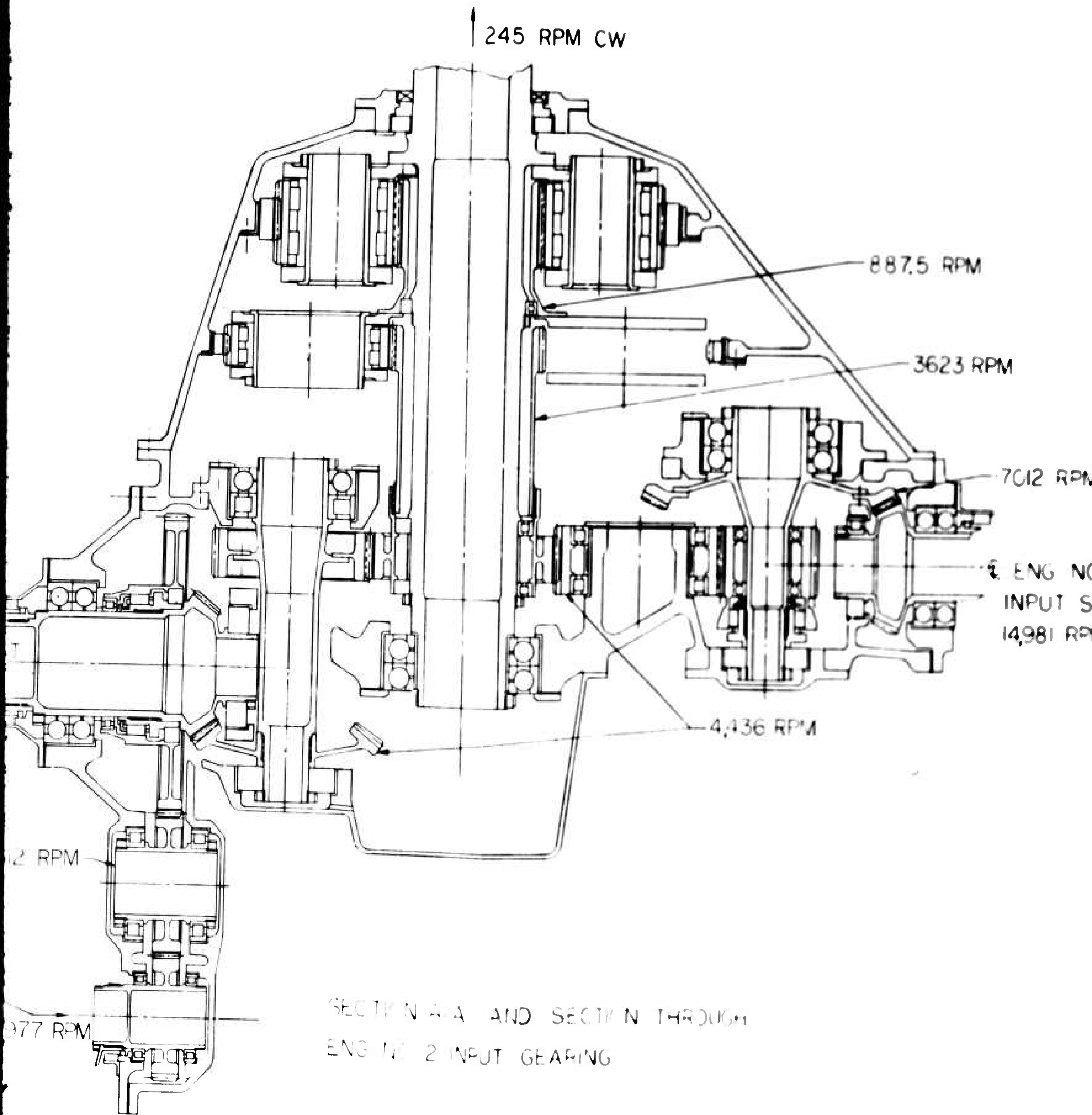


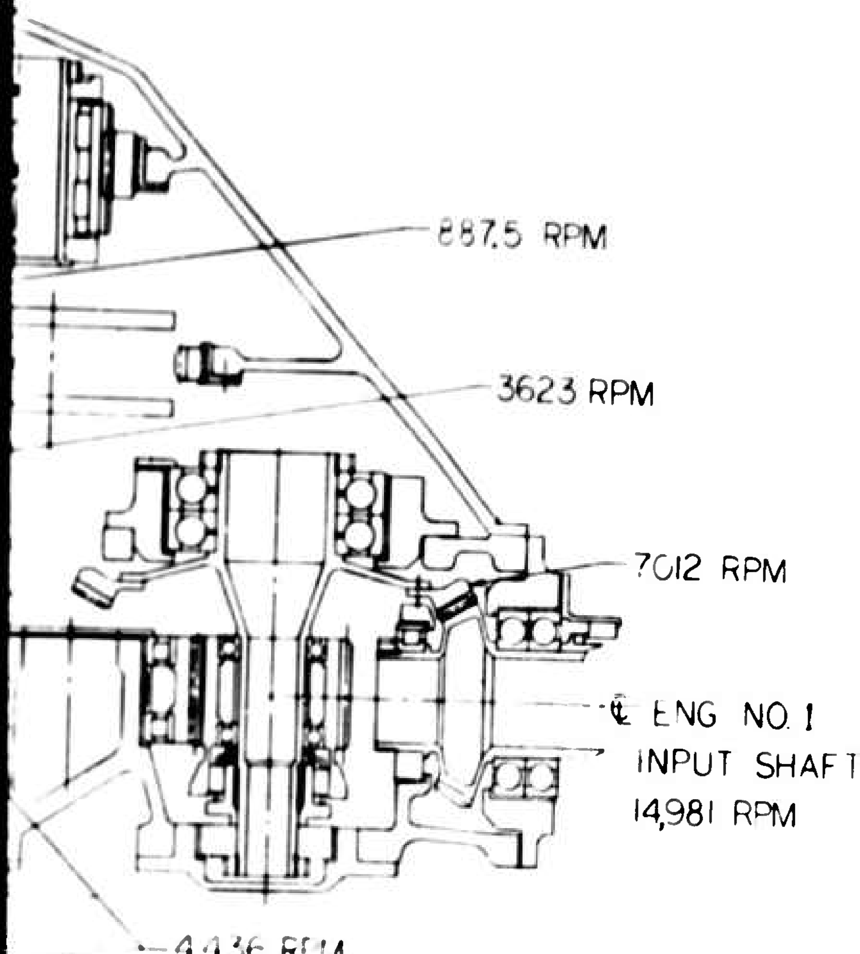
Figure 35. Configuration Number 11.



SECTION A-A AND SECTION THROUGH
ENG NO. 2 INPUT GEARING

B

.W



887.5 RPM
3623 RPM
7012 RPM
℄ ENG NO. 1
INPUT SHAFT
14,981 RPM
4,436 RPM

N 166 3061

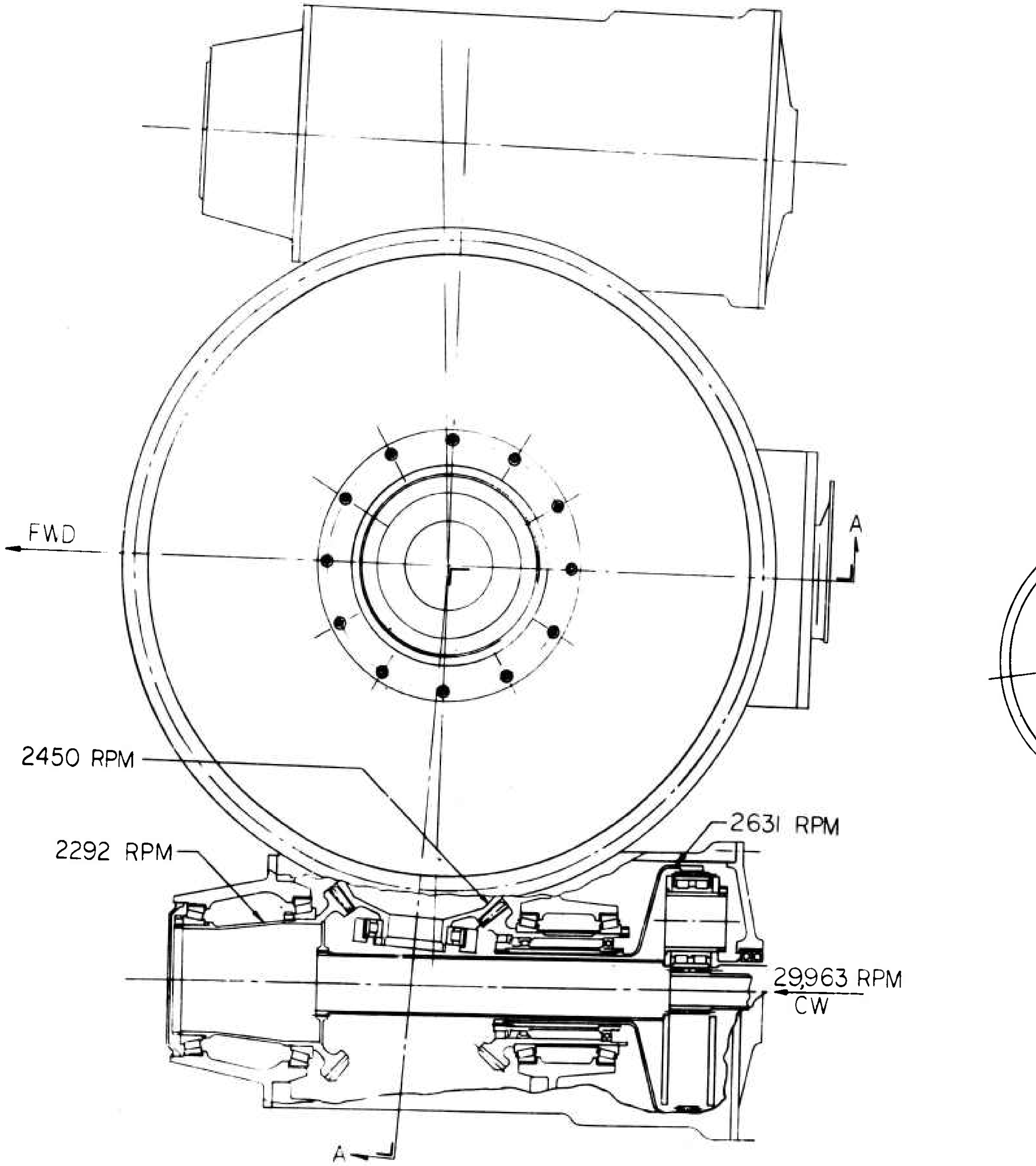
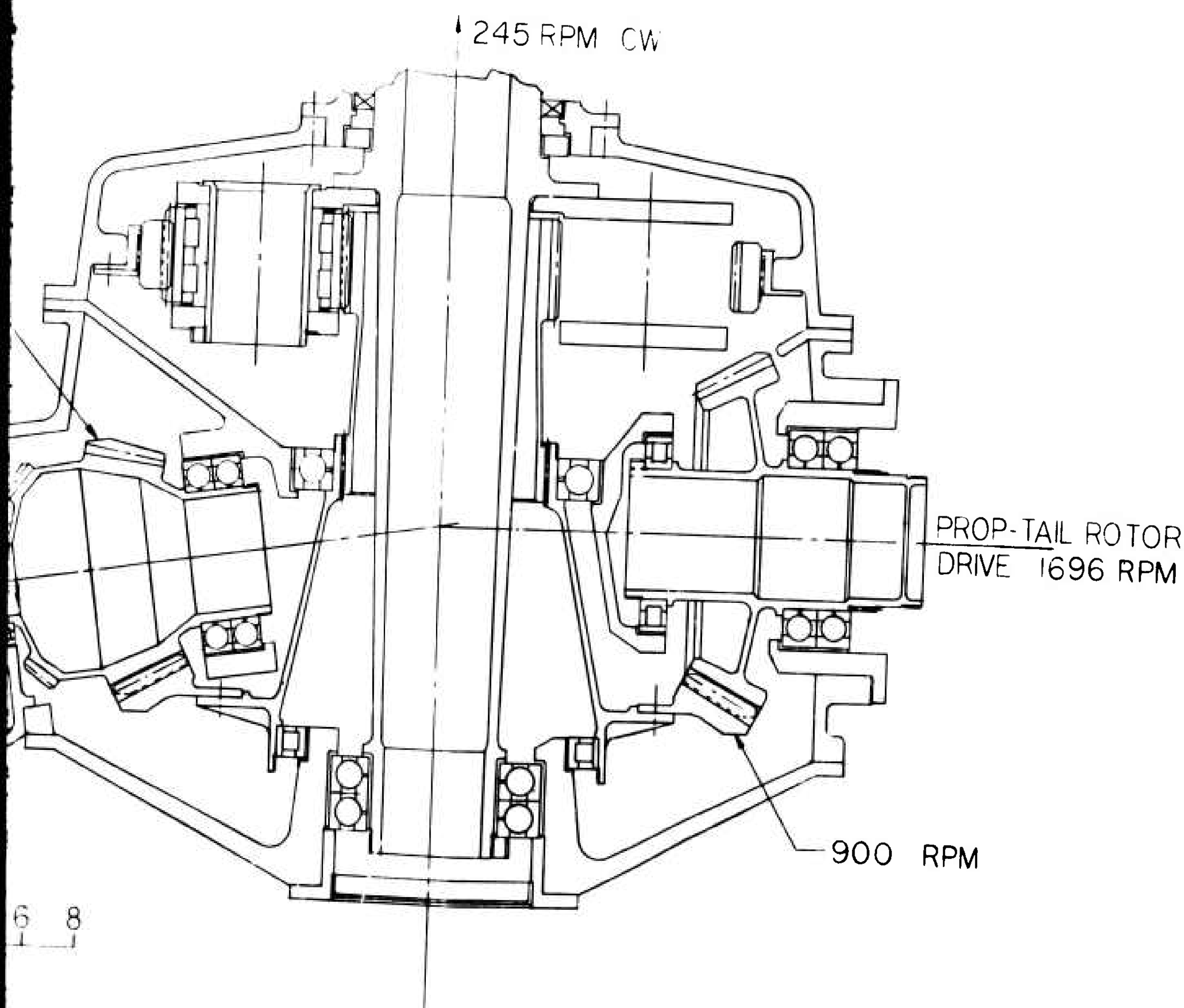


Figure 36. Configuration Number 12.

A



SECTION A-A

B

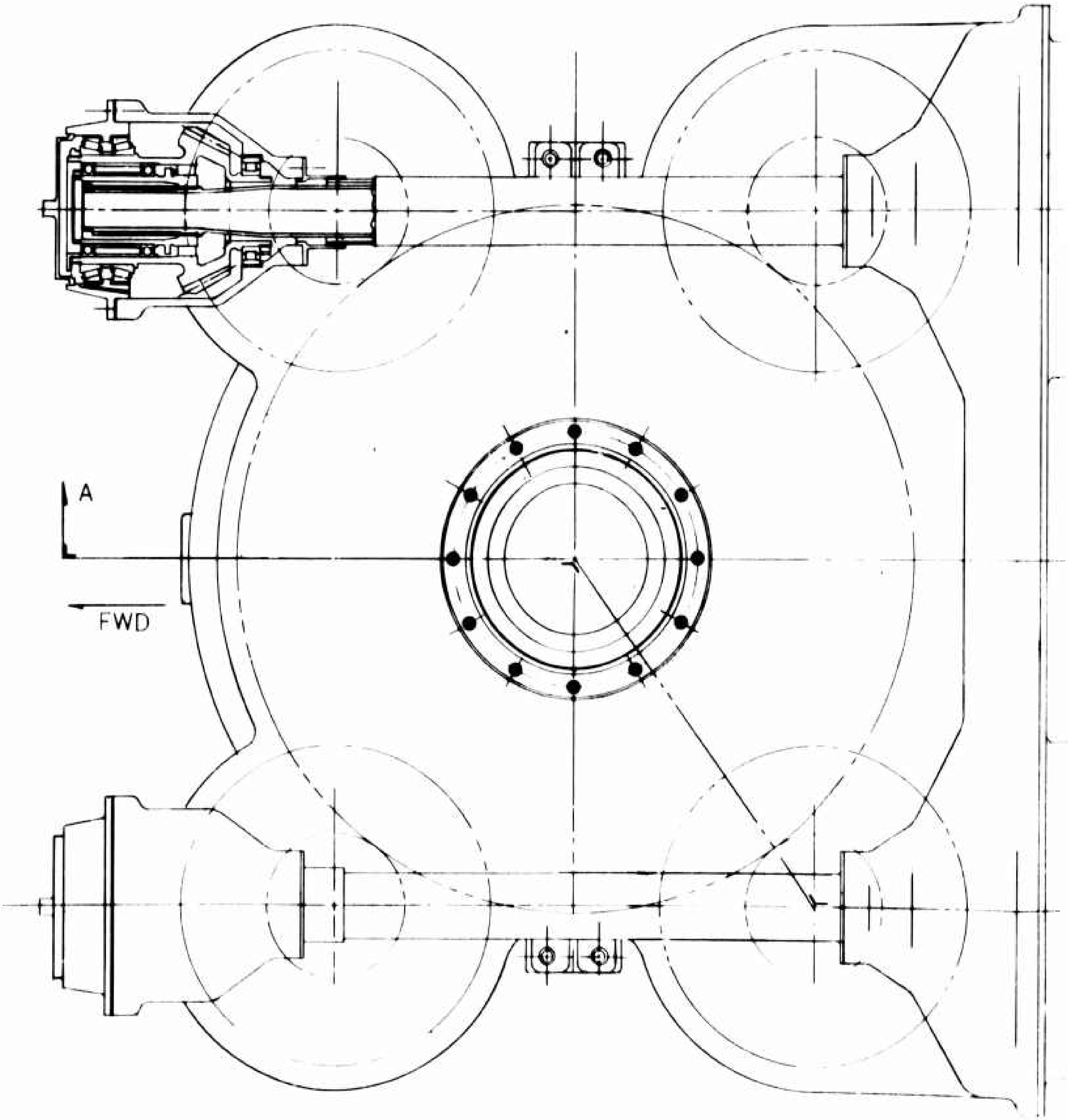
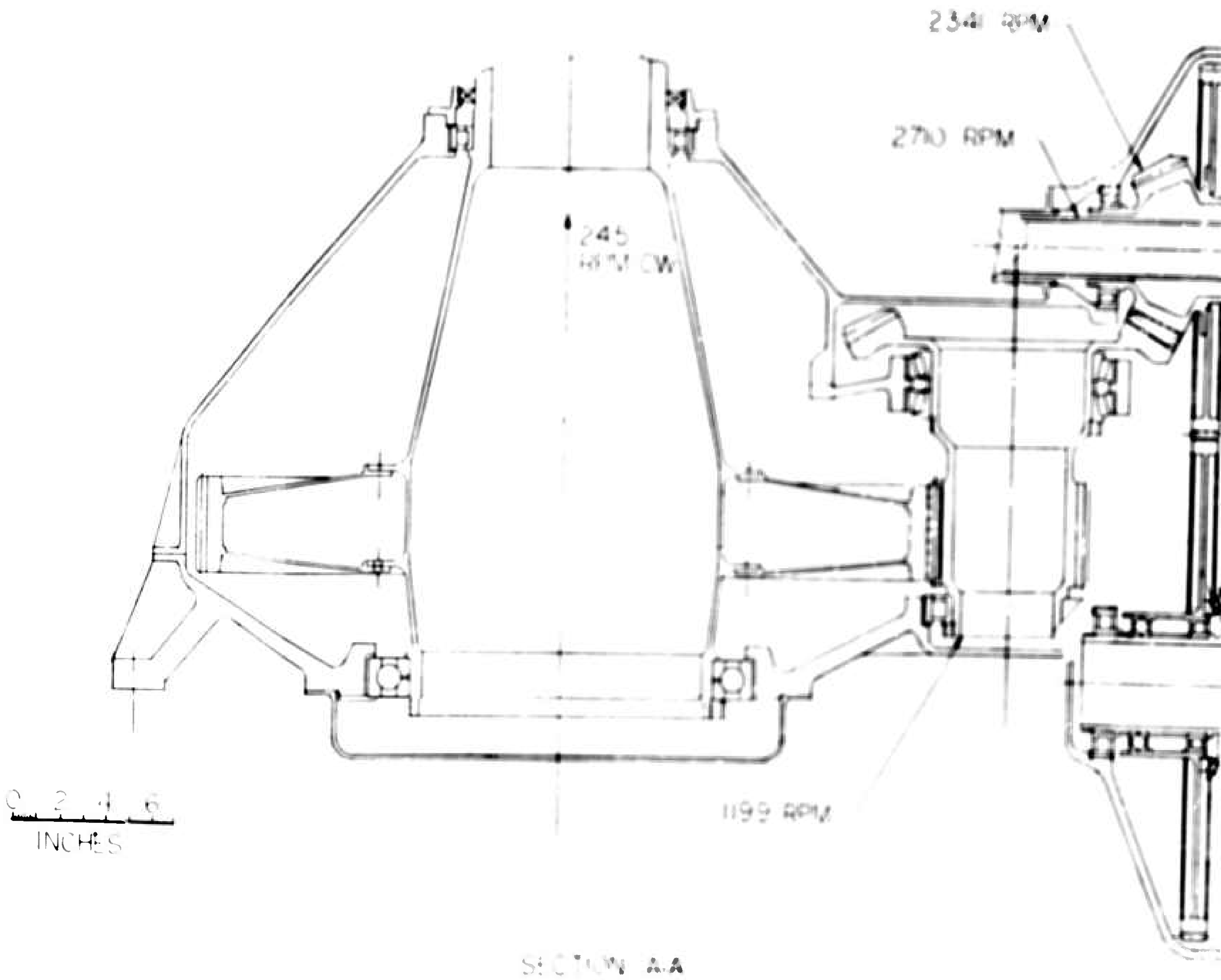


Figure 37. Configuration Number 13.

A



SECTION AA

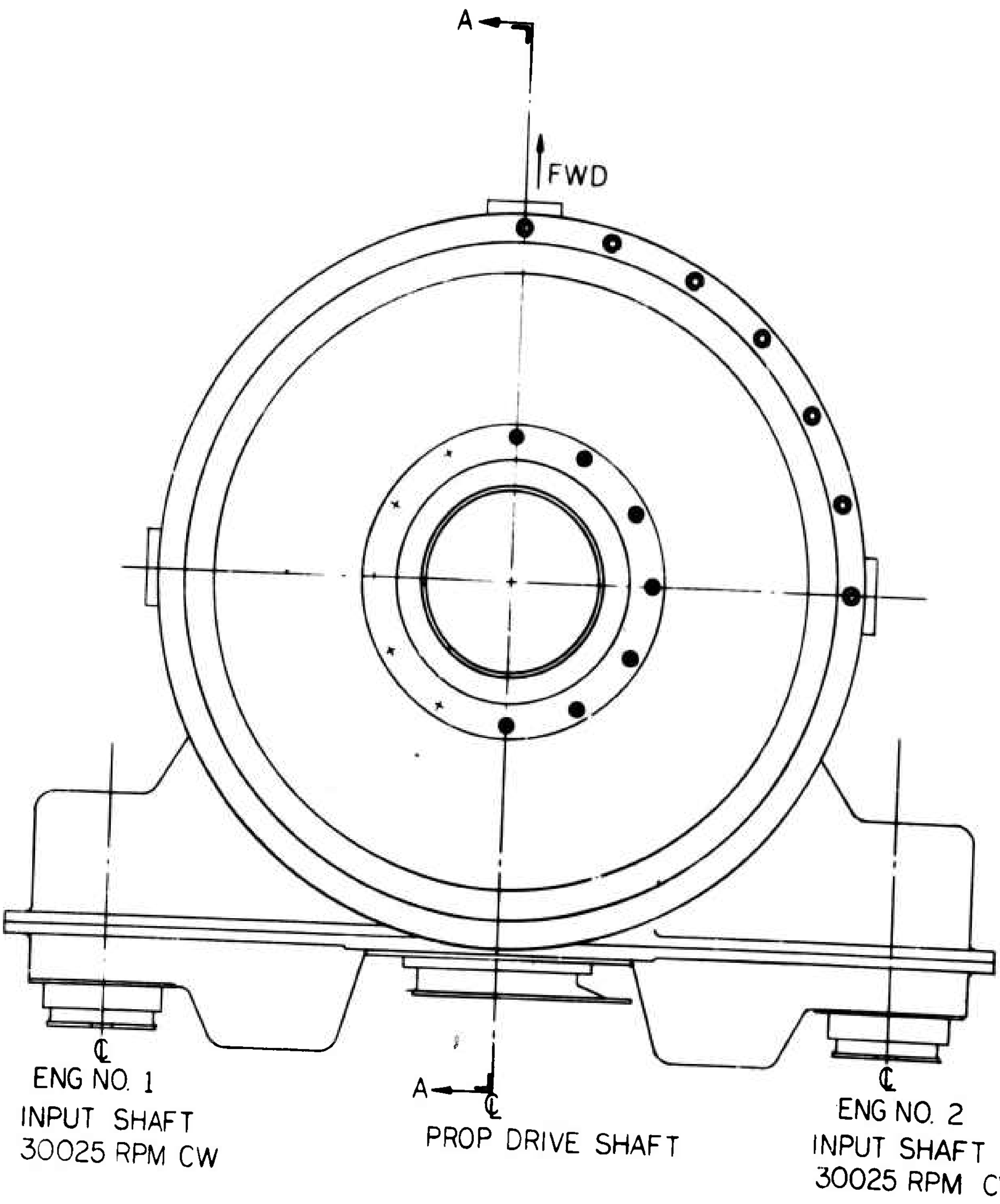
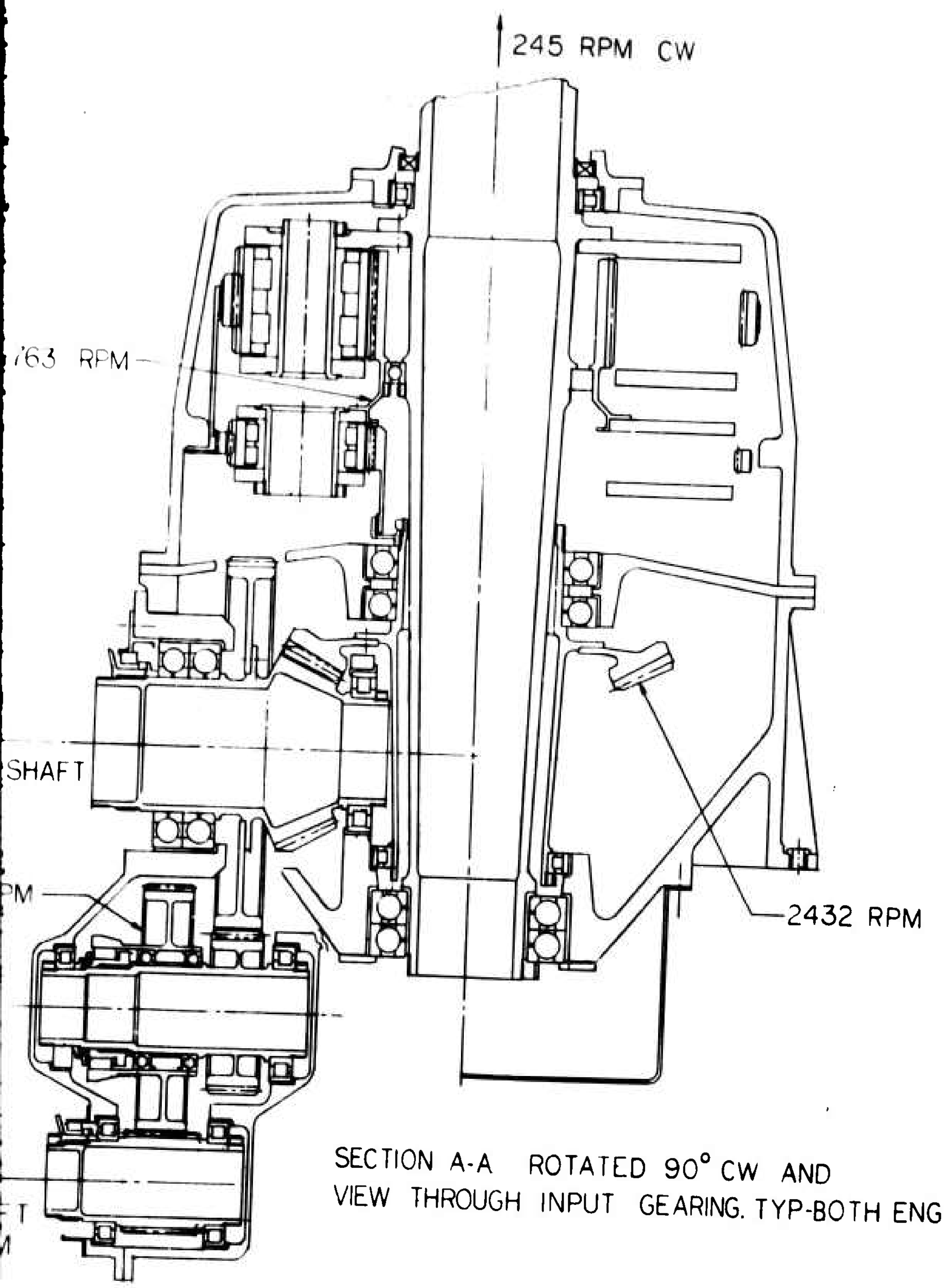
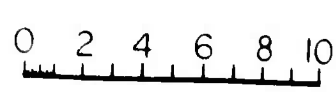


Figure 38. Configuration Number 14.

A



SECTION A-A ROTATED 90° CW AND
VIEW THROUGH INPUT GEARING. TYP-BOTH ENG



B

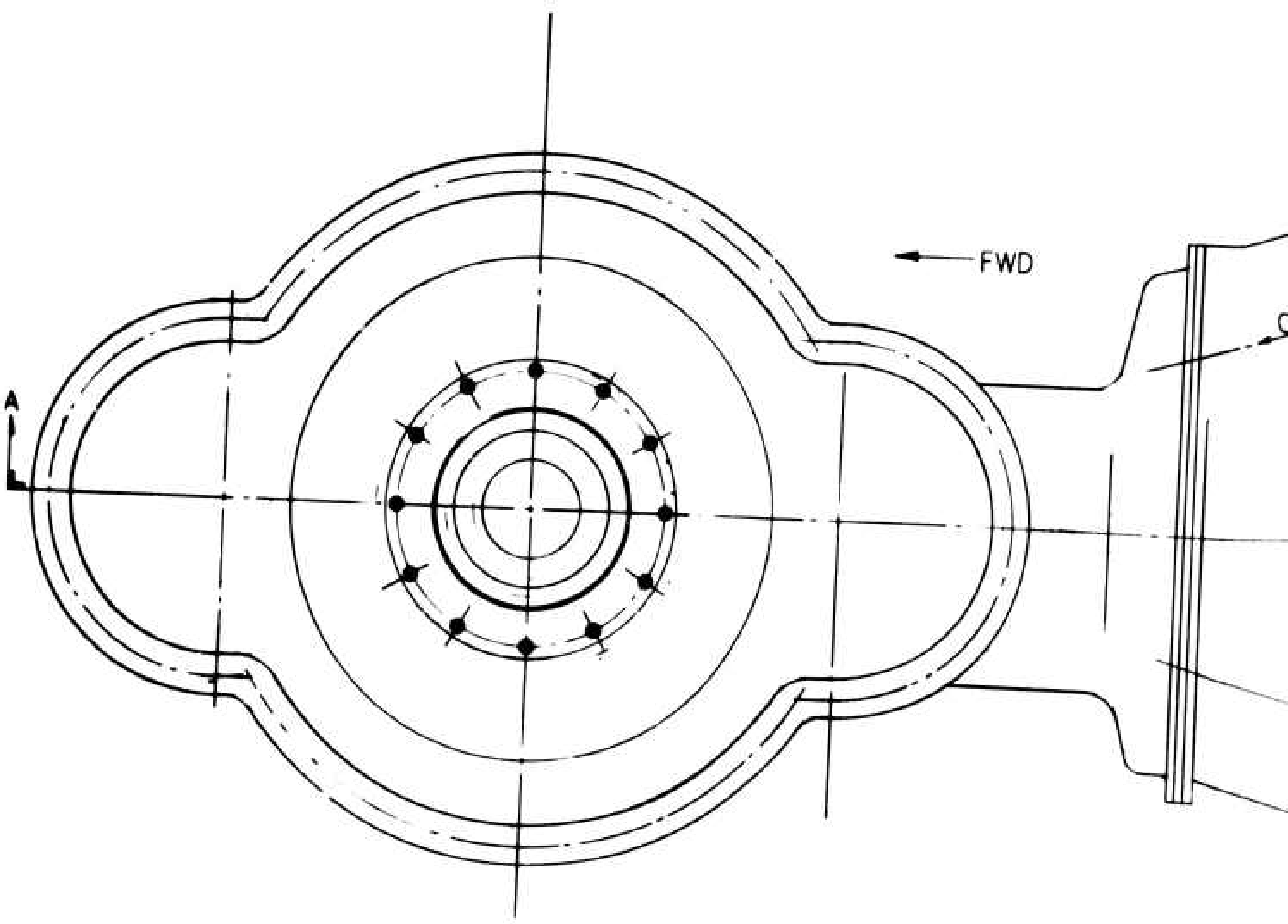
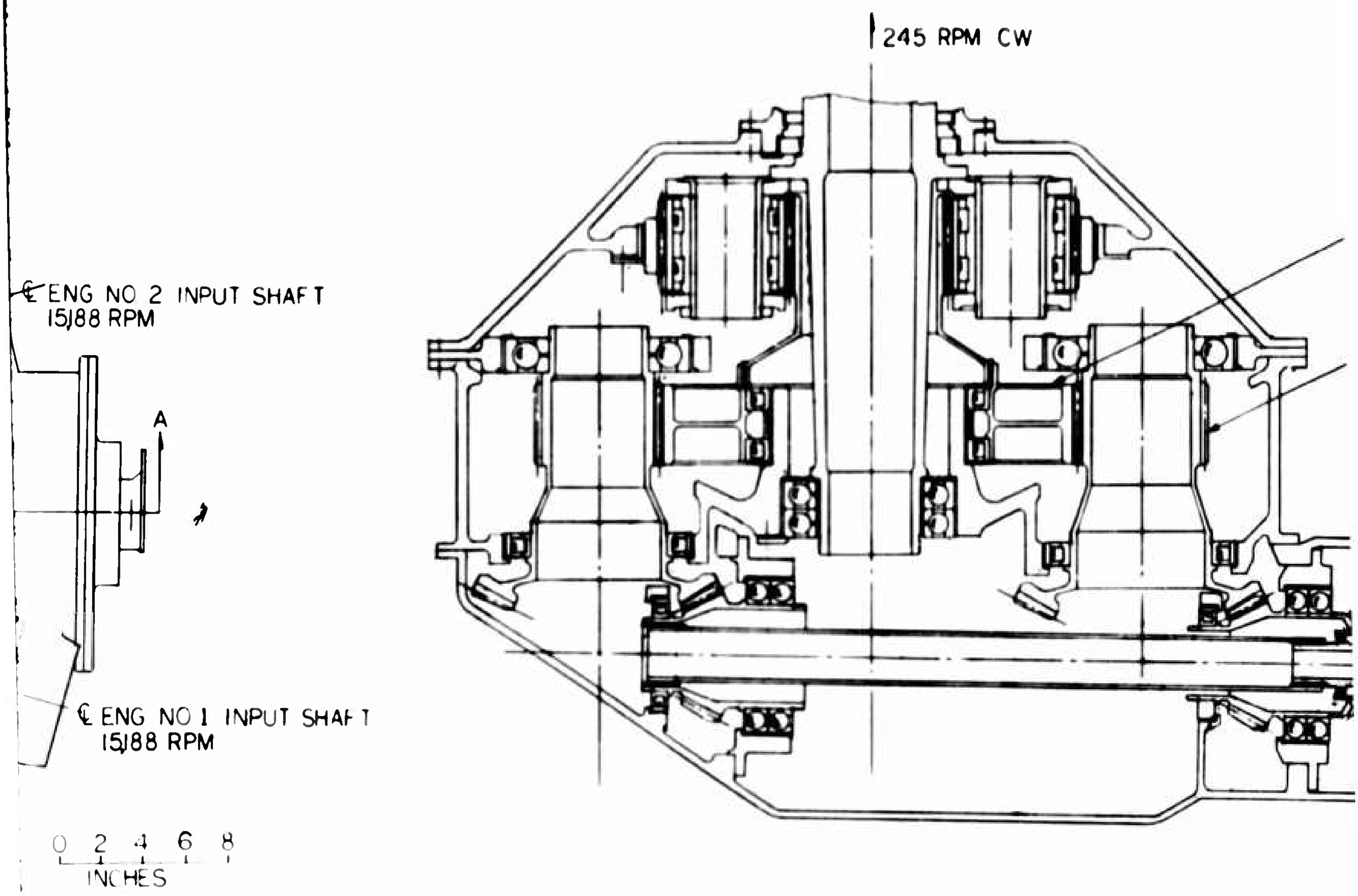


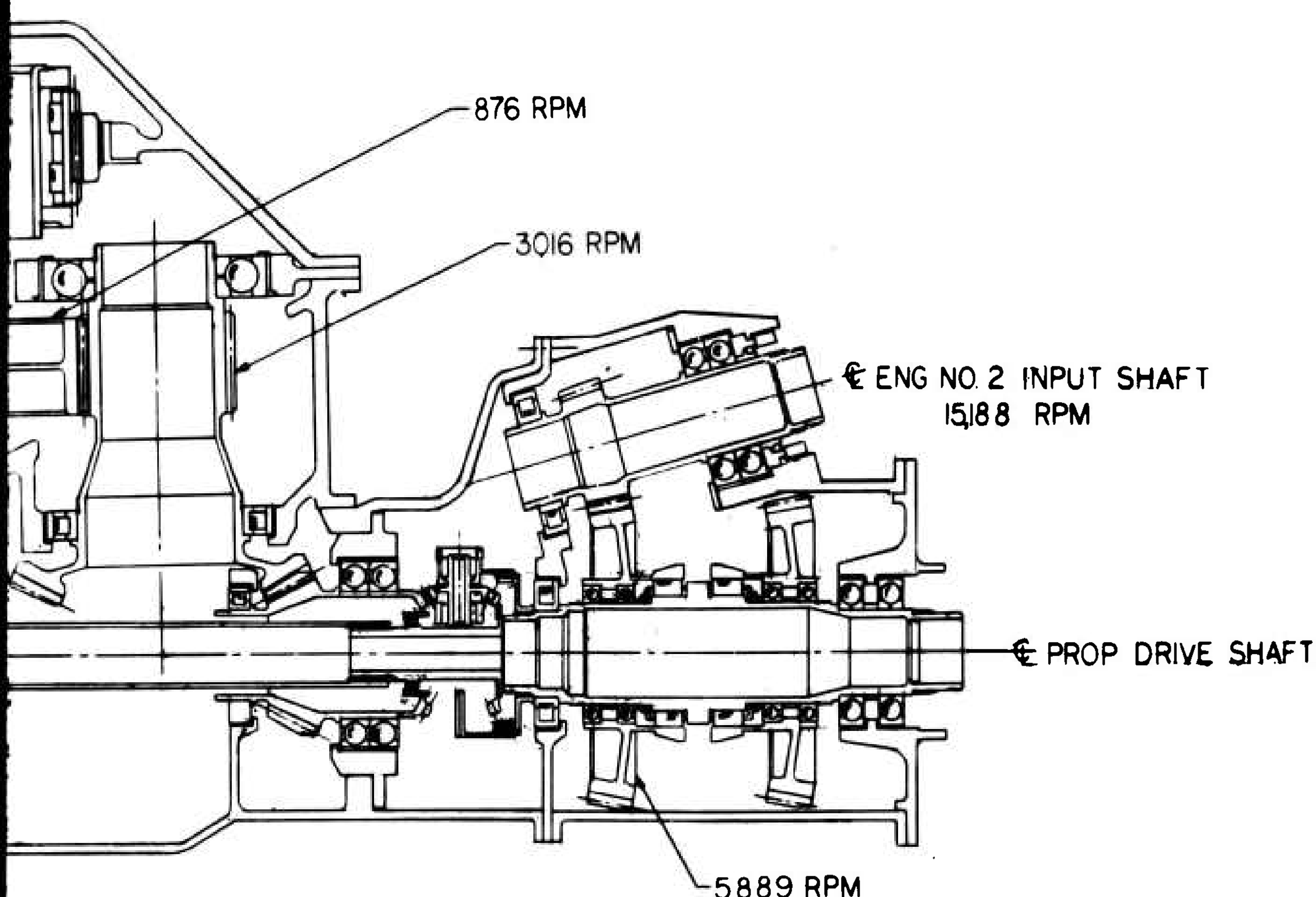
Figure 39. Configuration Number 15.

A



SECTION A-A AND SECTION THROUGH ENG NO.2 INPUT GEARI

B



ROUGH ENG NO.2 INPUT GEARING

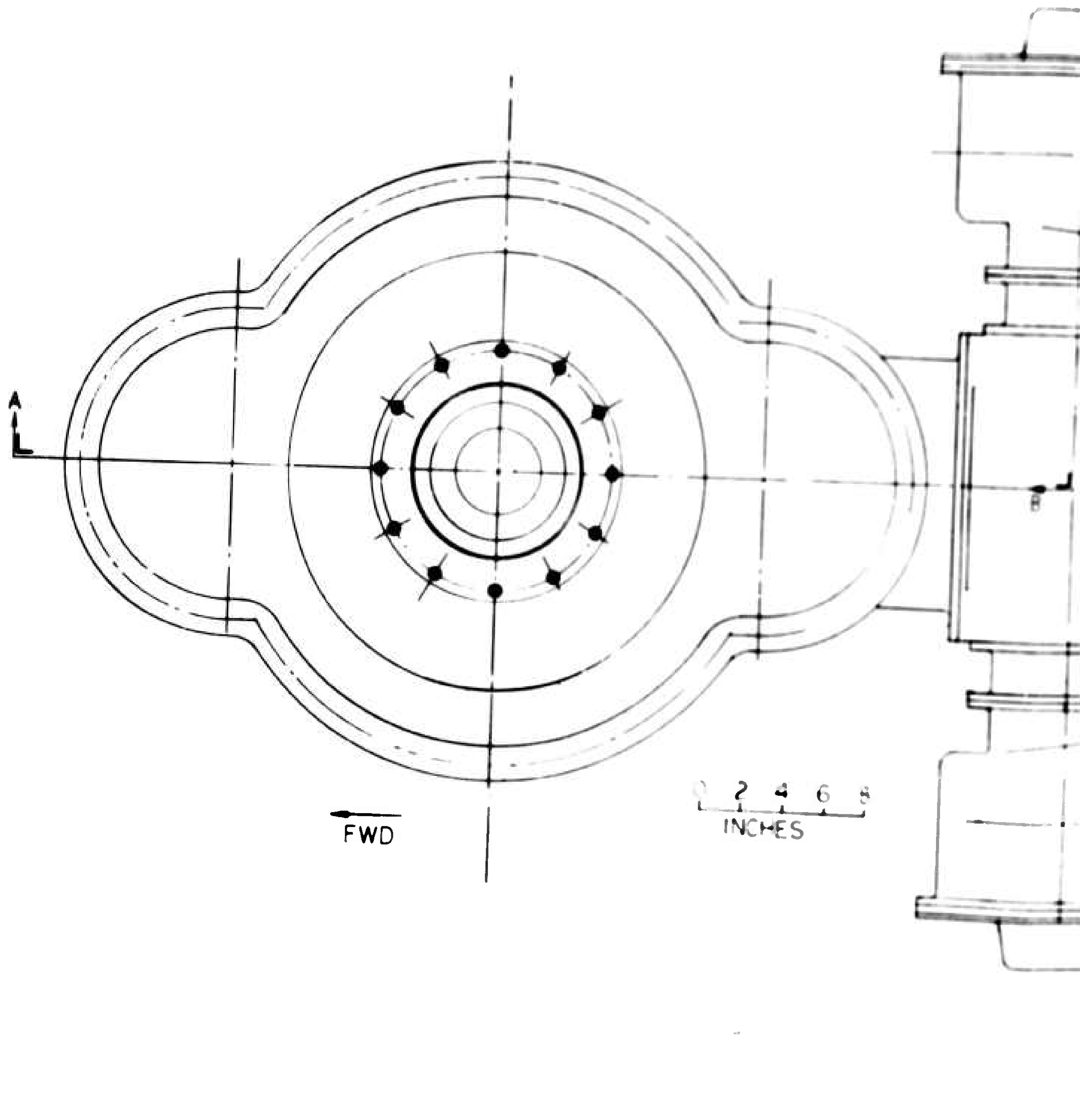
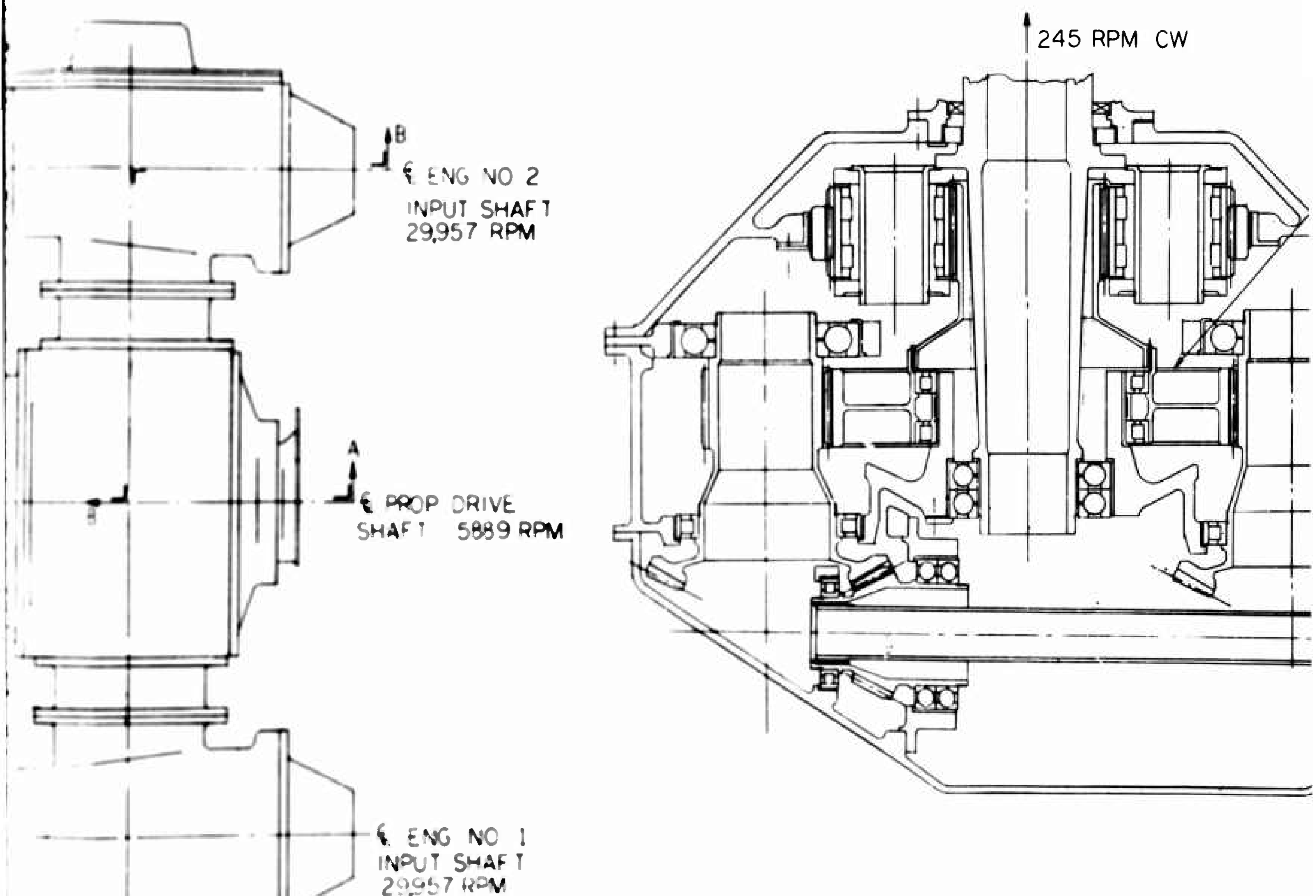


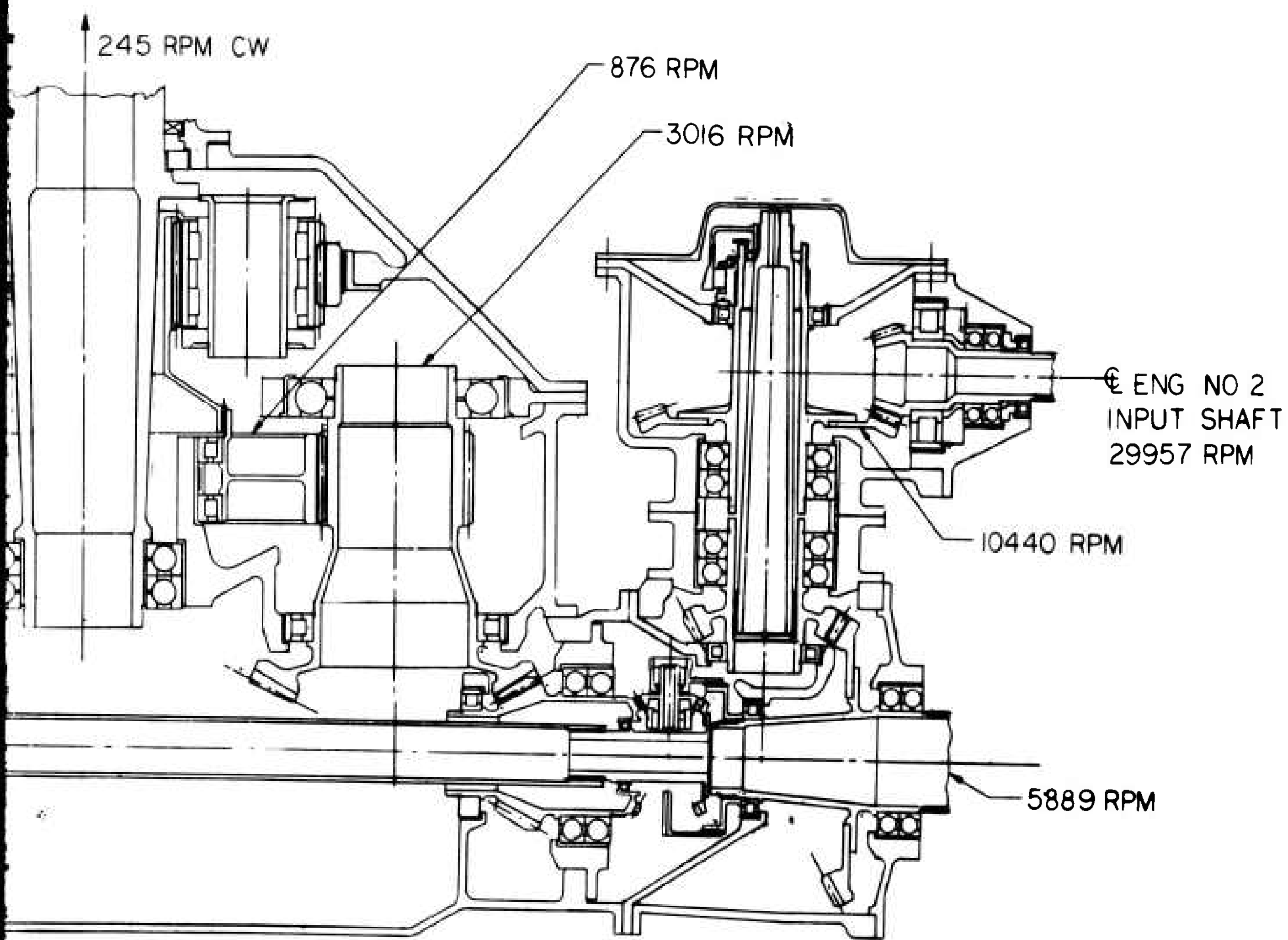
Figure 40. Configuration Number 16.

A



SECTION A-A AND SECTION B-B. SECTION B-B

B



AND SECTION B-B. SECTION B-B ROTATED UP 90°

Q

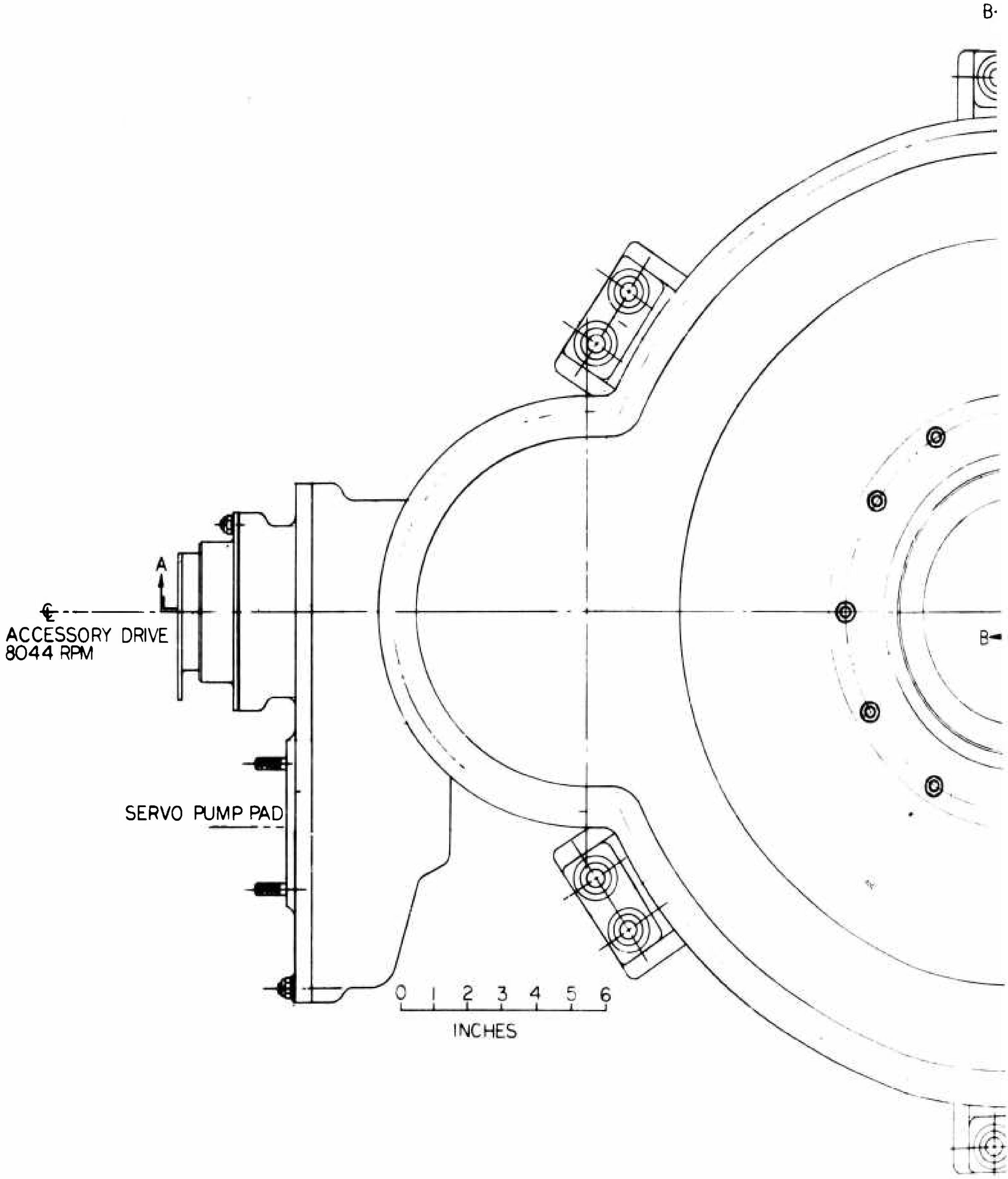
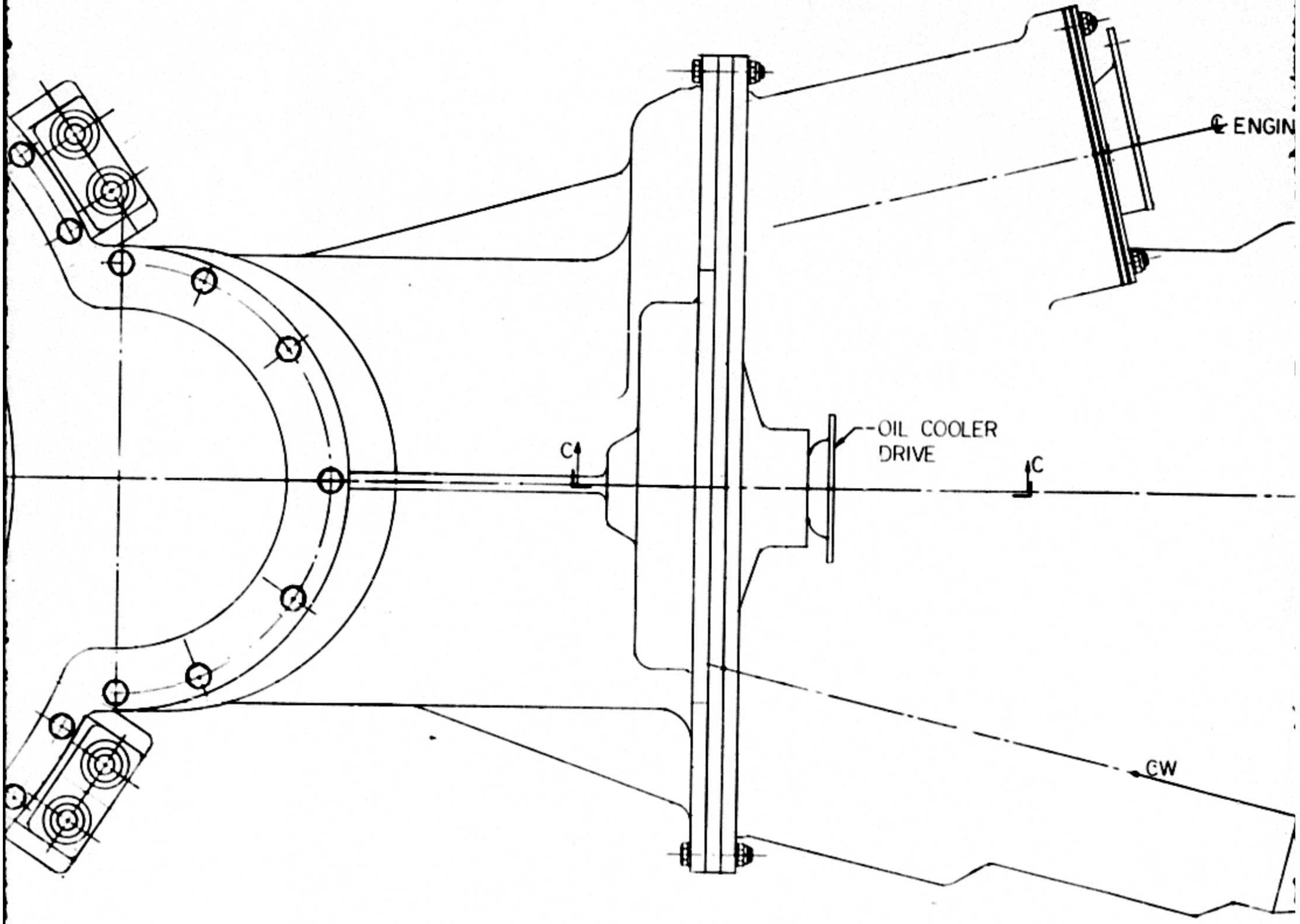
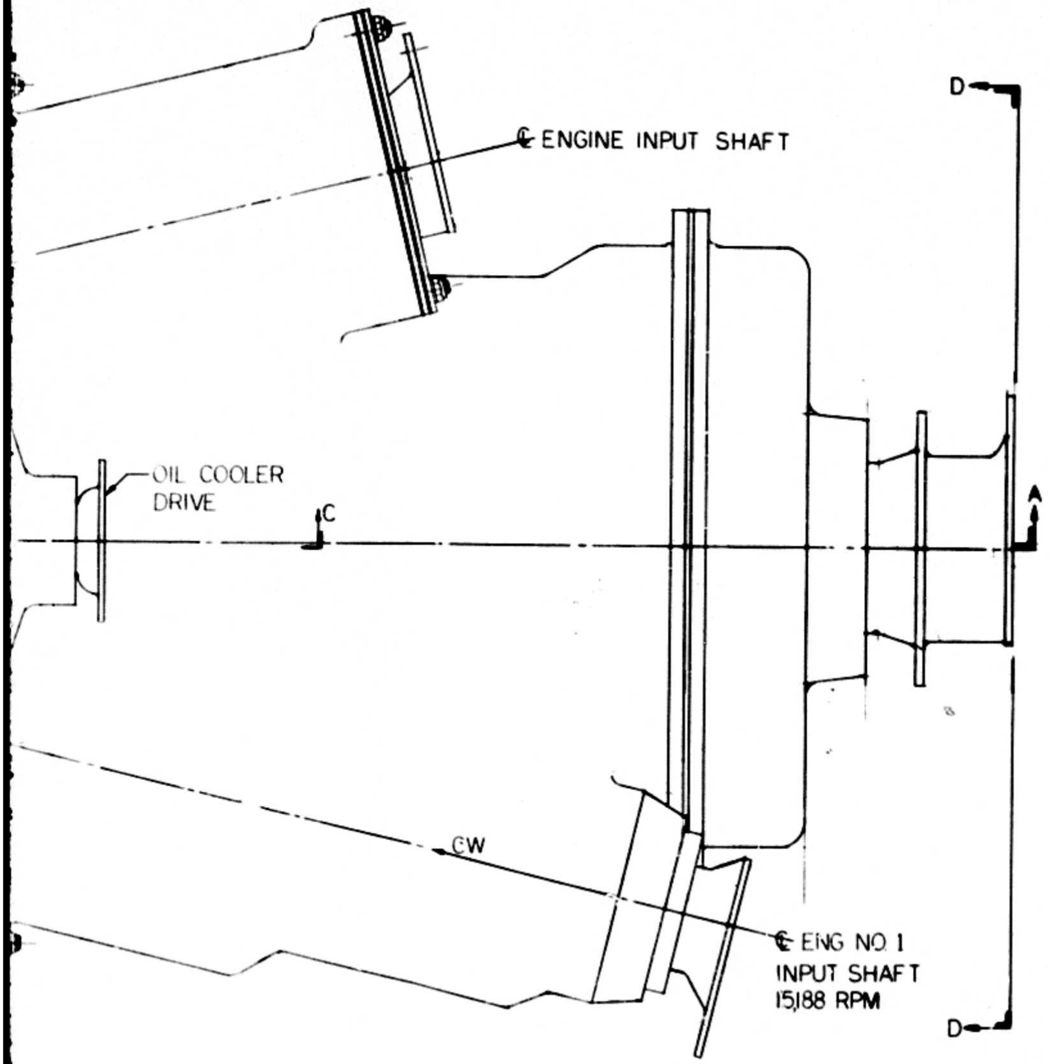


Figure 41. Main Gearbox, Compound Helicopter.

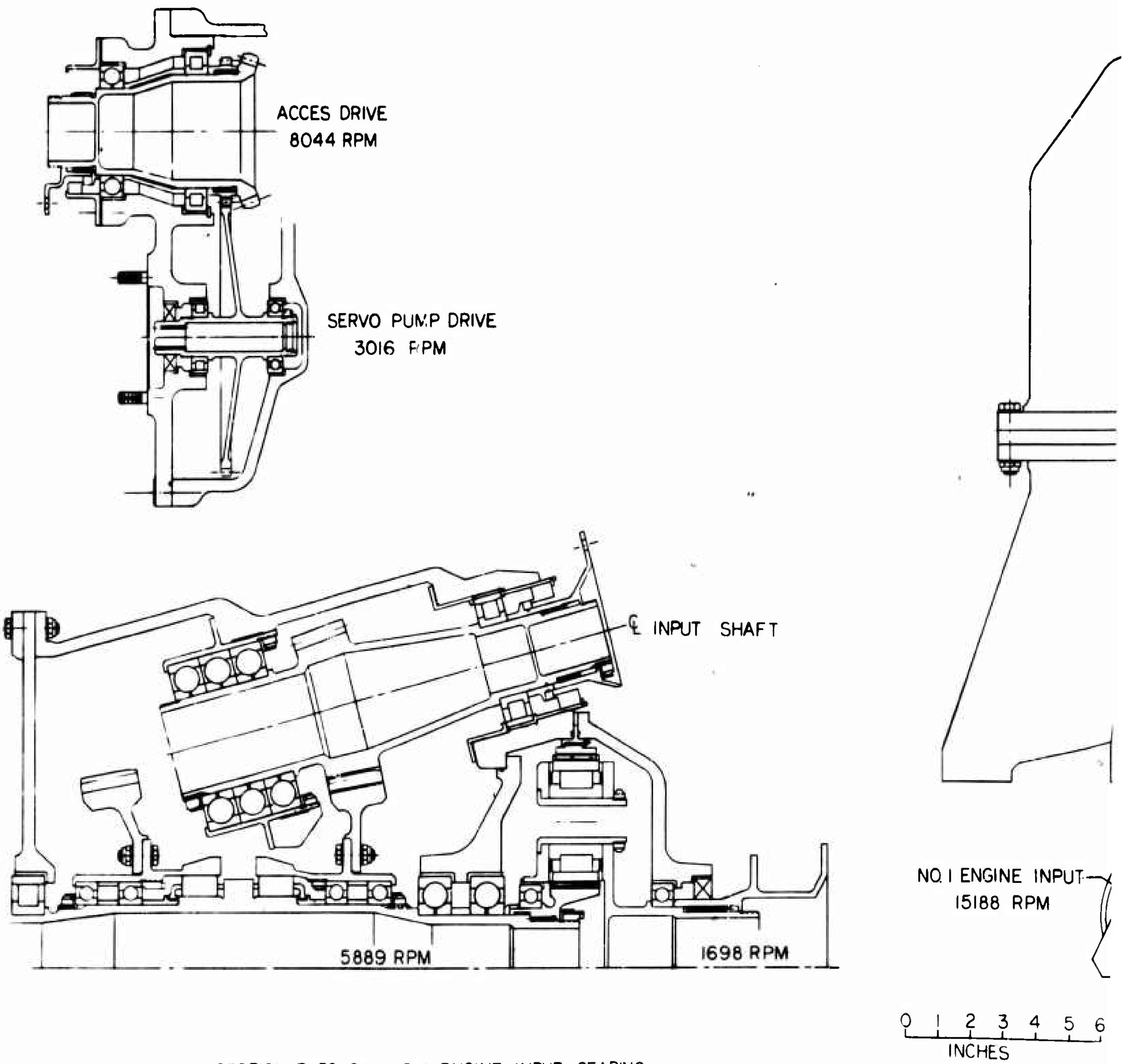
A



B



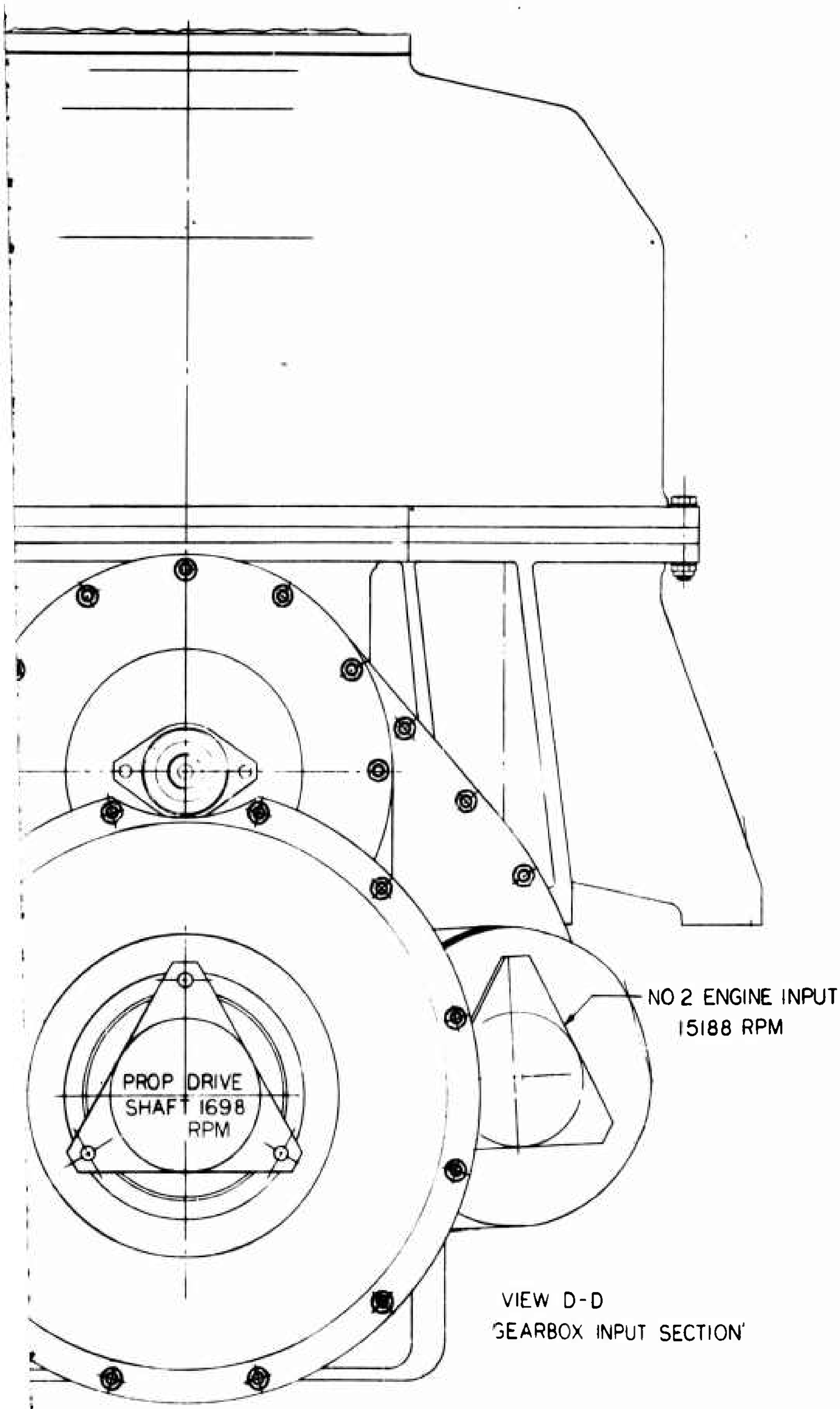
ENGINE INPUT SHAFT
OIL COOLER
DRIVE
C
D
A
CW
ENG NO. 1
INPUT SHAFT
15,188 RPM
D



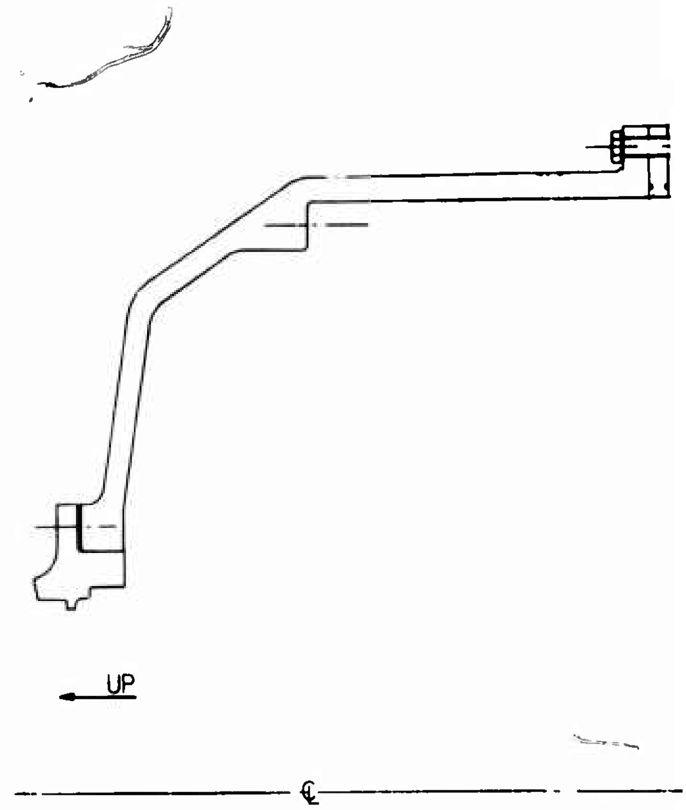
SECTION THROUGH NO 1 ENGINE INPUT GEARING

Figure 41. Continued

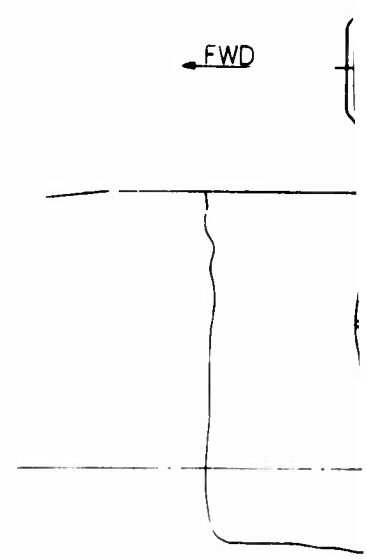
A



VIEW D-D
GEARBOX INPUT SECTION'

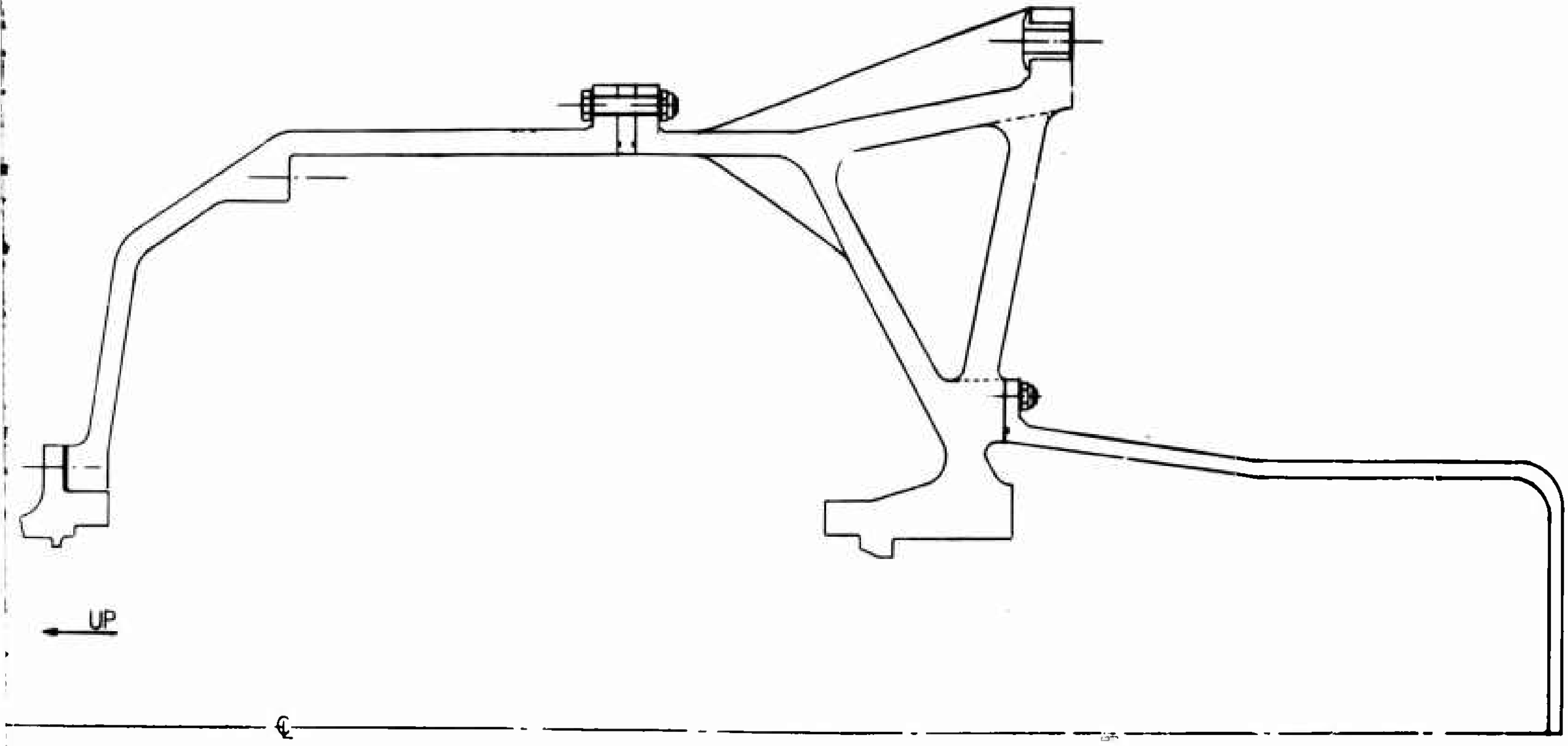


SECTION B-B. CASTING DETAILS ONLY

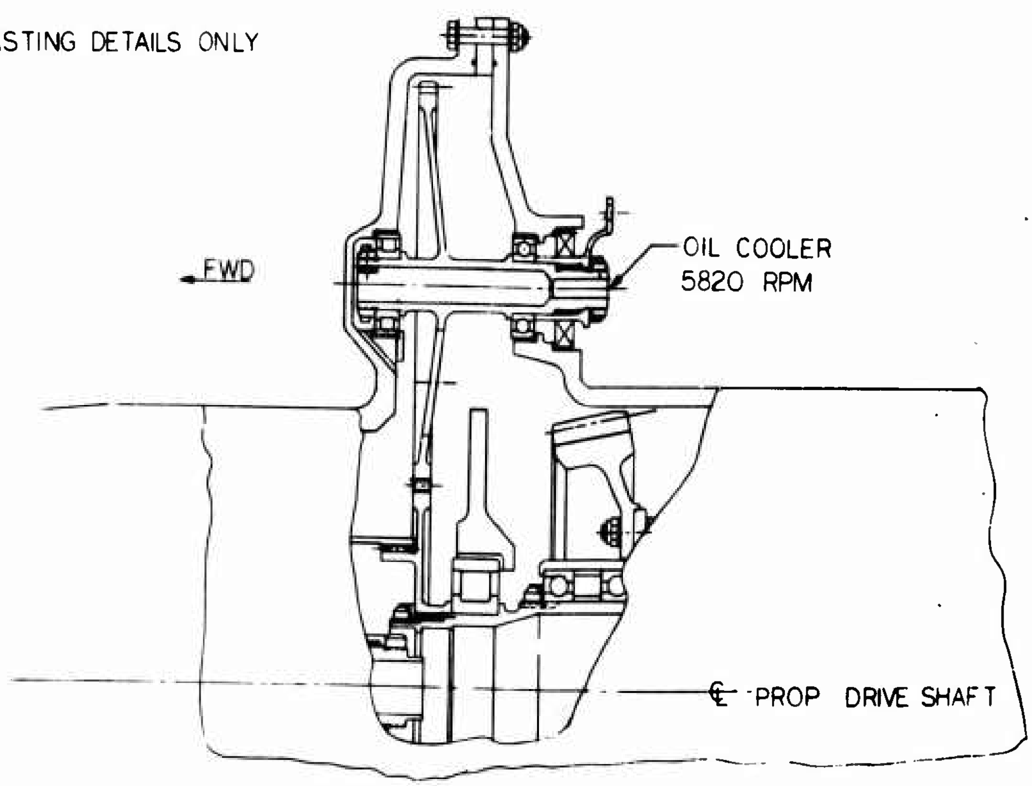


SEC

B



SECTION B-B. CASTING DETAILS ONLY



SECTION C-C OIL COOLER DRIVE

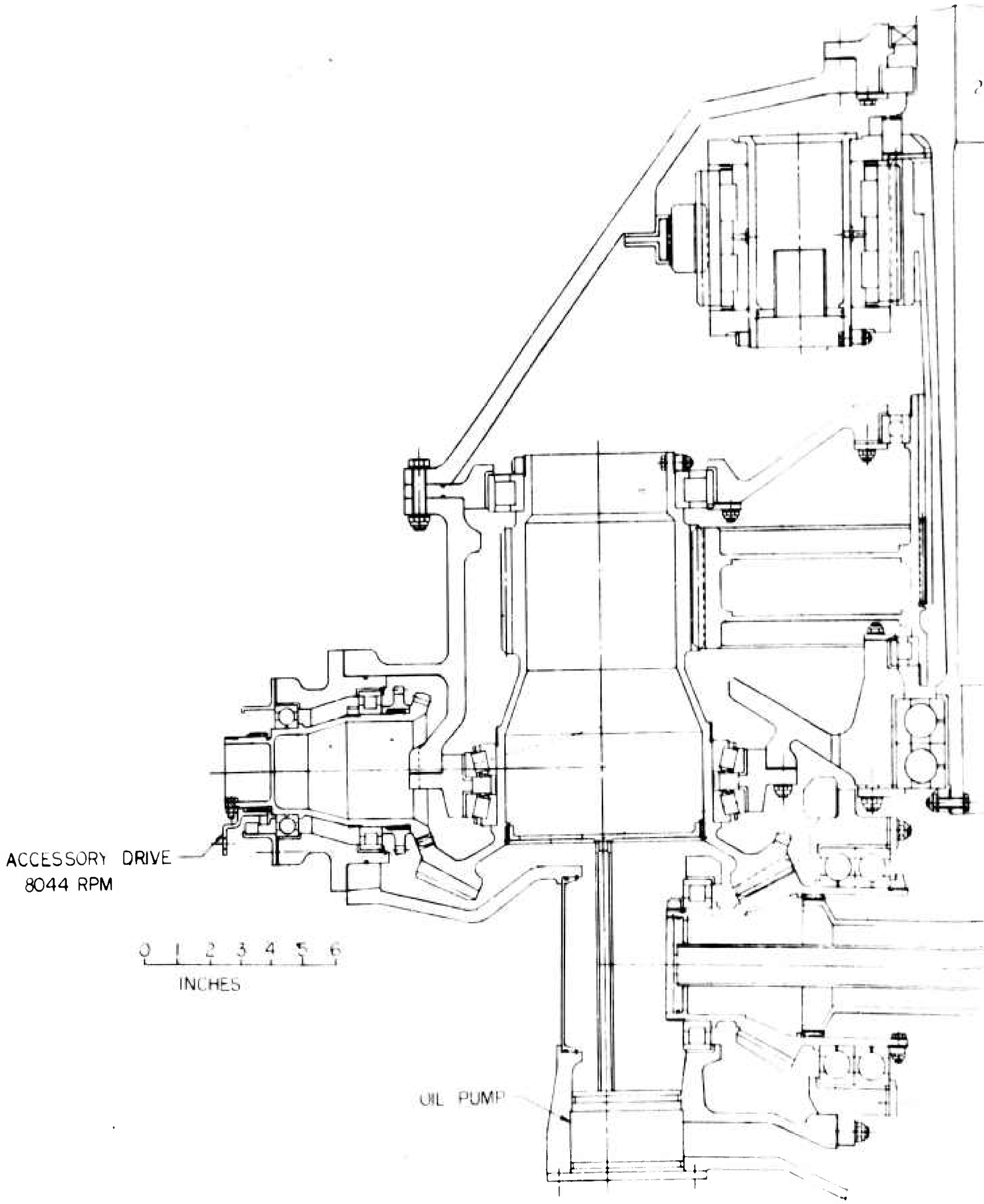
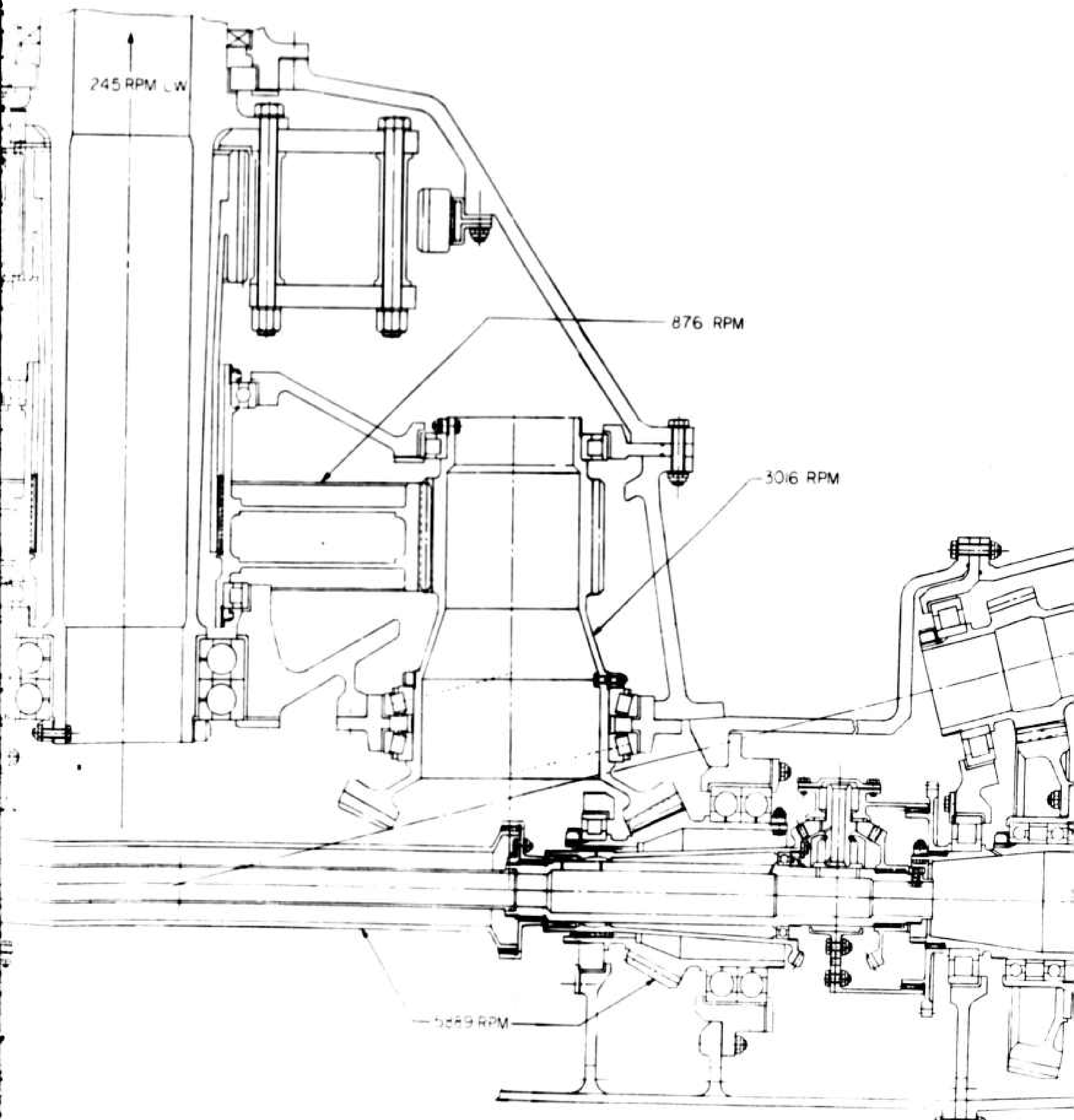
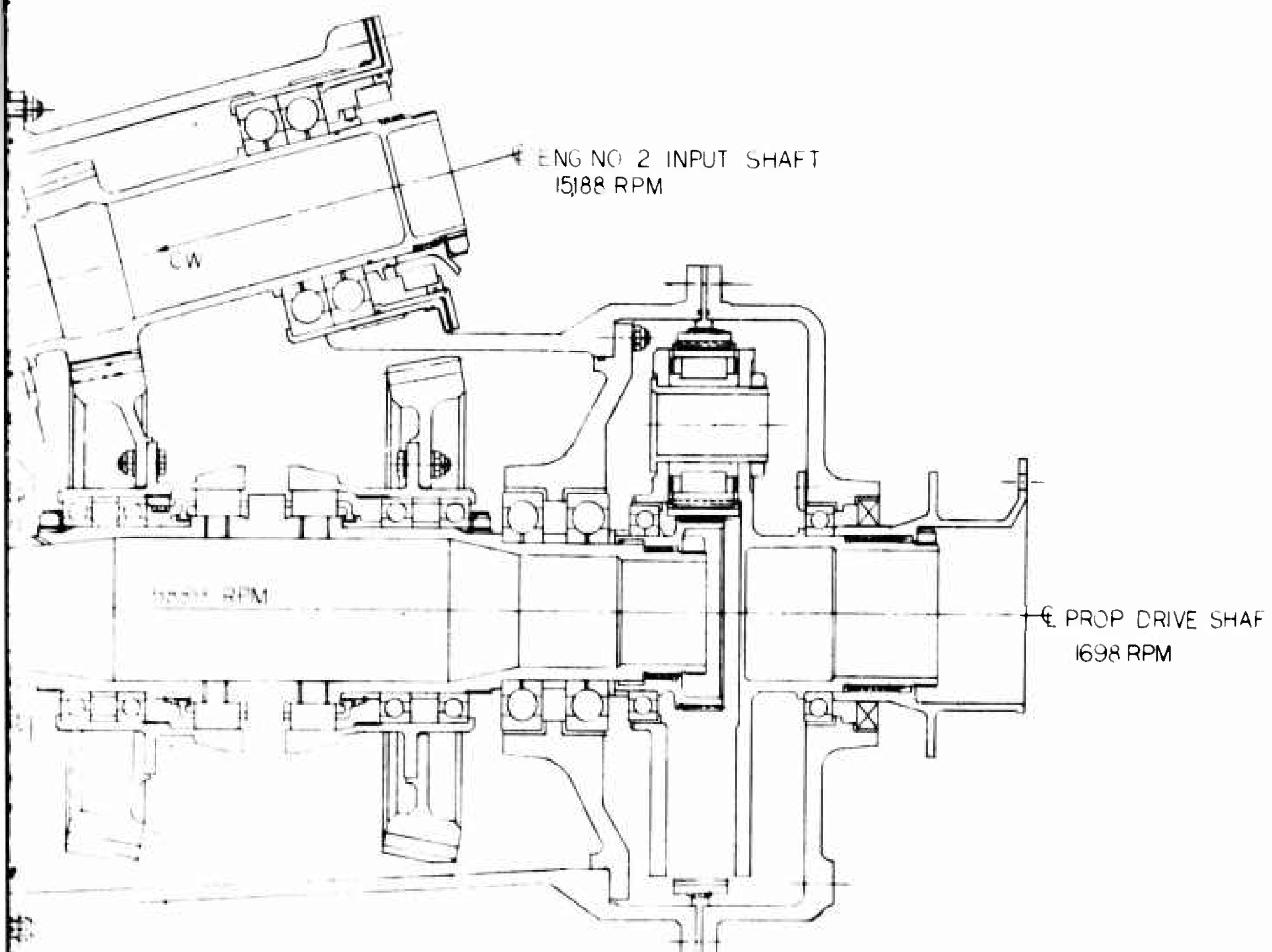


Figure 42. Section A-A, Main Gearbox, Compound Helicopter.

A



B



℄ ENG NO 2 INPUT SHAFT
15,188 RPM
CW
℄ PROP DRIVE SHAFT
1698 RPM

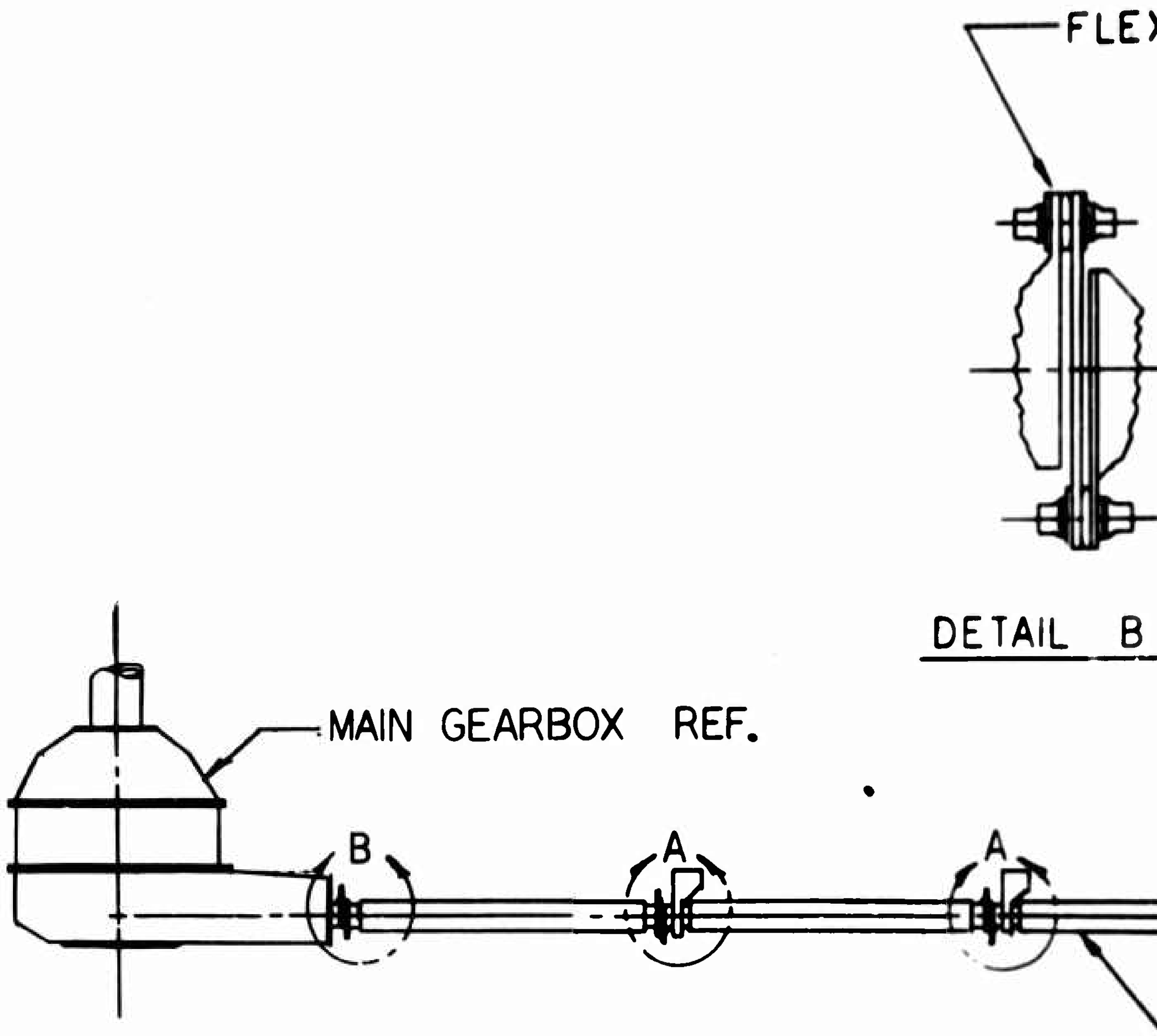
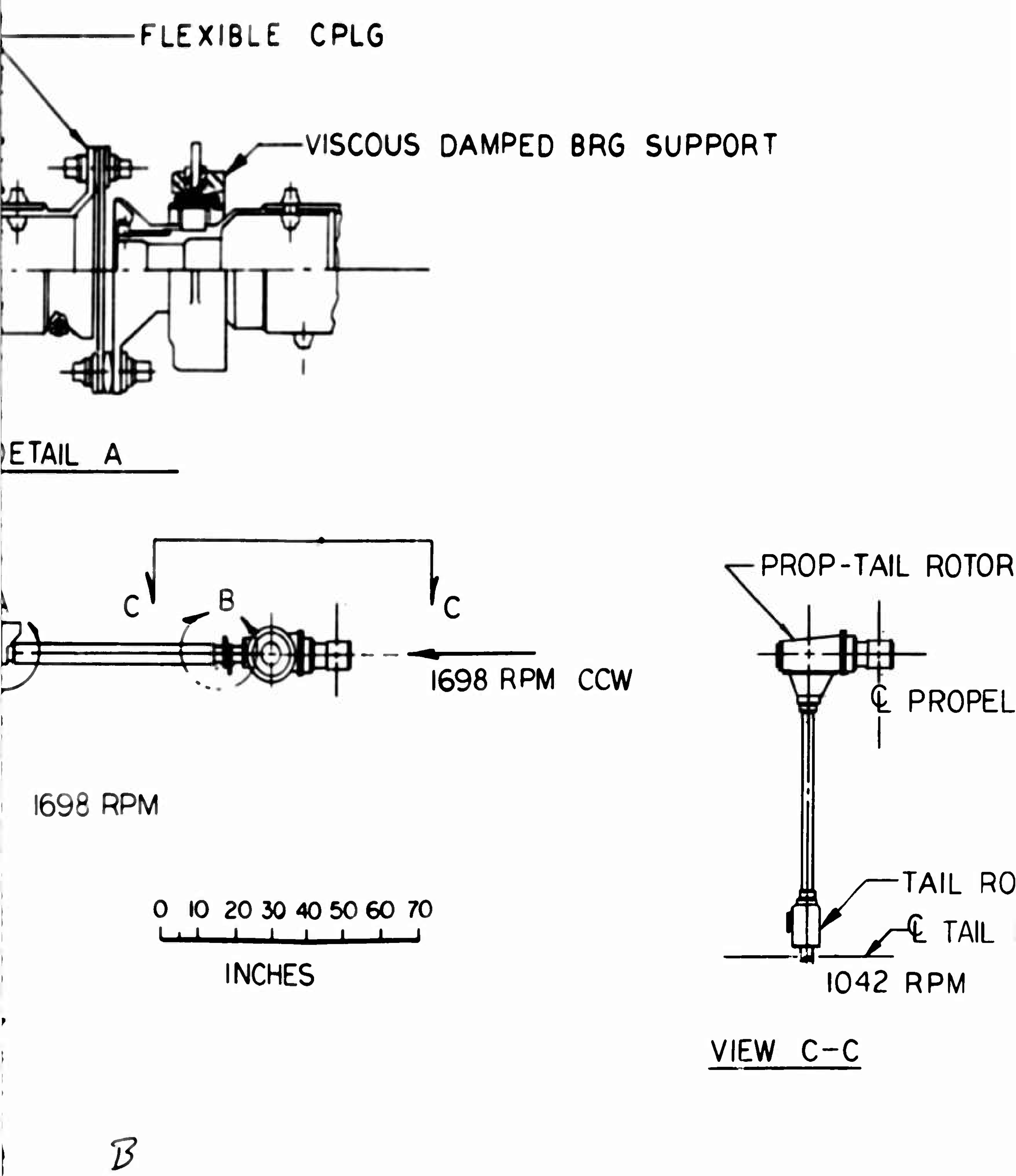


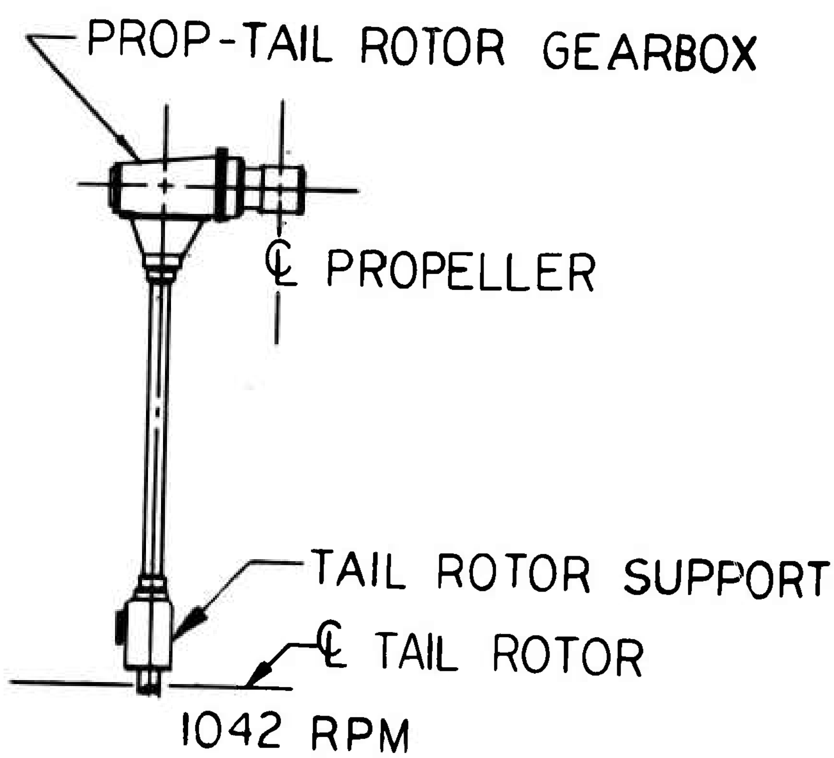
Figure 43. Propeller - Tail Rotor Drive System, Compound Helicopter.

A


FLEXIBLE CPLG
VISCOUS DAMPED BRG SUPPORT
ETAIL A
C
B
C
1698 RPM CCW
1698 RPM
0 10 20 30 40 50 60 70
INCHES
PROP-TAIL ROTOR
℄ PROPEL
TAIL RO
℄ TAIL
1042 RPM
VIEW C-C
B

RG SUPPORT

CW



VIEW C-C

C

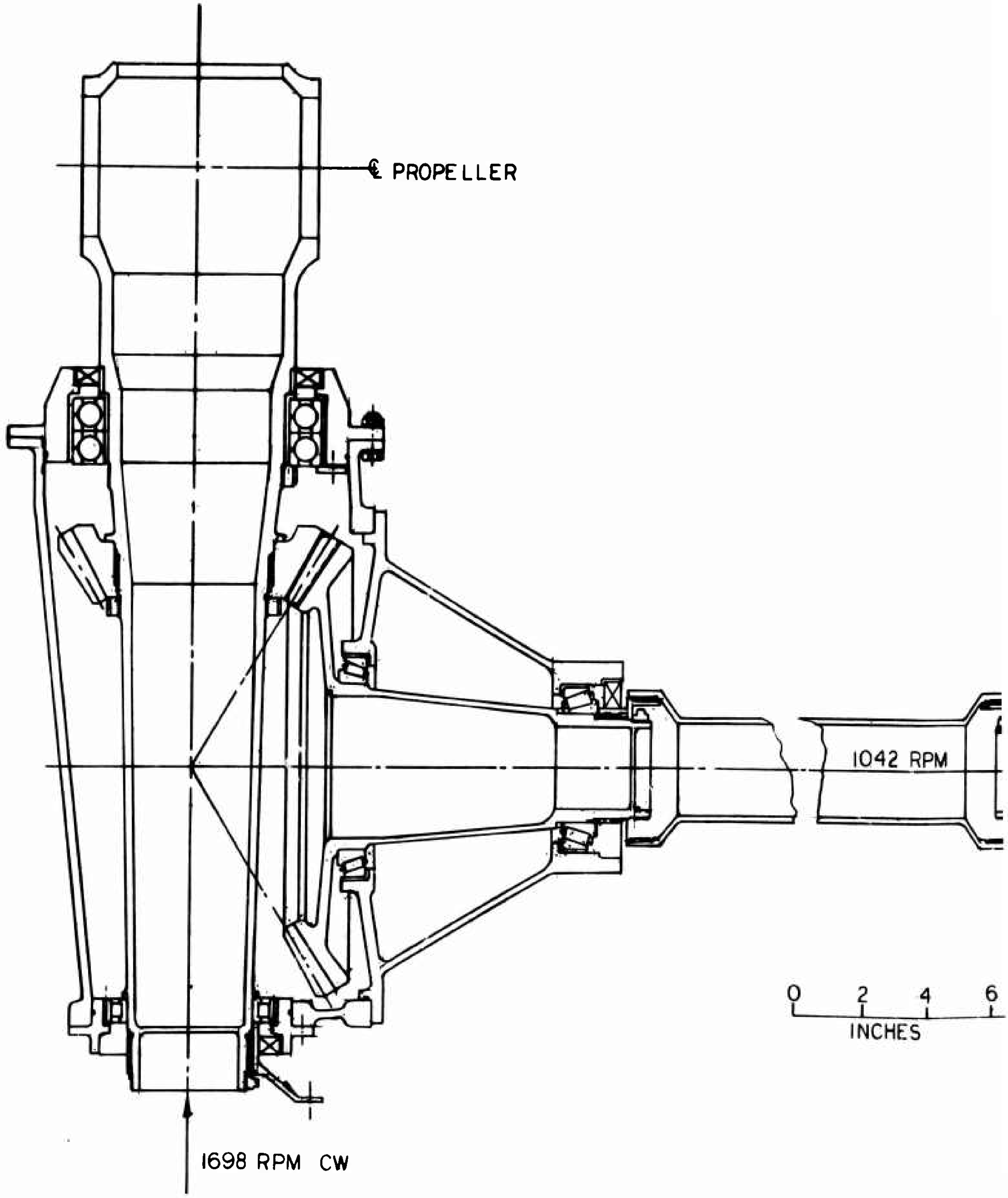
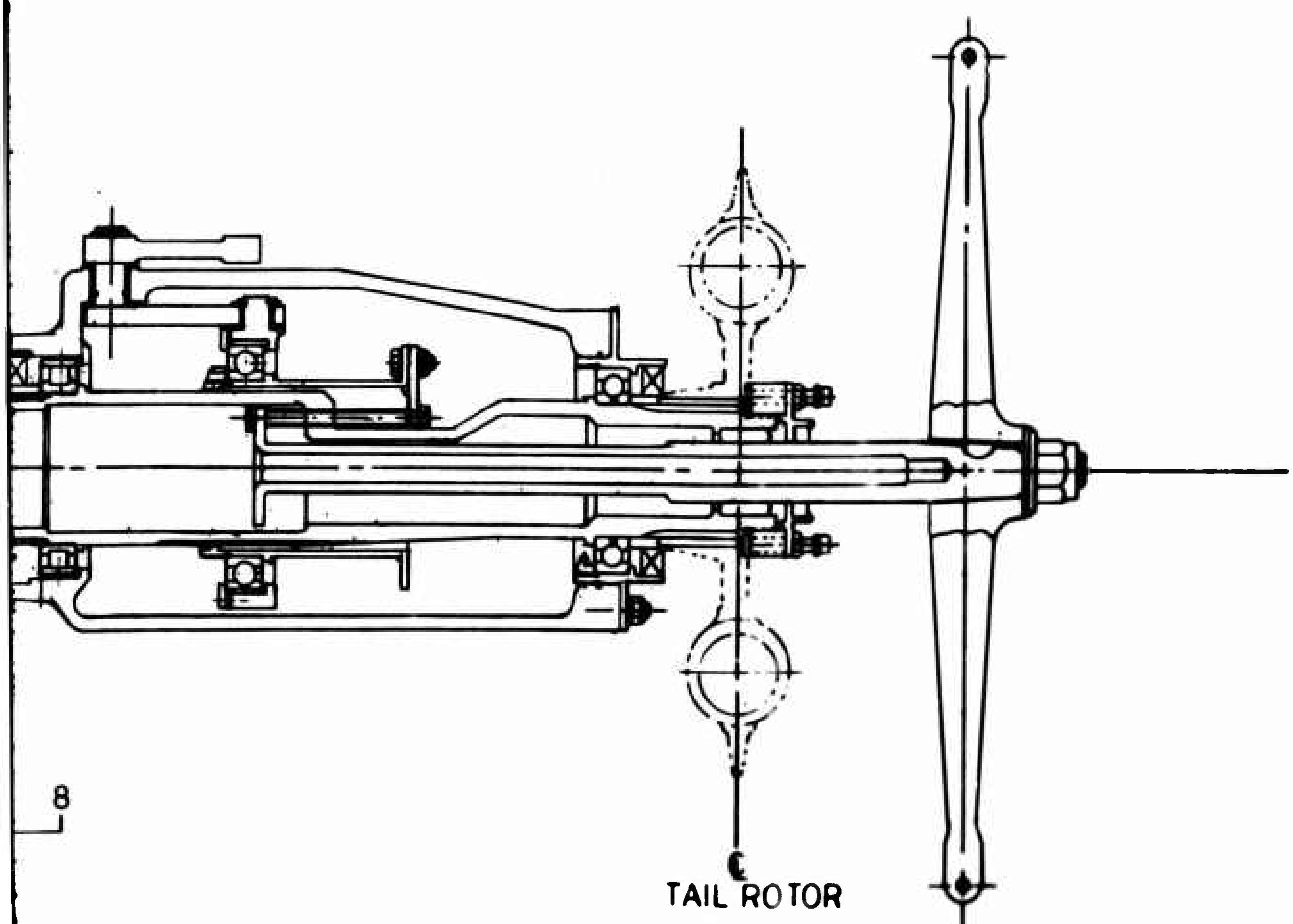


Figure 44. Propeller - Tail Rotor Gearbox Arrangement, Compound Helicopter.

A



B

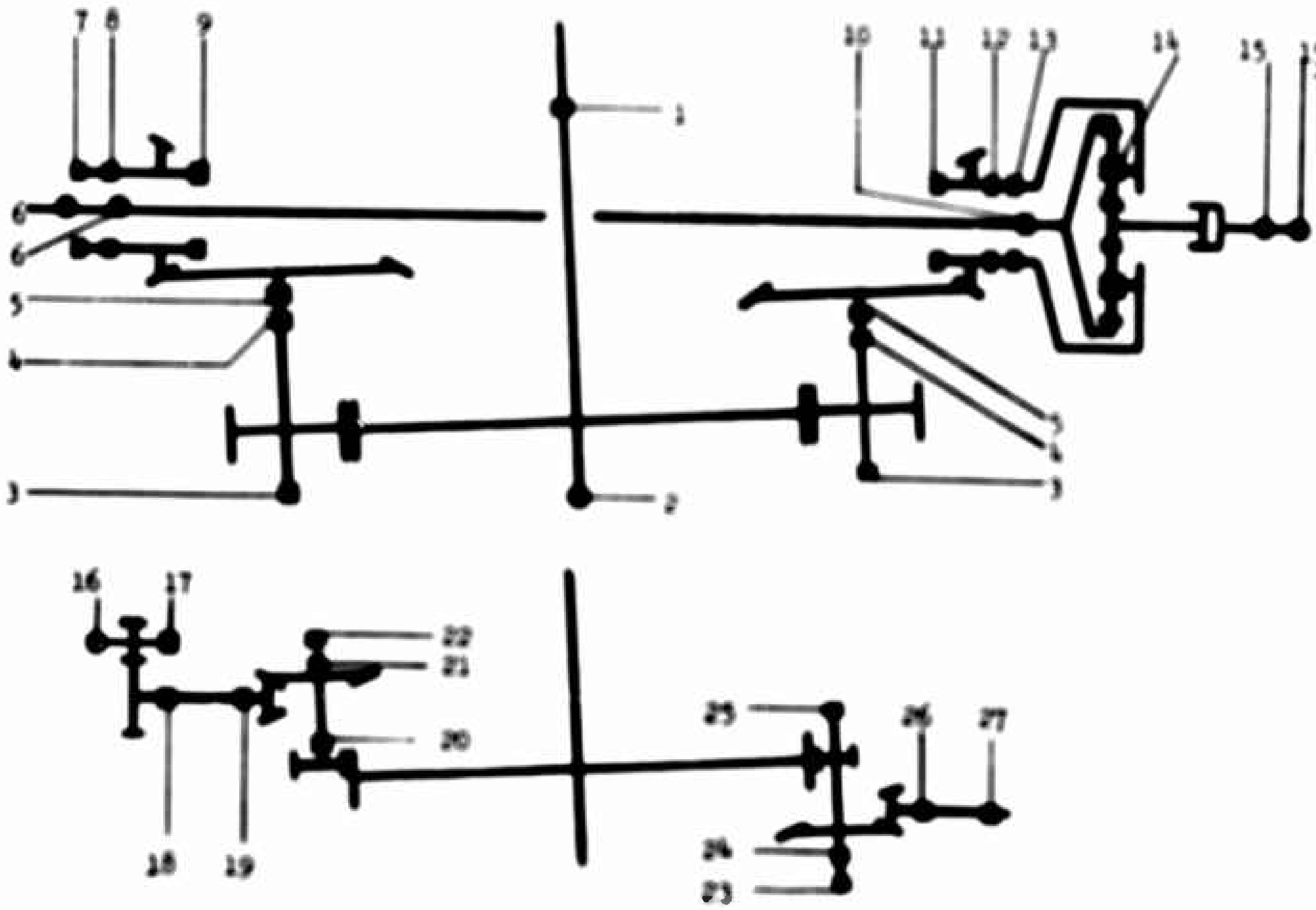


Figure 45. Bearing Data for Pure Helicopter Main Gearbox.

A

Bearing Position	Type of Bearing**	Size of Bearing (mm)	Speed rpm	B_{10} Life Unfactored (hours)
1	RB	210 x 260 x 24	224	8,000
2	BB	330 x 420 x 40	224	4,800*
3	RB	120 x 180 x 24	1199	3,400
4	TRB	178 x 227 x 30	1199	5,900
5	TRB	178 x 227 x 30	1199	6,700
6	BB	80 x 110 x 16	-	10,000
7	TRB	127 x 170 x 25	2713	3,500
8	TRB	127 x 170 x 25	2713	10,000
9	RB	80 x 125 x 22	2713	3,350
10	RB	65 x 90 x 13	5054	10,000
11	RB	80 x 125 x 22	2341	4,600
12	TRB	162 x 108 x 35	2341	3,300
13	TRB	146 x 108 x 22	2341	10,000
14	RB 2 Row	16 Rollers per Row 13 dia. x 13 long	10022	3,990
15	BB	85 x 120 x 18	30035	5,300
16	BB	45 x 75 x 16	8000	5,600
17	RB	45 x 75 x 16	8000	10,000
18	TRB	46 x 81 x 19	4210	5,800
19	TRB	51 x 81 x 18	4210	3,550
20	RB	110 x 140 x 16	1998	4,250
21	TRB	72 x 120 x 33	1998	3,900
22	TRB	72 x 120 x 33	1998	4,750
23	TRB	73 x 118 x 30	1427	4,200
24	TRB	73 x 118 x 30	1427	3,500
25	RB	60 x 95 x 18	1427	3,350
26	TRB	73 x 120 x 30	3005	3,400
27	TRB	70 x 113 x 22	3005	4,100

*B_{10} life x 3 using M-50 steel

**BB = Ball Bearing, RB = Roller Bearing, and TRB = Tapered Roller Bearing

B

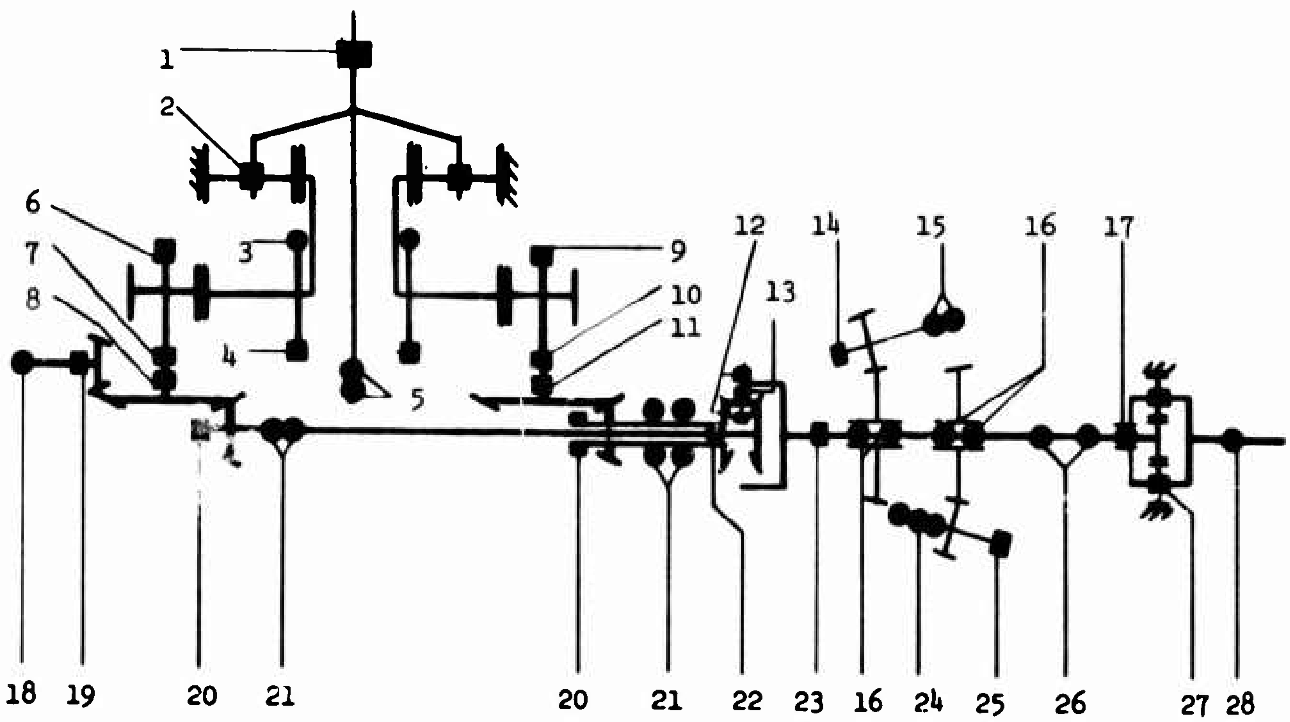


Figure 46. Bearing Data for Compound Helicopter Main Gearbox.

A

Bearing Position	Type of Bearing **	Size of Bearing (mm)	Speed (rpm)	B_{10} Life Unfactored (hours)
1	RB	170 x 220 x 20	245	11,000
2	RB 2 Row	20 Rollers per Row 12.7 dia x 26 long	810	3,690
3	RB	180 x 225 x 22	876	10,000
4	RB	170 x 225 x 22	876	10,000
5	Duplex	130 x 210 x 35	245	6,990*
6	RB	120 x 165 x 22	3016	3,850
7	TRB	171 x 222 x 25	3016	10,000
8	TRB	171 x 222 x 25	3016	10,000
9	RB	120 x 165 x 22	3016	3,100
10	TRB	171 x 222 x 25	3016	3,650
11	TRB	171 x 222 x 25	3016	3,050
12	NRB	22 x 35 x 19	-	10,000
13	NRB	22 x 42 x 3	..	10,000
14	RB	80 x 140 x 26	15188	3,100
15	Duplex	80 x 140 x 26	15188	3,200
16	BB	95 x 130 x 18	-	10,000
17	BB	90 x 125 x 18	4191	10,000
18	BB	65 x 100 x 18	8044	4,200
19	RB	85 x 120 x 18	8044	10,000
20	RB	85 x 130 x 22	5889	3,250
21	Duplex	120 x 180 x 28	5889	3,000
22	BB	50 x 72 x 12	-	10,000
23	RB	80 x 140 x 26	5889	4,300
24	Triplex	80 x 140 x 26	15188	2,520
25	RB	45 x 75 x 16	15188	3,300
26	Duplex	80 x 140 x 26	5889	10,000
27	RB	12 Rollers 12.7 dia x 26 long	5720	3,370
28	BB	90 x 125 x 18	1698	10,000

* B_{10} life x 3 using M-50 steel
**RB = Roller Bearing, BB = Ball Bearing
TRB = Tapered Roller Bearing, NRB = Needle Roller Bearing

B

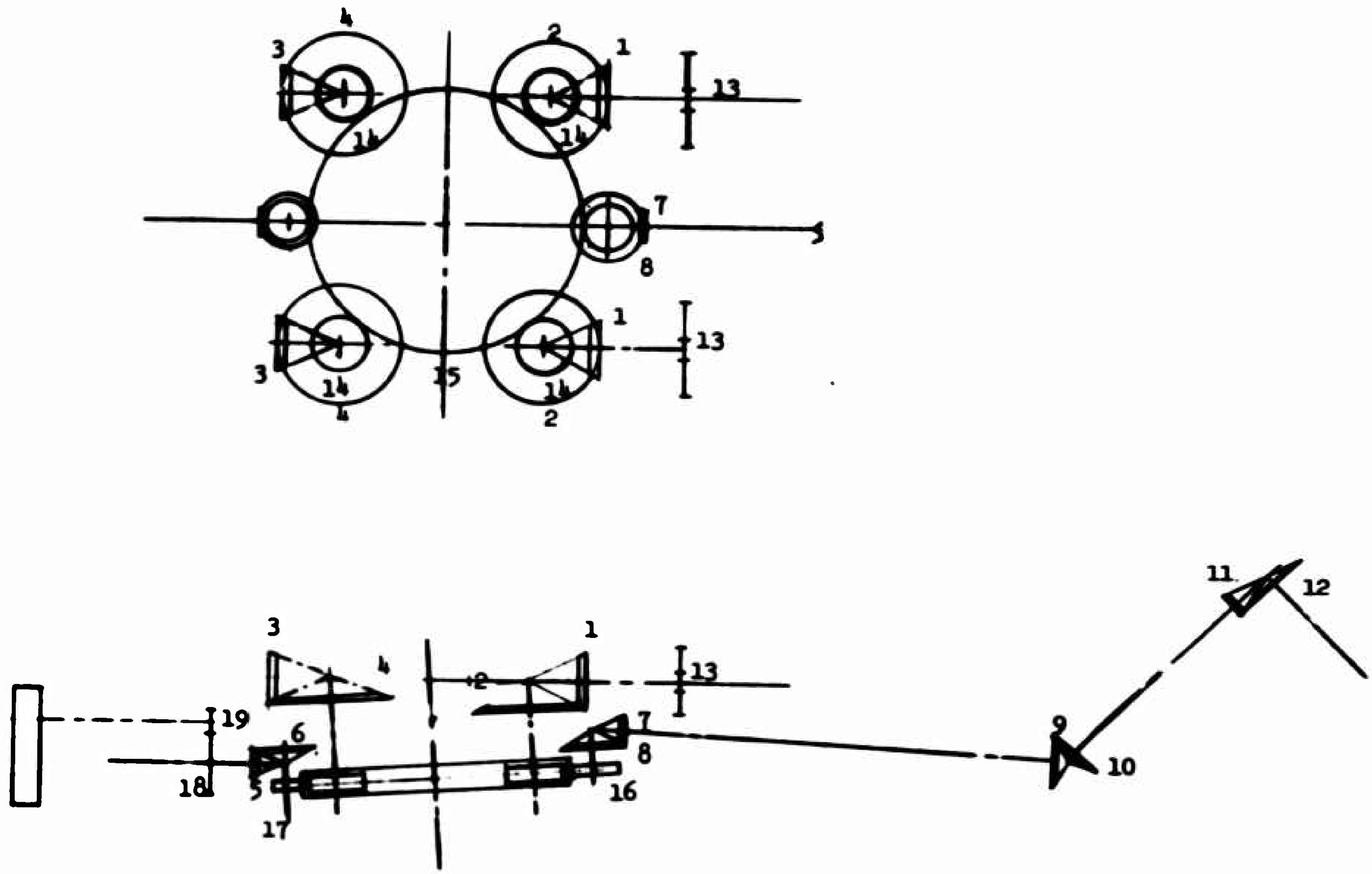


Figure 47. Gearing Schematic, Pure Helicopter Drive Train.

TABLE XXXI

SPIRAL BEVEL GEAR DAT

PURE HELICOPTER DRIVE TR

Item	Main Gearbox					
Name	Input Pinion	Output Gear	Input Pinion	Output Gear	Acc. TO Pinion	Acc. TO Gear
Location (See Fig. 47)	1	2	3	4	5	6
Diametral Pitch, p_d	2.838	2.838	2.921	2.921	8.80	8.80
Pitch Diameter (in.)	7.400	14.448	6.504	14.720	3.182	6.704
No. of Teeth (N)	21	41	19	43	28	59
Face Width (in.)	2.4	2.4	2.3	2.3	0.66	0.66
Pressure Angle	20°	20°	20°	20°	20°	20°
Spiral Angle	30°	30°	30°	30°	35°	35°
Shaft Angle	86°	86°	94°	94°	94°	94°
Pitch Angle	$26^\circ15'$	$59^\circ45'$	$24^\circ27'$	$69^\circ33'$	$26^\circ8'$	$67^\circ52'$
Hand of Spiral	LH	RH	LH	RH	LH	RH
Direction of Rotation	CW	CCW	CW	CCW	CW	CCW
Face Contact Ratio	1.48	1.48	1.47	1.47	1.51	1.51
Design hp	1010	1010	990	990	125	125
rpm	2341	1199	2713	1199	4210	1998
Bending Stress (psi)	26,900	27,000	27,400	27,500	21,000	21,000
Hertz Stress (psi)	198,000	198,000	201,000	201,000	130,000	130,000
Pitch Line Velocity (fpm)	4530	4530	4620	4620	3340	3340

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TABLE XXXI

SPIRAL BEVEL GEAR DATA,

URE HELICOPTER DRIVE TRAIN

n Gearbox				Intermediate Gearbox		Tail Rotor Gearbox	
Acc. TO Pinion	Acc. TO Gear	TTO Pinion	TTO Gear	Input Pinion	Output Gear	Input Pinion	Output Gear
5	6	7	8	9	10	11	12
8.80	8.80	4.75	4.75	4.923	4.923	5.25	5.25
3.182	6.704	4.00	8.421	6.50	7.110	4.00	5.25
28	59	19	40	32	35	21	55
0.66	0.66	1.4	1.4	1.18	1.18	1.7	1.7
20°	20°	20°	20°	20°	20°	20°	20°
35°	35°	35°	35°	35°	35°	35°	35°
94°	94°	82°30'	82°30'	133°	133°	90°	90°
26°8'	67°52'	23°55'	58°35'	60°40'	72°20'	20°54'	69°6'
LH	RH	RH	LH	RH	LH	RH	LH
CW	CCW	CW	CCW	CCW	CW	CCW	CW
1.51	1.51	1,70	1.70	1.52	1.52	2.30	2.30
125	125	370	370	370	370	370	370
4210	1998	3005	1427	3005	2747	2747	1049
21,000	21,000	27,500	27,300	30,000	29,800	28,500	28,300
130,000	130,000	130,000	194,000	197,000	197,000	194,000	194,000
3340	3340	3160	3160	5140	5140	2860	2860

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TABLE XXXII
SPUR GEAR SUMMARY,
PURE HELICOPTER DRIVE TRA

Component and Location	Diametral Pitch P_d	Pitch Dia (in.)	No. of Teeth (N)	Face Width (in.)	Pressure Angle	X Factor (function N and P_d)
Main Gearbox (see Figure 47)						
13 Planetary						
Sun	12	2.0833	25	1.850	22° 30'	.090
Pinion	12	4.6667	56	1.660	22° 30'	.101
Ring	12	11.4167	137	1.200	22° 30'	.128
Carrier						
14 Input Pinion	4	6.25	25	4.20	25°	.246
15 Output Gear	4	33.50	134	4.09	25°	.339
16 TTO Pinion	4	5.25	21	1.80	25°	.231
17 Acc Drive Pinion	4	3.75	15	0.805	25°	.201
18 Acc Gear	10	5.7	57	0.47	22° 30'	.115
19 Acc Pinion	10	3.0	30	0.46	22°30'	.096
Accessory Gearbox (see Figure 34)						
Input Pinion	10	4.3	43	.312	22° 30'	.107
Utility Pump	10	8.7	87	.25	22° 30'	.125
Servo Pump	10	8.2	82	.25	22° 30'	.123
Oil Pump	10	6.7	67	.25	22° 30'	.119
Generator	10	4.3	43	.25	22° 30'	.107
APU Input	10	4.3	43	.25	22° 30'	.107
Tach Gen Gear	10	5.3	53	.25	22° 30'	.113
Tach Gen Pinion	10	2.8	28	.25	22° 30'	.094

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RY,
TRAIN

or ion of Pd)	K Factor (function of root radius)	Design hp	rpm	Bending Stress (psi)	Hertz Stress (psi)	Pitch Line Velocity (fpm)
	1.265	2000	30,035	18,100	120,000	15,070
	1.293		-10,022	15,700	120,000	
	1.293	990	- 2,713	16,950	55,900	
		1010	+ 2,341			
	1.0	1010	1,199	24,500	144,500	1,960
	1.05	1010	223.67	19,100	144,500	1,960
	1.0	370	1,427	22,400	144,000	1,960
	1.0	125	1,998	19,500	129,000	1,960
	1.21	114	4,210	20,600	140,000	6,280
	1.17	114	8,000	23,800	140,000	6,280
	1.19	80	8,000	15,500	98,000	9,000
	1.25	78	3,951	6,200	98,000	9,000
	1.25	67	4,192	15,400	85,000	9,000
	1.22	56	5,130	15,600	85,000	9,000
	1.19	54	8,000	13,200	95,000	9,000
	1.19	80	8,000	19,600	111,000	9,000
	1.20	1	4,223	400	20,000	5,870
	1.17	1	8,000	425	20,000	5,870

B

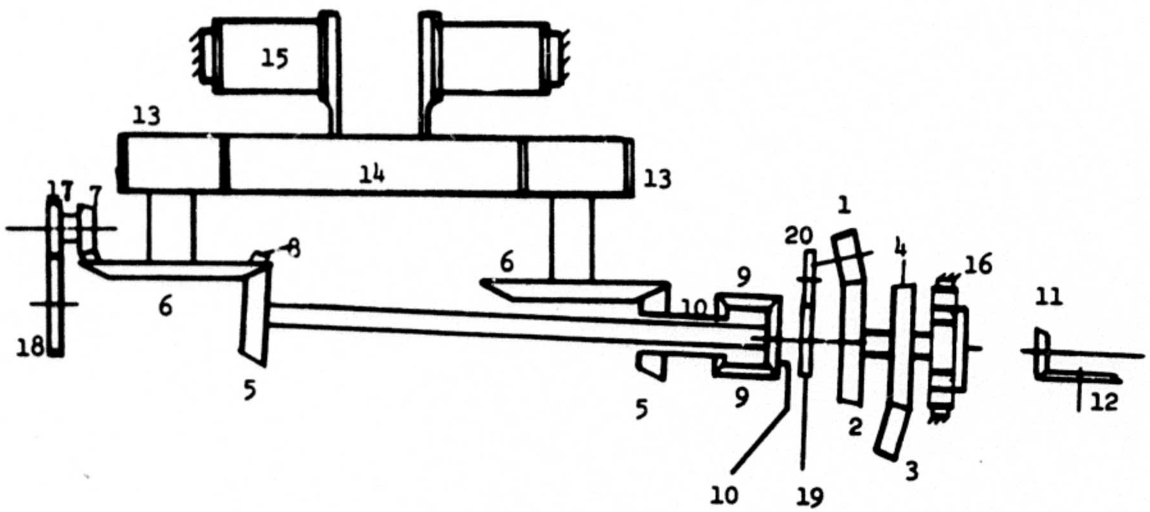

Figure 48. Gearing Schematic, Compound Helicopter Drive Train.

TABLE XXXIII
SPIRAL BEVEL GEAR DATA,
COMPOUND HELICOPTER DRIVE TRA

Item	Main Gearbox						
Name	Engine Input Pinion	Driven Cear	Engine Input Pinion	Driven Gear	Input Pinion	Output Gear	A H
Location (See Fig. 48)	1	2	3	4	5	6	7
Diametral Pitch, P_d	4.355	4.355	4.355	4.355	3.417	3.417	7
Pitch Diameter (in.)	4.362	11.250	4.362	11.250	6.146	11.999	4
No. of Teeth (N)	19	49	19	49	21	41	3
Face Width (in.)	1.75	1.75	1.75	1.75	2.125	2.125	0
Pressure Angle	20°	20°	20°	20°	20°	20°	2
Spiral Angle	33°	33°	33°	33°	35°	35°	4
Shaft Angle	15°	15°	15°	15°	91°	91°	9
Pitch Angle	4°11'	10°49'	4°11'	10°49'	27°20'	63°40'	2
Hand of Spiral	LH	RH	RH	LH	LH	RH	LH
Direction of Rotation	CW	CCW	CW	CCW	CCW	CW	CW
Face Contact Ratio	1.625	1.625	1.625	1.625	1.945	1.945	1.
Design hp	2000	2000	2000	2000	1650	1650	12
rpm	15,188	5,889	15,188	5,889	5,889	3,016	8,
Bending Stress (psi)	22,900	22,900	22,900	22,900	27,900	27,900	11
Hertz Stress (psi)	202,000	202,000	202,000	202,000	190,000	190,000	90
Pitch Line Velocity (fpm)	17,400	17,400	17,400	17,400	9,500	9,500	9,

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				Tail Rotor Gearbox	
c. TO nion	Acc. TO Gear	Torque Divider Pinion	Torque Divider Side Gear	Input Pinion	Output Gear
	8	9	10	11	12
517	7.517	6.0	6.0	3.375	3.375
39	11.71	2.5	4.33	8.0	12.741
	88	15	27	29	47
62	0.62	0.625	0.625	2.1	2.1
°	20°	20°	20°	20°	20°
°	40°	0°	0°	35°	35°
°	90°	90°	90°	90°	90°
°34'	69°26'	29°3'	59°57'	31°40'	58°20'
	RH	LH	RH	RH	LH
	CCW	-	-	CCW	CW
47	1.47	-	-	1.85	1.85
5	125	550	550	320	320
044	3,016	0	3,016	1,698	1,042
,000	11,000	99,700	99,700	27,900	27,900
,000	90,000	-	-	195,000	195,000
260	9,260	0	0	3,560	3,560

B

TABLE XXXIV
SPUR GEAR SUMMARY,
COMPOUND HELICOPTER DRIVE TRA

Component and Location (See Figure 48)	Diametral Pitch P_d	Pitch Dia. (in.)	No. of Teeth (N)	Face Width (in.)	X Factor (function of N and P_d)	K Fac (functi root r
13 Spur Pinion	6	6.000	36	3.60	.168	1.00
14 Spur Gear	6	20.667	124	3.56	.213	1.05
15 Planetary (RR = 3.57)						
Sun	6	7.833	47	4.34	.204	1.00
Pinion (6 req'd)	6	6.167	37	4.09	.198	1.00
Ring	6	20.167	121	2.06	.237	1.00
Carrier						
16 Planetary (RR = 3.46)						
Sun	14	5.071	71	1.73	.096	1.38
Pinion (6 req'd)	14	3.714	52	1.48	.094	1.37
Ring	14	12.500	175	1.30	.107	1.46
Carrier						
17 Servo Pump Pinion	8	4.750	38	0.25	.129	1.09
18 Servo Pump Gear	8	12.625	101	0.25	.159	1.20
19 Spur Pinion	10	8.40	84	0.25	.124	1.25
20 Spur Gear	10	8.50	85	0.25	.124	1.25
Accessory Gearbox (same as Pure Helicopter Gearbox)						

Note: All of the above spur gears have a pressure angle of $22^{o}30'$.

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XXXIV
SUMMARY,
ER DRIVE TRAIN

of d)	K Factor (function root radius)	Design hp	rpm	Bending Stress (psi)	Hertz Stress (psi)	Pitch Line Velocity (fpm)
	1.00	1650	3016	28,800	143,000	4,750
	1.05	1650	876	25,600	143,000	
	1.00	3300	876	18,000	144,000	1,300
	1.00		810	18,500	144,000	
	1.00		0	30,700	127,500	
			245			
	1.38	2700	5889	27,600	120,500	5,570
	1.37		5720	28,000	120,500	
	1.46		0	28,000	114,000	
			1698			
	1.09	11	8044	1,850	35,300	10,000
	1.20	11	3016	1,650	35,300	
	1.25	10	5889	1,550	26,800	13,000
	1.25	10	5820	1.550	26,800	

B

TABLE XXXV
REPRESENTATIVE POWER SPECTRUM,
PURE HELICOPTER

Mission Environment: 4,000 feet, 95°F

Description	Percent Time	Input hp	Main Rotor hp	Tail Rotor hp	Time (Minutes)
Warm-up, takeoff	1.20	2700	2290	216	2
Cruise at 150 kn	64.00	1490	1266	60	
1.5 g man.	4.15	2017	1713	80	115.5
2.0 g man.	0.69	2544	2164	102	
2.5 g man.	0.35	3070	2610	122	
Dash at 157 kn	9.15	1538	1307	61	
1.5 g man.	0.59	2049	1741	82	16.5
2.0 g man.	0.10	2560	2176	102	
2.5 g man.	0.05	3070	2610	122	
Loiter at 80 kn	3.00	1188	1010	72	
1.5 g man.	0.30	1815	1543	109	5.7
2.0 g man.	0.05	2442	2076	146	
2.5 g man.	0.05	3070	2610	184	
Hover OGE	4.90	2734	2323	219	
Climb	6.47	2915	2476	233	
Turns	1.65	3070	2610	120	27.3
	1.65	3070	2610	245	
	1.65	3070	2610	370	
	100.00				167.0

TABLE XXXVI
REPRESENTATIVE POWER SPECTRUM,
PURE HELICOPTER

Mission Environment: Sea Level Standard Day

Description	Percent Time	Input hp	Main Rotor hp	Tail Rotor hp	Time (Minutes)
Warm-up, Takeoff	1.20	3620	3080	290	2
Cruise at 150 kn	64.00	2004	1706	80	
1.5 g man.	4.15	2669	2277	107	115.5
2.0 g man.	0.69	3334	2835	133	
2.5 g man.	0.35	4000	3400	160	
Dash at 157 kn	9.15	2199	1869	88	
1.5 g man.	0.59	2799	2380	112	16.5
2.0 g man.	0.10	3400	2890	136	
2.5 g man.	0.05	4000	3400	160	
Loiter at 80 kn	3.0	1579	1342	95	
1.5 g man.	0.30	2386	2030	143	
2.0 g man.	0.05	3193	2712	192	5.7
2.5 g man.	0.05	4000	3400	240	
Hover OGE	4.90	3279	2787	262	
Climb	6.47	3496	2972	280	27.3
Turns	4.95	4000	3400	320	
	100.00				167.0

TABLE XXXVII
REPRESENTATIVE SPECTRUM,
ROTOR FLAPPING ANGLE - β
PURE HELICOPTER

Mission Environment: 4,000 feet, 95°F

Description	Percent Time	$\pm\beta$ Degrees Main Rotor	$\pm\beta$ Degrees Tail Rotor
Warm-up, Takeoff	1.20	3.2	2.05
Cruise at 150 kn	64.00	2.4	1.55
man.	2.22	2.4	1.1
man.	2.22	2.5	1.85
man.	0.75	4.25	1.55
Dash at 157 kn	9.15	2.4	1.55
man.	0.32	2.4	1.1
man.	0.31	2.5	1.85
man.	0.11	4.25	1.55
Loiter at 80 kn	3.00	2.4	1.55
man.	0.17	2.4	1.1
man.	0.18	2.5	1.85
man.	0.05	4.25	1.55
Hover	4.90	2.35	1.33
man.	3.82	3.2	4.5
man.	3.80	2.4	3.5
man.	3.80	5.9	1.33
	100.00		

TABLE XXXVIII
REPRESENTATIVE SPECTRUM,
ROTOR FLAPPING ANGLE – β
PURE HELICOPTER

Mission Environment: Sea Level Standard Day

Description	Percent Time	$\pm\beta$ Degrees Main Rotor	Tail Rotor
Warm-up, Takeoff	1.20	3.9	2.45
Cruise at 150 kn	64.00	2.9	1.85
man.	2.22	2.9	1.3
man.	2.22	3.0	2.2
man.	0.75	5.15	1.85
Dash at 157 kn	9.15	2.9	1.85
man.	0.32	2.9	1.3
man.	0.31	3.0	2.2
man.	0.11	5.15	1.85
Loiter at 80 kn	3.00	2.9	1.85
man.	0.17	2.9	1.3
man.	0.18	3.0	2.2
man.	0.05	5.15	1.85
Hover	4.90	2.8	1.6
man.	3.82	3.9	5.4
man.	3.80	2.9	4.2
man.	3.80	7.05	1.6
	100.00		

TABLE XXXIX
REPRESENTATIVE POWER SPECTRUM,
COMPOUND HELICOPTER

Mission Environment: 4,000 feet, 95°F

Description	Percent Time	Input hp	Main Rotor hp	Tail Rotor hp	Propeller	Time (Minutes)
Warm-up, Takeoff	1.5	2700	2226	216	67	2
Cruise at 200 kn	59.8	2307	441	75*	1522	
1.5 g man.	3.88	2561	486	75*	1690	86.6
2.0 g man.	0.64	2815	534	75*	1857	
2.5 g man.	0.33	3070	583	75*	2024	
Dash at 210 kn	8.64	2480	474	75*	1633	
1.5 g man.	0.56	2677	510	75*	1763	12.5
2.0 g man.	0.09	2874	546	75*	1894	
2.5 g man.	0.05	3070	583	75*	2024	
Loiter at 80 kn	3.74	774	504	75*	154	
1.5 g man.	0.40	1539	999	75*	315	5.7
2.0 g man.	0.05	2304	1494	110	476	
2.5 g man.	0.05	3070	1989	146	637	
Hover OGE	6.10	2762	2277	221	69	
Climb	8.01	2916	2405	234	73	27.2
Turn	6.16	3070	2536	246	77	
	100.00					134.0

*Theoretical tail rotor horsepowers are very low because of low antitorque requirements. However, 75 horsepower is considered a minimum power requirement to rotate the tail rotor during cruise.

TABLE XL
REPRESENTATIVE POWER SPECTRUM,
COMPOUND HELICOPTER

Mission Environment: Sea Level Standard Day

Description	Percent Time	Input hp	Main Rotor hp	Tail Rotor hp	Propeller	Time (Minutes)
Warm-up, Takeoff	1.5	3620	2986	290	90	2
Cruise at 200 kn	59.8	2840	544	75*	1822	
1.5 g man.	3.88	3227	618	75*	2126	
2.0 g man.	0.64	3614	692	75*	2380	86.6
2.5 g man.	0.33	4000	766	75*	2633	
Dash at 210 kn	8.64	3052	584	75*	2008	
1.5 g man.	0.56	3368	645	75*	2216	12.5
2.0 g man.	0.09	3684	705	75*	2424	
2.5 g man.	0.05	4000	766	75*	2633	
Loiter at 80 kn	3.74	864	566	75*	174	
1.5 g man.	0.40	1909	1252	90	392	5.7
2.0 g man.	0.05	2955	1938	140	510	
2.5 g man.	0.05	4000	2623	192	828	
Hover OGE	6.10	3277	2695	262	82	
Climb	8.01	3462	2855	277	86	27.2
Turn	6.16	4000	3300	320	100	
	100.00					134.0

*Theoretical tail rotor powers are very low. However, 75 horsepower is considered a minimum power requirement to rotate the tail rotor during cruise.

TABLE XLI
REPRESENTATIVE SPECTRUM,
ROTOR FLAPPING ANGLE - β
COMPOUND HELICOPTER

Mission Environment: 4,000 feet, 95°F

Description	Percent Time	$\pm\beta$ Degrees Main Rotor	Tail Rotor
Warm-up, Takeoff	1.50	2.85	2.0
Cruise at 200 kn	59.80	1.05	0.1
man.	2.02	1.05	0.4
man.	2.02	1.25	0.5
man.	0.81	2.75	0.1
Dash at 210 kn	8.60	1.00	0.1
man.	0.32	1.00	0.4
man.	0.32	1.10	0.5
man.	0.10	2.65	0.1
Loiter	3.74	2.15	1.5
man.	0.22	2.15	1.1
man.	0.22	2.25	1.8
man.	0.06	3.70	1.5
Hover	6.10	2.10	1.3
man.	4.72	2.85	4.35
man.	4.72	2.15	3.35
man.	4.73	5.25	1.3
	100.00		

TABLE XLII
REPRESENTATIVE SPECTRUM,
ROTOR FLAPPING ANGLE – β
COMPOUND HELICOPTER

Mission Environment: Sea Level Standard Day

Description	Percent Time	$\pm\beta$ Degrees Main Rotor	Tail Rotor
Warm-up, Takeoff	1.50	3.50	2.40
Cruise at 200 kn	59.80	1.30	0.10
man.	2.02	1.30	0.40
man.	2.02	1.50	0.50
man.	0.81	3.30	0.10
Dash at 210 kn	8.60	1.25	0.10
man.	0.32	1.25	0.40
man.	0.32	1.35	0.50
man.	0.10	3.15	0.10
Loiter	3.74	2.55	1.80
man.	0.22	2.55	1.20
man.	0.22	2.65	2.15
man.	0.06	4.40	1.80
Hover	6.10	2.55	1.55
man.	4.72	3.40	5.20
man.	4.72	2.55	4.00
man.	4.73	6.35	1.55
	100.00		

TABLE XLIII
ACCESSORY DRIVE REQUIREMENTS,
PURE AND COMPOUND HELICOPTERS

Accessory	Required hp	Drive Pad Specification
Generator No. 1	33.5 (or 53.6)	AND 20002
Generator No. 2	53.6 (or 33.5)	AND 20002
Tachometer	1.0	AND 20005
Oil Pump	2.0	SIKORSKY
Primary Servo Pump	11.0	AND 20001
Auxiliary Servo Pump	11.0	AND 20001
Utility Pump	11.0	AND 20001

APPENDIX III
ALGEBRAIC DESCRIPTION OF SPLIT POWER TRANSMISSIONS

RELATIONS FOR COUNTERROTATING EPICYCLIC GEAR

Speed Ratios

If all members of an epicyclic gear rotate, the speeds of each member are related by

$$rR + sS = c(R + S) \tag{2}$$

where r, s, and c denote the rotational speed of the ring gear, sun gear, and planet carrier respectively. Equation (2) can be derived by any conventional method. The direction of rotation for r,s, and c must be taken into account by using plus and minus signs for forward and reverse directions. R and S denote the pitch diameters, or numbers of teeth, in the ring gear and sun gear respectively.

Normally the ring gear of an epicyclic unit is fixed; thus, putting r = o in equation (2) gives the well-known result

$$\frac{s}{c}_{r=o} = \frac{R}{S} + 1 \tag{3}$$

Similarly, with a fixed planet carrier, c = 0 in equation (2) and the reduction ratio is

$$\frac{s}{r}_{c=o} = -\frac{R}{S} \tag{4}$$

Now if no member is fixed, the simple relationships given above by equations (3) and (4) are no longer applicable. One advantage of counterrotation, as used in various configurations in the body of this report, is that greater reduction ratios than are defined by equations (3) and (4) can be obtained even though the ring-gear-to-sun-gear ratio, R/S, remains unaltered.

Direction of Power Flow

To obtain a division of input power such that the power transmitted by the ring gear and the planet carrier are both less than the power input to the sun gear, the directions of rotation must be as shown in Figure 49. Then there will be no recirculation of power within the epicyclic unit since the output torques from the ring gear and the planet carrier act in the same sense as their respective directions of rotation. It will be noted from Figure 49 that for a given input torque to the sun gear, T_s, the torques delivered by the ring gear and the planet carrier are respectively, $(-R/S)T_s$ and $T_s(R + S)/S$, the negative sign indicating a reversal of direction.

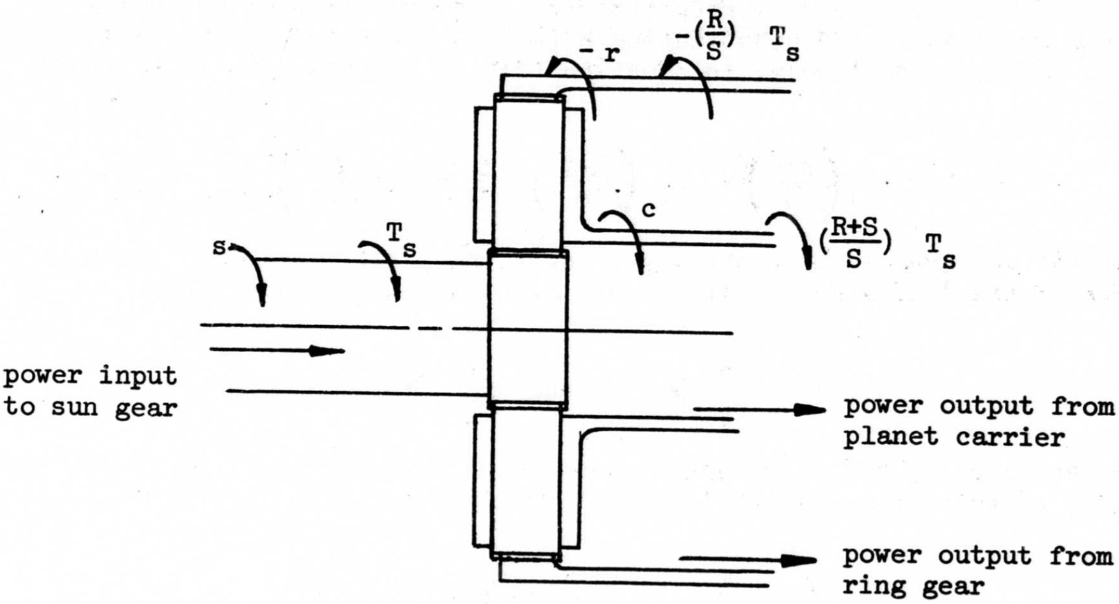

Figure 49. Directions of Rotation To Avoid Recirculation of Power.

Distribution of Input Power

The magnitude of power carried in each drive line can be found by multiplying the torque in the drive line by its speed of rotation. It follows from Figure 49 that:

$$\frac{p_c}{p_s} = \frac{c}{s}\left(\frac{R}{S} + 1\right) \tag{5}$$

$$\frac{p_r}{p_s} = - \frac{r\ R}{s\ S} \tag{6}$$

from which

$$\frac{p_c}{p_r} = - \frac{c}{r}\left(1 + \frac{S}{R}\right) \tag{7}$$

A more formal demonstration that the directions of rotation shown in Figure 49 avoid recirculation of power can be conveniently obtained from equation (11) of Reference 6. In this analysis R in equation (11) of Reference 6 is replaced by R' in order to distinguish it from R, the ring gear diameter. Defining R' as the ratio of planet carrier speed to sun gear speed at $r = o$, it follows that $R' = c/s = S/(R + S)$.

A comparison of R in Reference 6 with R' shows that n_1, n_2, and n_3 must be equivalent to s, r, and c, respectively. Then by assuming no losses (unity efficiency) and substituting S/(R + S) for R and n_1, n_2, and n_3 for s, r, and c, respectively, in equation (11) of Reference 6, it is found that

$$P_1\left(\frac{R+S}{S}\right) = P_2\left(\frac{R+S}{R}\right)\frac{s}{r} = -P_3\left(\frac{s}{c}\right) \qquad (8)$$

When the sun gear carries the input power, no recirculation of power will occur if the following conditions are simultaneously satisfied.

$$\frac{p_c}{p_s} > 0 \text{ and } \frac{p_r}{p_s} > 0 \qquad (9)$$

The ratios p_c/p_s and p_r/p_s can be identified in Reference 6 as $-P_3/P_1$ and $-P_2/P_1$ respectively. It follows that equation (8) transforms into

$$p_s\left(\frac{R+S}{S}\right) = -p_r\left(\frac{R+S}{R}\right)\frac{s}{r} = p_c\left(\frac{s}{c}\right)$$

Thus, $\frac{p_c}{p_s} = \left(1 + \frac{R}{S}\right)\frac{c}{s}$, which is positive when the planet carrier rotates in the same direction as the sun gear. $\frac{p_r}{p_s} = -\frac{Rr}{Ss}$ is also positive, providing that the ring gear rotates in the opposite direction to the sun gear. Therefore, both of the conditions of equation (9) assuring no recirculation of power have been satisfied with the directions of rotation shown in Figure 49.

Equal and Opposite Speeds of Ring Gear and Planet Carrier

In an arrangement such as that shown in Figure 49, the ring gear and the planet carrier can be made to rotate at any desired speed. An obvious choice, however, is for equal and opposite speeds of the ring gear and planet carrier, as further gear reductions in the main rotor drive train can have the same ratio in each drive line. Then putting r = -c in equation (2) gives

$$\frac{s}{c} = \left(\frac{2R}{S}\right) + 1$$

which can be rewritten as

$$\frac{s}{c} = \left(\frac{R}{S} + 1\right) + \frac{R}{S} \qquad (10)$$

The ratio of sun gear speed to planet carrier speed can be seen from equation (10) to be the reduction obtained if the ring were fixed, i.e., (R/S) + 1, plus the reduction obtained if the planet carrier were fixed, i.e., R/S. This facility of obtaining a large speed reduction is one reason for the use of a counterrotating planetary unit.

The relative amount of power transmitted by each drive line is given by

$$\frac{p_c}{p_r} = \frac{c\ T_c}{r\ T_r} = \angle \frac{c}{r}\left(\frac{S}{R} + 1\right)$$

and if c = - r, as in the case being discussed,

$$\frac{p_c}{p_r} = \left(\frac{S}{R} + 1\right) \tag{11}$$

It is seen from equation (11) that the planet carrier and the ring gear drive lines cannot be designed to transmit equal power (when $c = -r$), but the drive powers approach each other as $S \ll R$, which is also a condition for large reduction ratios.

Condition for Planet Carrier and Ring Gear to Transmit Equal Power

Equal power can be transmitted by the planet carrier and the ring gear by arranging these members to rotate at certain speeds. Disregarding losses, it then follows that the two power flows will each be equal to half the power input to the sun gear. The condition for $p_c = p_r$ is found from equation (7) to be

$$r = -c\left(\frac{S}{R} + 1\right) \tag{12}$$

Therefore, the ring gear must rotate faster than the planet carrier by the factor (S/R) +1 and also in an opposite direction.

Reduction Ratios When Input Power is Split into Two Equal Parts

The condition for equal power outputs from the planet carrier and the ring gear is given by equation (12). If this condition is substituted in equation (2), then

$$\frac{s}{r} = -\frac{2R}{S} \tag{13}$$

and

$$\frac{s}{c} = 2\left(\frac{R}{S} + 1\right) \tag{14}$$

It follows from equations (3), (4), (13), and (14) that with equal power outputs,

(a) The ring gear speed reduction is twice that of a fixed planet carrier system.

(b) The planet carrier reduction is twice that of a fixed ring gear system.

FREEWHEEL ACTION AND CAPACITY

In a conventional transmission the freewheel necessary for each engine must be sized to carry the rated power of the engine. With a split-power transmission, however, a distinctly advantageous feature is that the freewheel can be positioned after the power split, thereby allowing a freewheel of reduced torque capacity. Some caution must be used when positioning the freewheel since, as the following paragraphs show, the overrun speed of the freewheel varies according to whether the freewheel is driven by the ring gear or the planet carrier. In either case, however, on shutting-down an engine, the counterrotation ceases and the ring gear and the planet carrier rotate in the same direction.

Location of Freewheel Unit

(1) Freewheel Unit Driven by Ring Gear of Epicyclic Unit

In all the split-power transmissions considered, the sun gear of the counterrotating epicyclic unit carries full engine power. Since the torque delivered by the ring gear is always less than that delivered by the planet carrier, it is logical to place the freewheel after the ring gear unless assembly and accessibility problems preclude this position.

When the freewheel is engaged the ring gear and the freewheel rotate together at speed r rpm in a sense opposite to that of the sun gear and planet carrier; equation (2) rearranged shows that

$$r = c\left(1 + \frac{S}{R}\right) - s\ \frac{S}{R} \qquad (15)$$

Now, on engine shutdown, the sun gear speed is zero, which gives

$$r_{s=o} = c\left(1 + \frac{S}{R}\right) \qquad (16)$$

The planet carrier still rotates at the speed given by equation (15), since it is driven by the main rotor. Thus, on engine shutdown, with s = o, the ring gear and the planet carrier rotate in the same direction as a result of the term 1 + (S/R) in equation (16) always being positive.

It follows that the members of the freewheel will have the speeds shown in Figure 50; the relative speed across the freewheel will be the difference of the speeds shown, i.e., freewheel overrun speed = $s \frac{S}{R}$ rpm.

(2) Freewheel Driven by Planet Carrier of Epicyclic Gear

In this case the freewheel is placed as shown in Figure 51 and, on engine shutdown, the ring gear continues to rotate at its normal speed since it is still connected by gearing to the main rotor. When the freewheel is driving, the planet carrier speed is seen from equation (2) to be

$$c = \frac{rR + sS}{R + S}$$

and on stopping the engine,

$$c_{s=o} = \frac{rR}{R + S},$$

which indicates that at s = o, c and r rotate in the same direction.

Referring to Figure 51, it follows that the relative speed across the freewheel will be

$$\text{freewheel overrun speed} = \frac{sS}{R + S}$$

In conclusion, it should be noted that, in a split-power transmission of the type under consideration, the power capacity of the freewheel need be equal only to the power delivered by the ring gear, or the power delivered by the planet carrier, according to the freewheel position. But the overrun speed of the freewheel will be greater than its driving speed, such that the product of freewheel torque capacity and overrun speed is equal to the engine power rating.

POWER DISTRIBUTION IN TRANSMISSION CONFIGURATIONS 6 and 12

From Figure 52 it is seen that the counterrotating bevel gears driven by members of the input epicyclic unit mesh with a combining, or output, bevel gear designated B_o. Thus the pitch line velocities of all 3 bevel gears must be equal. Any engine slope, however, means that the counterrotating bevel gears will be of different diameter and must, therefore, have different rotational speeds. The following analysis shows how the sizes of bevel gears are related to the power distribution, the engine slope, and the reduction ratio from the engine to the combining bevel gear.

Initially, a geometric similarity to Figure 52 is assumed; that is, the engine slope and directions of rotation are such as to permit the planet

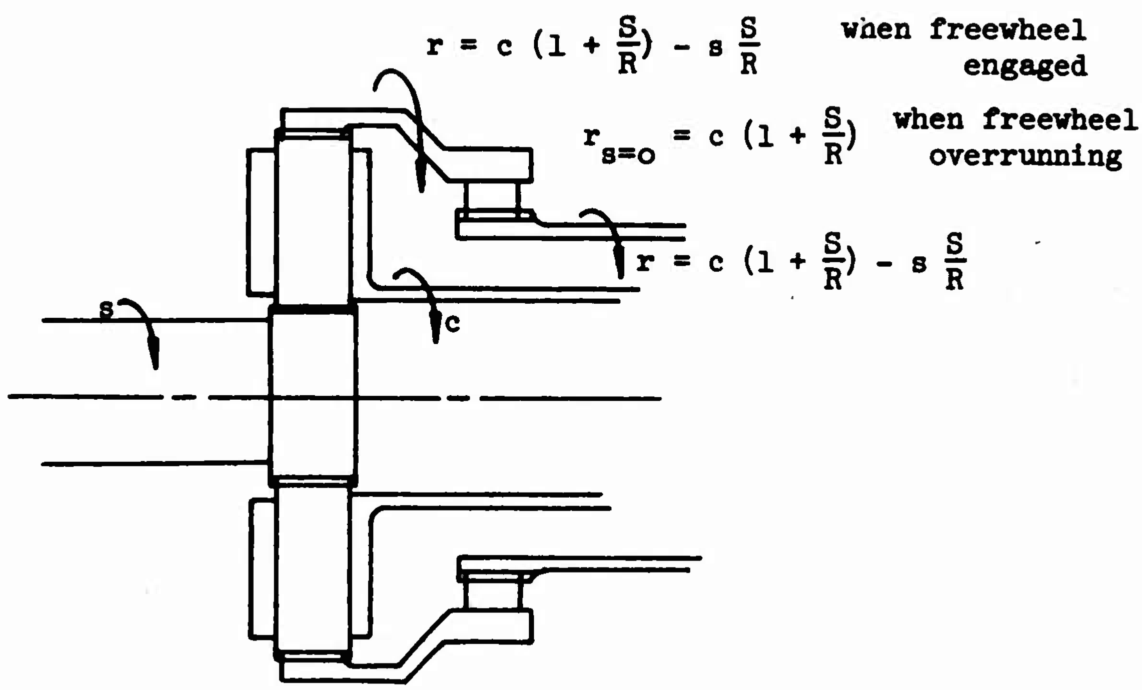

Figure 50. Speeds When Freewheel Is Driven by Ring Gear.

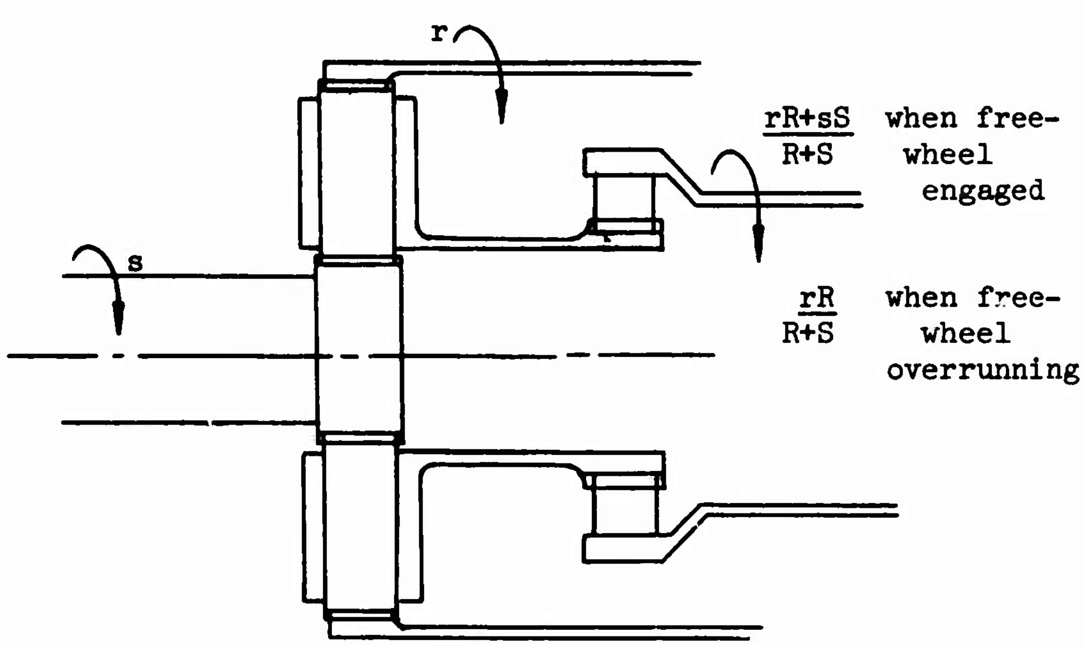

Figure 51. Speeds When Freewheel Is Driven by Planet Carrier.

carrier to be coupled to the larger diameter bevel gear as shown in Figure 52. If n_o is the speed of the output or combining spiral bevel, and if the planet carrier rotates in the same (positive) direction as the sun gear, then the planet carrier speed is

$$c = n_o B_o/B_c \tag{17}$$

and the ring gear speed is

$$r = -n_o B_o/B_r. \tag{18}$$

Eliminating c and r between equations (2), (17) and (18) gives

$$\frac{s}{n_o} = \frac{B_o}{B_c}\left[\frac{R}{S}\left(\frac{B_c}{B_r} + 1\right) + 1\right] \tag{19}$$

The distribution of engine input power between the planet carrier and the ring gear drive paths is

$$\frac{P_c}{P_r} = \frac{B_r}{B_c}\left(1 + \frac{S}{R}\right) \tag{20}$$

and the relative size of B_r and B_c will vary according to the separation fixed by B_o and the angle θ, where θ is defined as the included shaft angle between the axes of the input and output bevel gears, as shown in Figure 52. It is seen from Figure 52 that

$$2 B_o \cos \theta = B_c - B_r. \tag{21}$$

Eliminating B_c between equations (20) and (21) gives

$$\frac{P_c}{P_r} = \frac{B_r \quad (R + S)}{R\ 2\ B_o \cos \theta + B_r} \tag{22}$$

The conditions for half engine power to be transmitted by the planet carrier and the ring gear are found by putting equations (20) and (22) equal to unity; in this case, from equation (20),

$$B_c = B_r\left(1 + \frac{S}{R}\right) \tag{23}$$

and from equation (22),

$$SB_r = 2 R B_o \cos \theta. \tag{24}$$

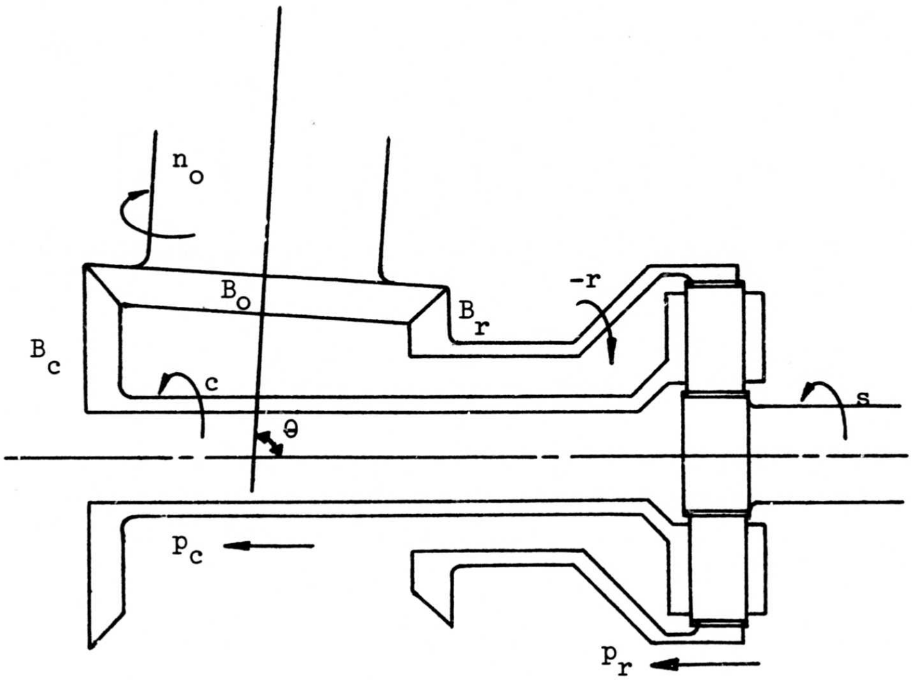

Figure 52. Arrangement of Epicyclic Unit and Bevel Gears in Configurations 6 and 12.

APPENDIX IV
MAIN GEARBOX DESIGN INCORPORATING "FREE-PLANET" PLANETARY

INTRODUCTION

This appendix briefly describes an alternative lightweight main transmission that includes a novel and efficient planetary gear with a high reduction ratio. This "free-planet" planetary reduction concept eliminates the planet carriers and planet pinion bearings and the failure patterns associated therewith. Several facets of this design have yet to be tested experimentally, and for this reason the transmission concept could not be considered for production at the beginning of the 1970 to 1975 time frame. Because of its potential advantages, however, the transmission concept has been evaluated for the pure helicopter arrangement depicted in Figure 1. This appendix is a complete entity, including symbols and terminology, since the "free-planet" concept involves unique definitions and analyses.

DESIGN REQUIREMENTS

The gearbox design embodying this free-planet concept is based on preliminary design data as listed below. No iterative design procedures were conducted since the goals of this effort were to explore the concept and define a gearbox incorporating the free-planet concept.

Bearings

Input Bevel	1135 hp at 30,000 rpm
Rotor Shaft	Thrust = 19,620 lb
	Horizontal Force = 605 lb
	Head Moment = 115,000 lb-in.
Tail Rotor Takeoff	$<$ 400 hp at 6000 rpm
Accessory Drive	125 hp at 6000 rpm

Gearing

Input Bevel Gears	2000 hp at 30,000 rpm
Planetary Reduction	3600 hp at 224 rpm output
Tail Rotor Drive	$<$ 400 hp at 6000 rpm
Accessory Drive	125 hp at 6000 rpm

Rotor Shaft

Thrust	19,620 lb
Horizontal Force	1,380 lb
Head Moment	260,000 in.-lb
Output Torque	1,012,000 in.-lb

Overruning Clutches

Torque	4200 in.-lb at 30,000 rpm

GEARBOX DESCRIPTION

The gearbox shown in Figure 53 incorporates this free-planet planetary reduction concept. It also utilizes overrunning spring clutches which are light in weight and permit high operating and overrunning speeds. A large speed reduction is taken at the input bevel gear mesh, since this gives a relatively simple lightweight gearbox in which only 5 gear meshes are required to achieve the desired 134:1 overall ratio. Accessory and tail rotor drives are simple and direct.

While concept feasibility has not been demonstrated experimentally in a transmission environment, related experience with "Power Hinges" and extensive analyses show that the fundamental concepts involved are sound. Allowable stress levels are similar to those specified in the Basic Data Section of the report, while actual stress levels are intended to provide for an approximate 20 percent growth in output torque.

A 30,000-rpm input shaft speed is considered in this design, but a 15,000-rpm option is possible by changing the input bevel gearing. The use of overrunning clutches capable of operation at engine output speed allows immediate combining of their outputs on a single bull bevel gear which drives the input sun gear of a free-planet planetary reduction system. The free-planet system is unique in that it uses no conventional planet carriers and bearings. The design is illustrated in Figure 53, and the installation arrangement is shown schematically in Figure 54. Other elements of the transmission (main rotor shaft, accessory and tail rotor drives, lubrication system, etc.) are essentially conventional elements conveniently arranged.

The principal characteristics of the transmission system are a weight of 1032 pounds and an estimated efficiency of 97.3 percent.

Free-Planet Reduction Stages

The free-planet transmission concept covers broadly those planetary arrangements wherein the planets are not constrained by being mounted on a carrier or spider. The configuration described offers the potential advantages listed below:

1. Eliminates planet bearing power losses and failures.
2. Has low planetary weight.
3. Permits high reduction in two compound stages at high efficiency.
4. Provides sufficient flexibility and self-centering to give good load distribution between planet pinions.
5. Construction effectively isolates planetary elements from deflections of housing.
6. Inherently favors uniform facewise tooth loading.
7. Potentially greater operating time after loss of lubricant, since planet pinion bearings are eliminated.

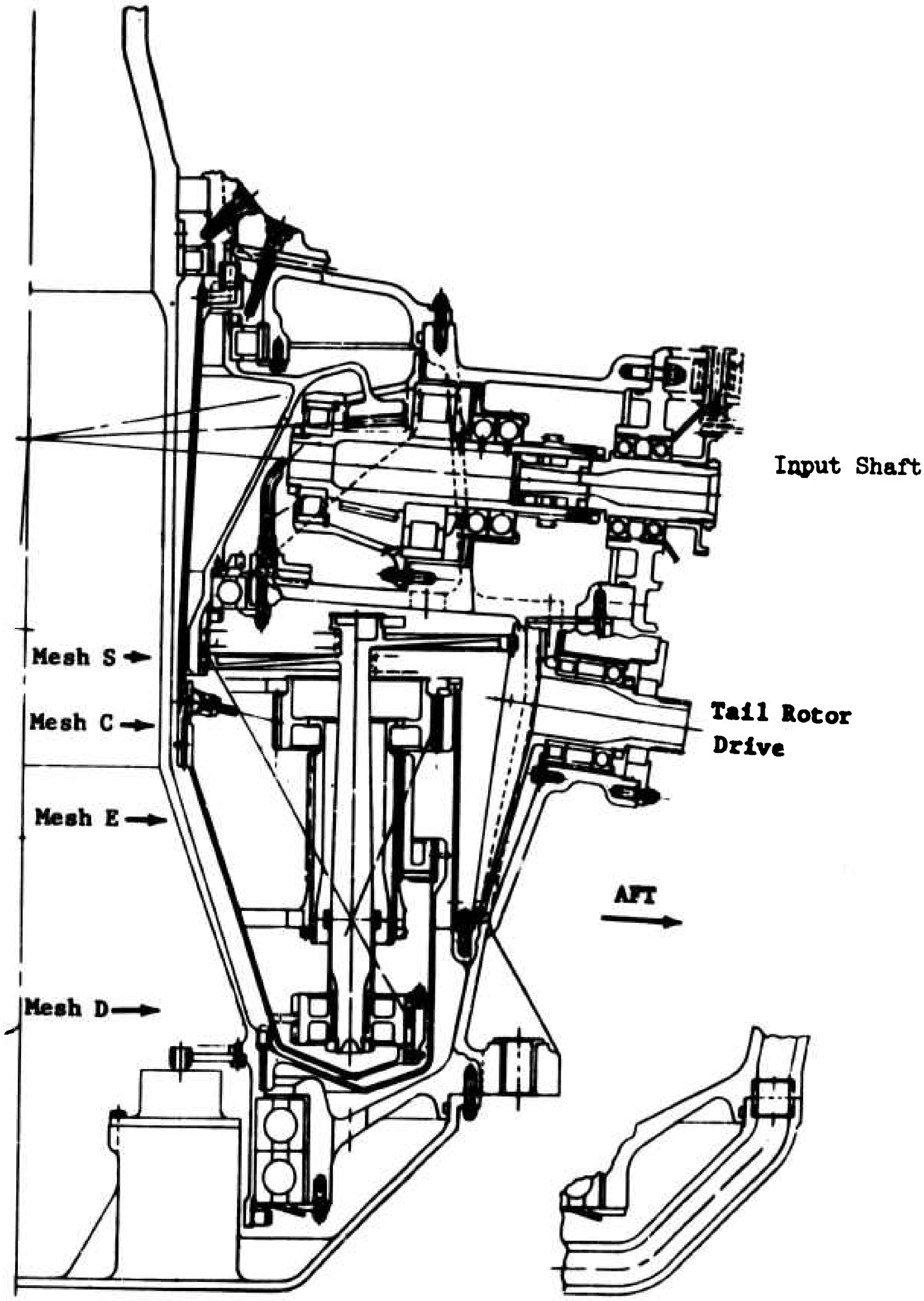

Figure 53. Main Gearbox.

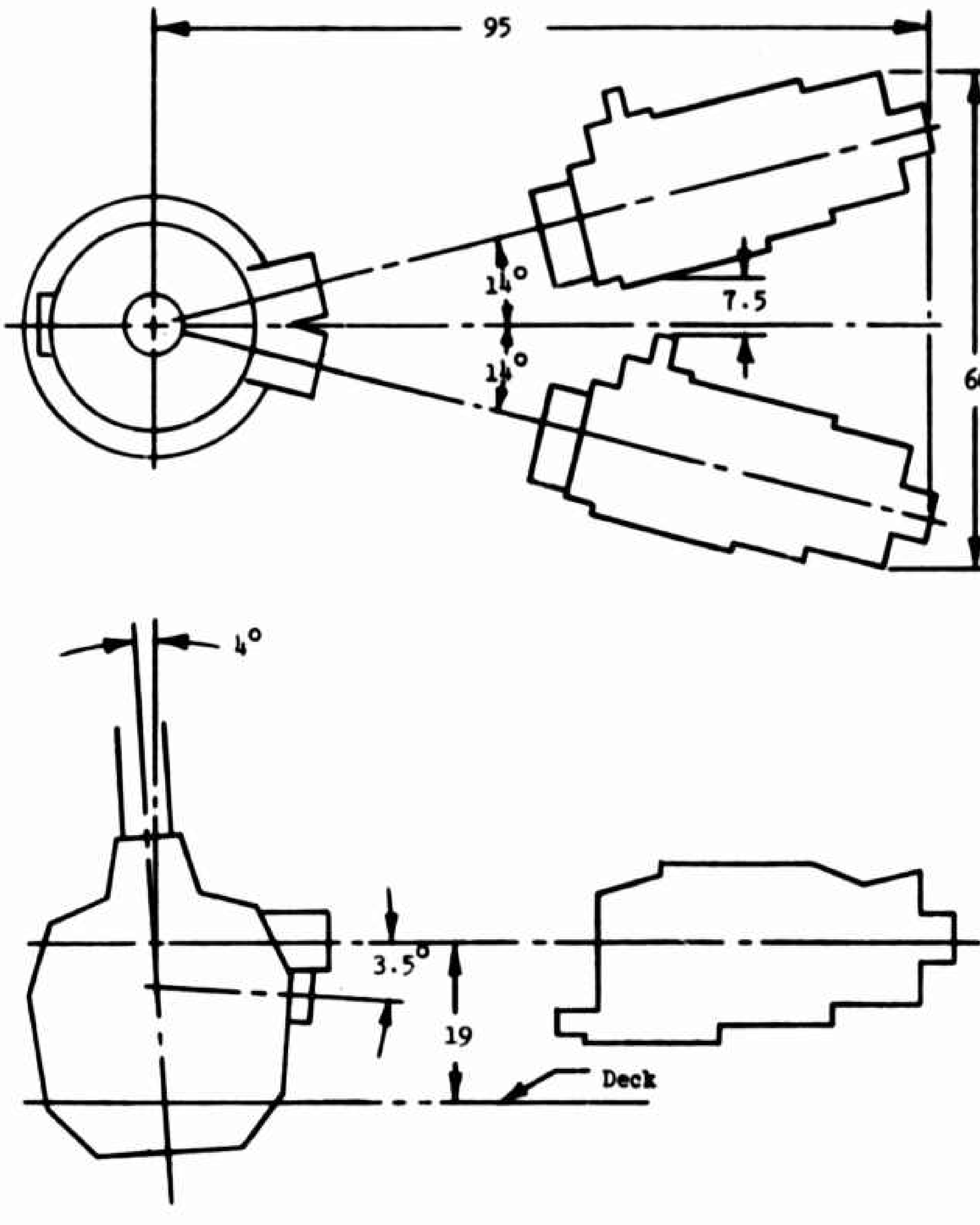

Figure 54. Engine - Gearbox Installation Arrangement.

Input Bevel Gear Drive

The input bevel gear drive, shown in Figure 55, was selected to provide a maximum practicable reduction ratio. The gears are proportioned to be consistent with current production helicopter applications and will maintain proper tooth loading patterns with either or both engines driving.

Symmetrical Power Hinge

This proven actuation concept, shown in Figure 56, is introduced as a background and an introduction to the proposed construction because of its many similarities and its satisfactory history of performance.

The radial plane view of Figure 56 is a partial cross section through the coaxial input/output rotational center cutting one of several planets. The input, S mesh, introduces a tangential force F_S (as shown in the transverse plane view) to the C planet at its inner diameter. The outer mesh of this planet is in engagement with a C mesh ring gear (assumed fixed) which provides a reaction force F_C. The two interconnected E ring gears meshing with appropriate planet gears of a smaller diameter provide the output with a combined tangential force F_E. The relative magnitude of these forces is established by the various gear diameters picked to provide the desired reduction ratio; i.e.,

$$RR = \frac{2\,x\,(1 - x + xz)}{S\,(x - xz)} = \frac{2\,(1 - x + xz)}{S\,(1 - z)}$$

If, as is assumed, these forces are applied symmetrically, as is illustrated in the tangential plane view with the 2 E ring gears, there is no planet skewing tendency and the sum of the planet moments taken about any point in this view is zero.

The radial tooth separating forces at the various meshes and the counteracting reactions of the free floating support rings rolling on cylindrical planet surfaces are illustrated by vertical arrows. Centrifugal forces will relieve the support ring reactions.

Unsymmetrical Free-Planet

The required conditions of equilibrium, easily followed in the symmetrical example shown in Figure 56, can be similarly applied to unsymmetrical arrangements such as the proposed configuration of Figure 53, using the following rationale:

- The application defines the forces and the associated geometry in the transverse plane.
- Appropriate free floating support (or retaining) rings react the loads in the radial plane.
- The axial distances separating the gear face centers in the tangential plane are selected to avoid planet skewing moments (i.e., the sum of moments about any point is zero).

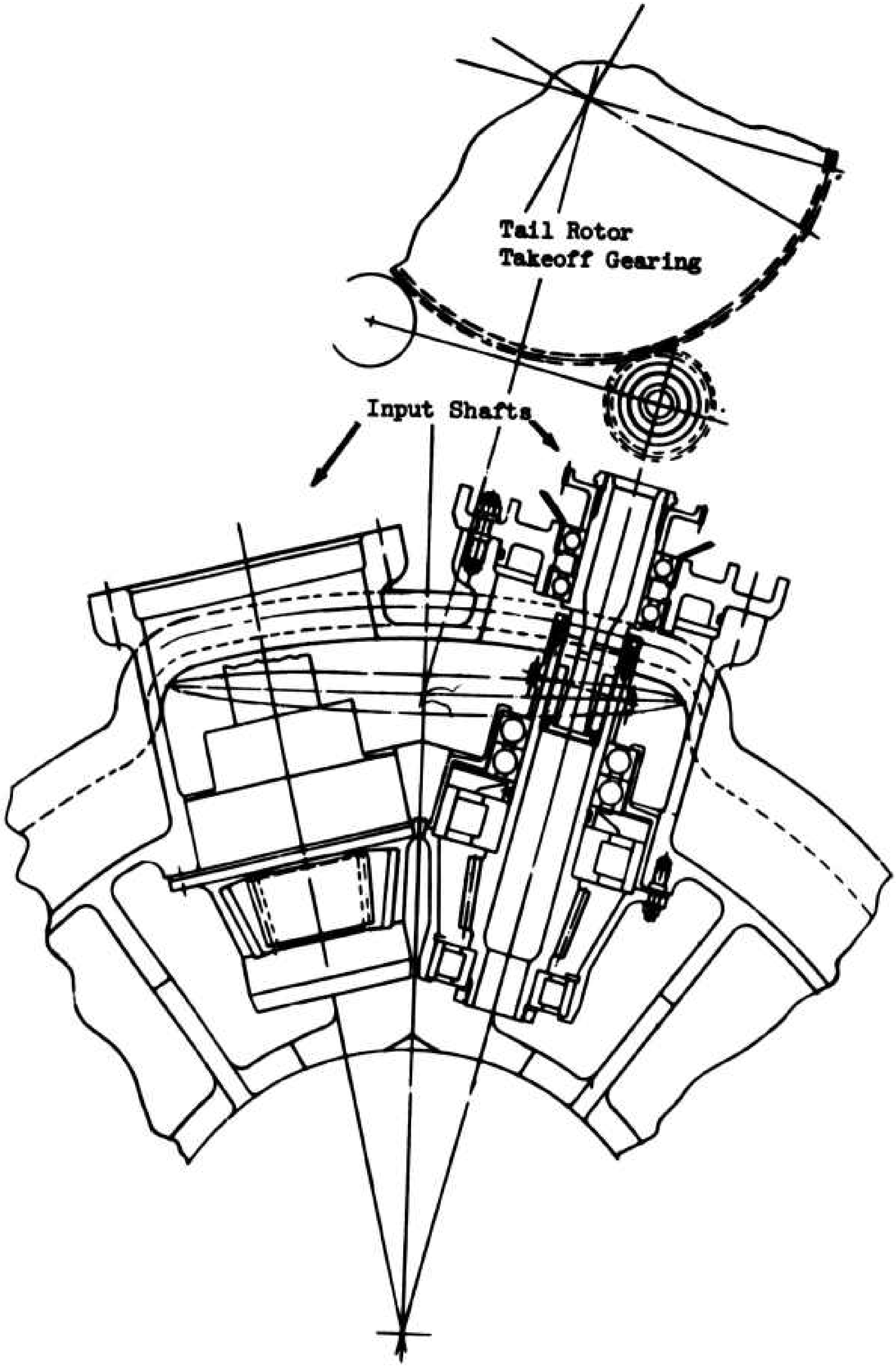

Figure 55. Input Bevel Gear Drive.

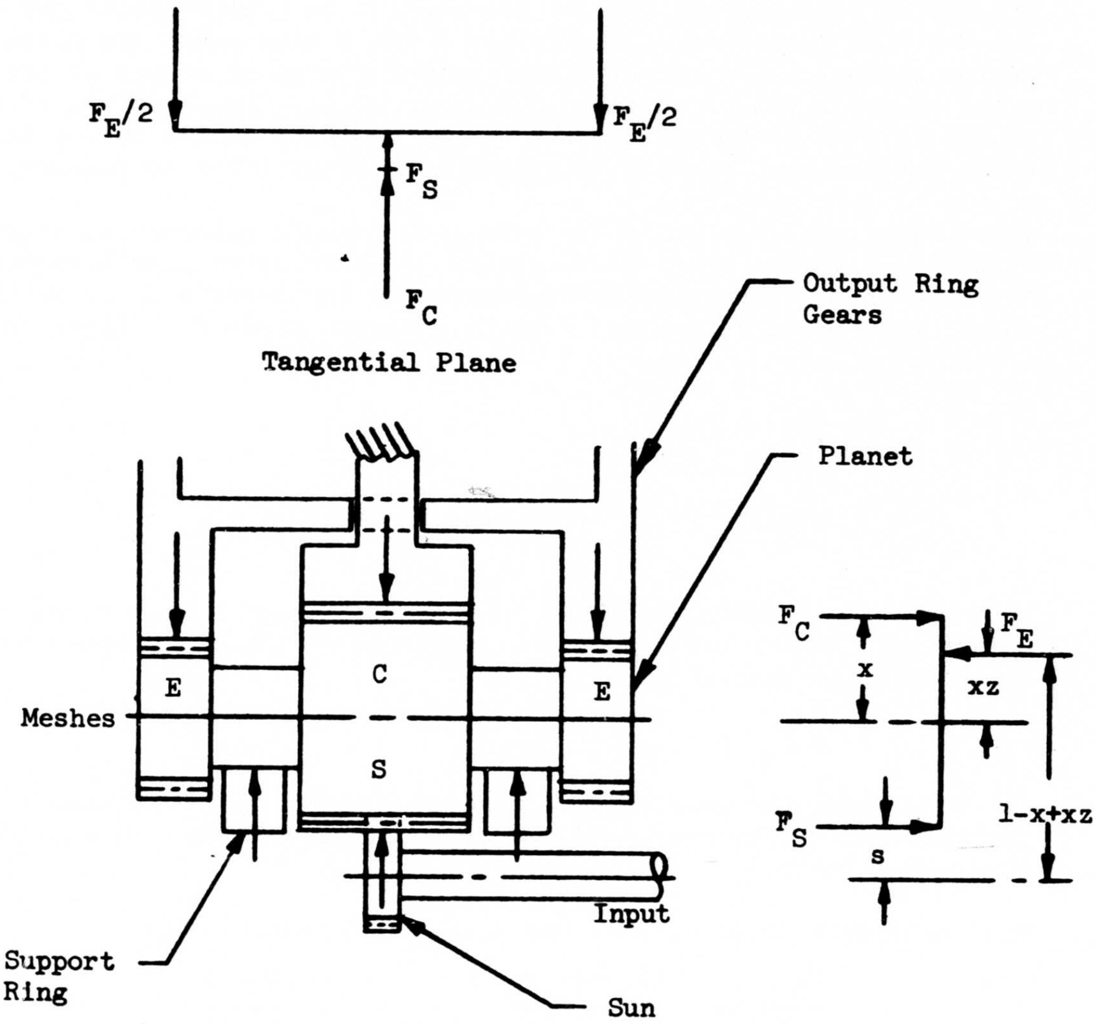

Figure 56. Power Hinge Schematic.

The proposed two-stage free-planet reduction arrangement is shown in cross section in Figure 53; views through various levels are illustrated in Figure 57. The following outline of operating principles is best understood, however, by reference to Figure 58, a schematic in which pertinent parameters are identified. In this schematic representation, reaction to radial forces by suitable rings similar to those in the power hinge of Figure 56 is assumed but not shown.

Output Stage

The output stage planet pinions are mounted on hollow shafts and have C and E mesh gear faces. The mating C and E ring gears are fixed and output members, respectively. The output planet is driven by the input stage by a roller surface located as shown. The location of this roller is such as to balance the forces acting parallel to the tangential plane, and the roller location is established as follows.

Three tangential forces act on each planet shaft assembly as shown in Figure 58. These forces are established by relative gear diameters as shown in the transverse plane where the input force F_R pries the output to produce F_E against a pivot reaction force F_C. These forces are related,

$$F_C + F_R = F_E$$

$$x\,(1 - z)\,F_E = x\,F_R \text{ or } (1 - z)\,F_E = F_R$$

With the relationships of these forces established it but remains to adjust the lengths c and e in the tangential plane to balance the planet against skewing moments.

$$c\,F_R = (c - e)\,F_E$$

(In this plane the gear applied forces, F_C and F_E, are assumed to be concentrated at the center of the appropriate gear face, i.e., midway along its length.)

The relationships of c, e, x and z can be determined by:

$$\frac{F_R}{c\,F_R} = \frac{(1 - z)\,F_E}{(c - e)\,F_E} \quad \text{or} \quad \frac{1}{c} = \frac{1 - z}{c - e}$$

A balance line drawn in the radial view intersecting the points of load application has a slope $\frac{c}{x} = \frac{e}{x\,z}$. Substitution of these values in the above relation results in an identity, proving that such a single line does define the desired balance relationships.

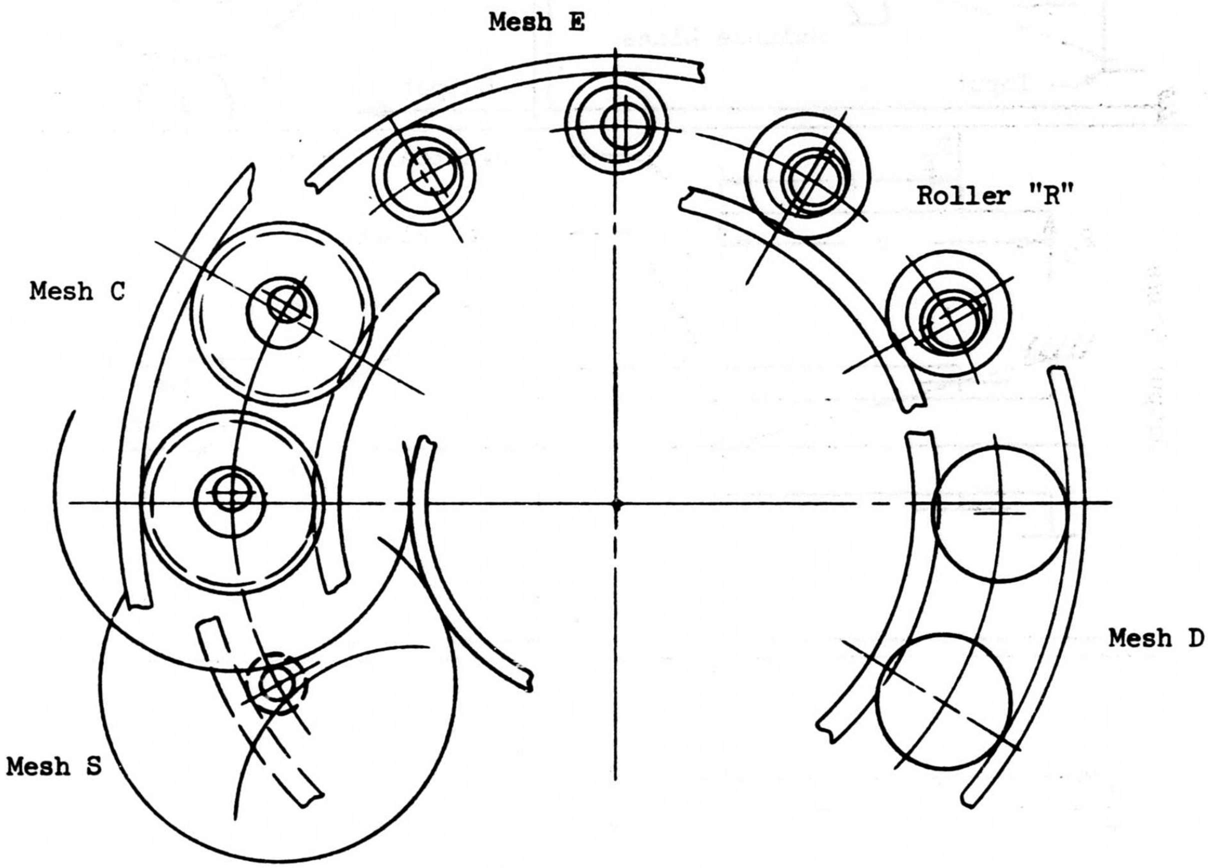

Figure 57. View Through Free-Planet Stages.

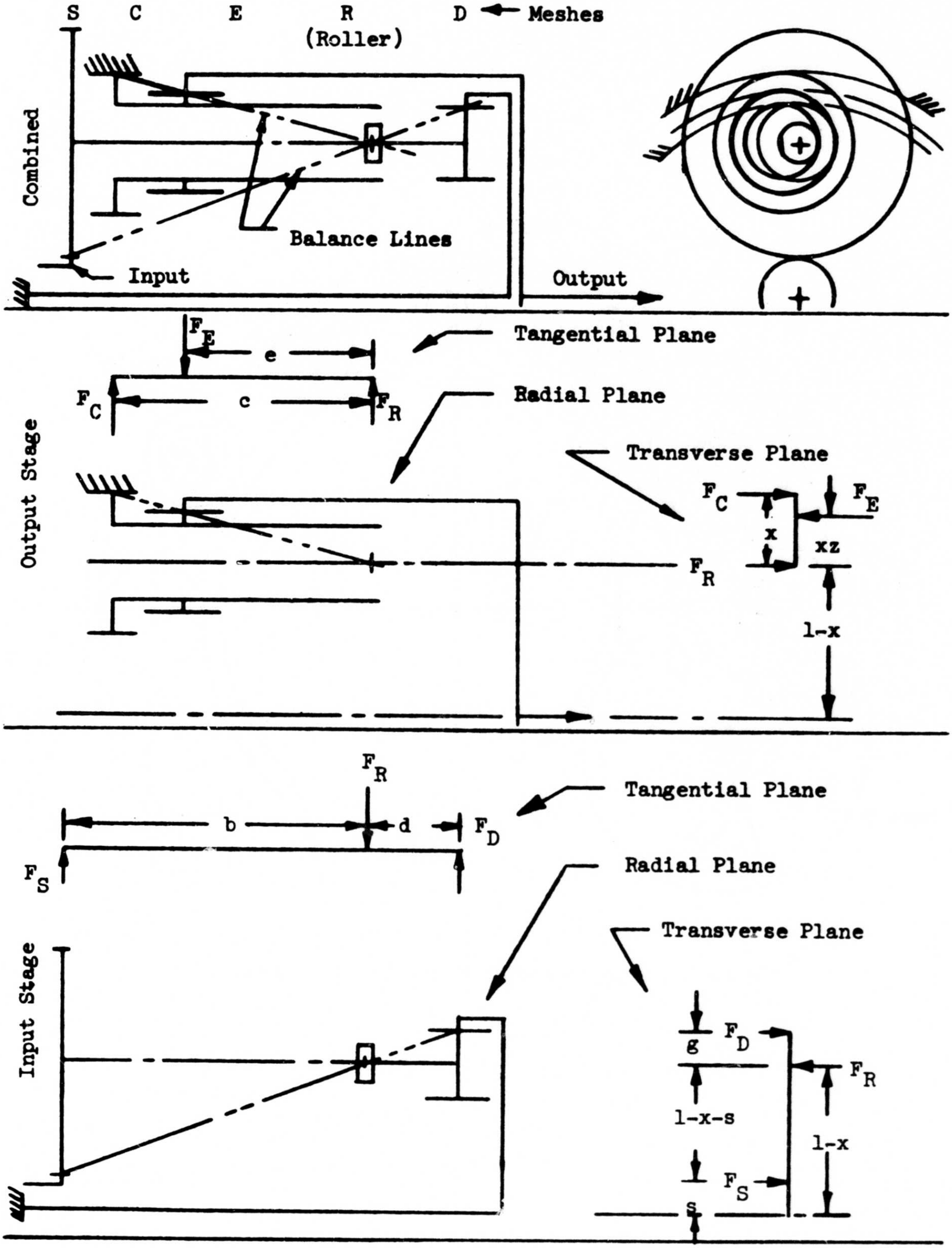

Figure 58. Schematic Diagram of Free-Planet Force Resolution.

Input Stage

The input stage, shown schematically in Figure 58, consists of two gear faces mounted on a shaft which goes through the output stage planet pinion shaft. The S mesh is driven by the sun gear, and the D reaction mesh engages a fixed internal gear. Output is taken from a roller intermediate between the two gear faces located in a plane coincidental with that of the point of application of F_R to the output stage; in addition, the input stage planet is balanced in the tangential plane as follows.

Again, the relationship of the input force, F_S, the reaction force, F_D, and the output force, F_R, is established by the gear diameters selected as illustrated in the transverse plane view of Figure 58.

$$F_S + F_D = F_R$$

$$g\ F_D = (1 - x - s)\ F_S$$

The skewing moments in the tangential view are zero if $b\ F_S = d\ F_D$ and

$$\frac{g\ F_D}{d\ F_D} = \frac{(1 - x - s)\ F_S}{b\ F_S}\ ;\quad \frac{g}{d} = \frac{1 - x - s}{b}$$

Again, the slope of a balance line drawn through the 3 load application points

$$\frac{b}{1 - x - s} = \frac{d}{g}$$

reduces the above equation to an identity and confirms the validity of geometry established by such a balance line.

Facewise Tooth Loading

The inherent tendency of a free planet to adjust its axis to produce uniform facewise tooth loading is illustrated in Figure 59, which refers to the output stage. (Similar reasoning applies to the input stage.) In (a), the assumed uniform planet tooth loading is identified by a series of short arrows, while the resultant of the sum of these forces is shown for each tooth by a long arrow at the center of each tooth. As explained previously, the axial tooth spacings as defined by e and c have been selected so that the net of moments in the tangential plane are zero; with the forces, as controlled by the proportions defined in the transverse plane view to give the desired reduction, etc., planet is in equilibrium. If the planet axis is skewed as shown in (b) or (c), the resulting end loading on the teeth changes e and c so that an unbalanced moment exists which tends to

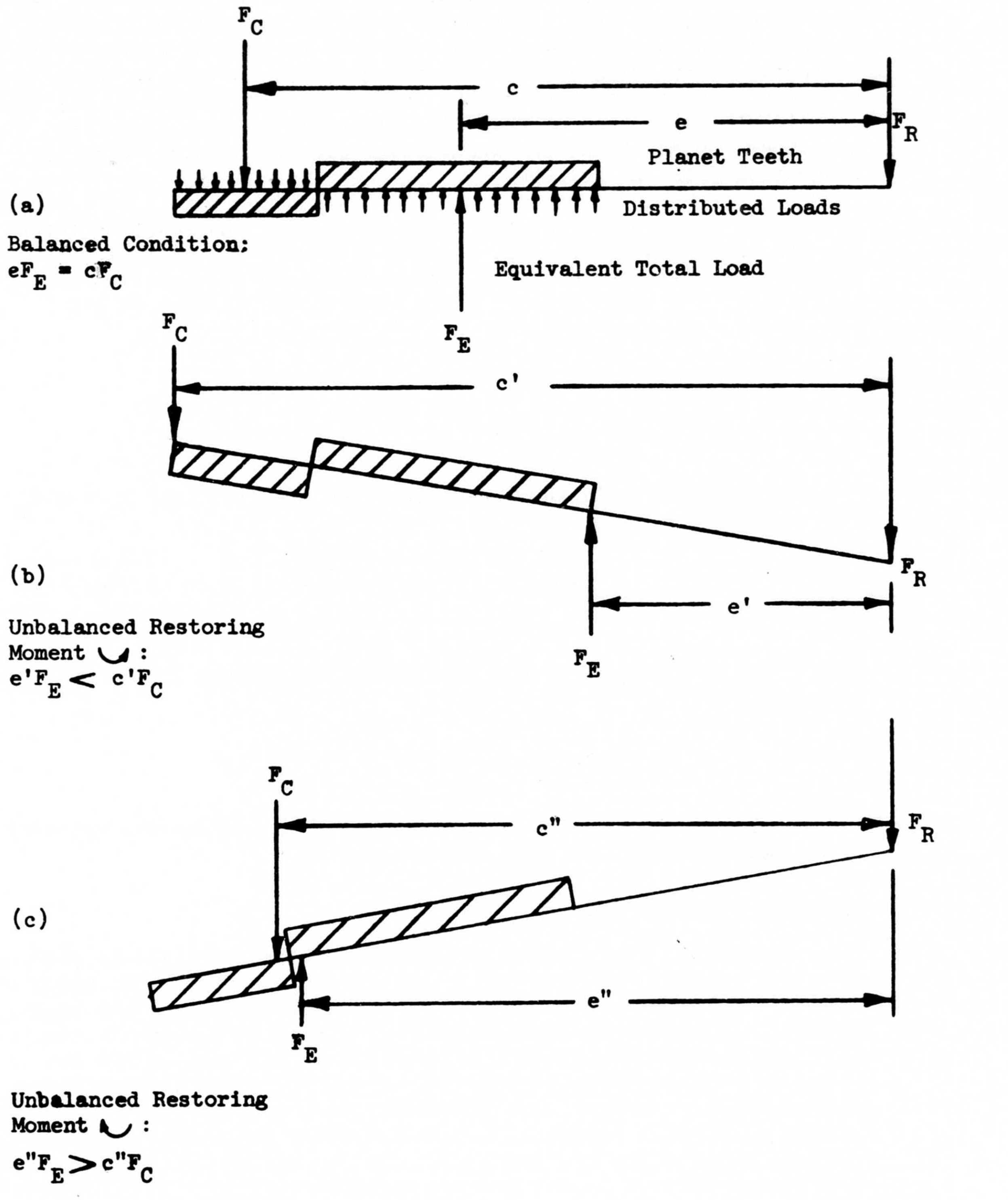

Figure 59. Schematic Diagram of Facewise Load Distribution.

restore the planet axis to the condition of (a) where tooth loads are evenly distributed.

There are several implications of this phenomenon. Not only does it eliminate the necessity of moment and load absorbing planet carrier bearings, but it suggests that customary gear length to diameter limitations may be challenged without consequent corner loading.

Upon further consideration, it is apparent that to achieve the uniform distribution desired, it is necessary to consider the effect of deformations under load, as they may influence the effective parallelism of the teeth in contact.

A preliminary analysis of major deflections in the output stage in which facewise tooth load distribution is most critical reveals that the proportions and arrangement of support points selected result in essentially no mismatching of the meshes involved.

Planet-to-Planet Load Distribution

In common with other planetary gear systems, consideration must be given to the division of load between planet pinions as influenced by manufacturing tolerances. In this instance these are composed of eccentricity discrepancies and a steady indexing error and cyclic variations in gear tooth location.

The effect of these factors on load distribution is a function of their magnitude and the stiffness of the planet-to-planet load carrying path. With the mechanism proposed it is possible, by appropriate inspection type fixturing, to keep the steady indexing error small without imposing strict manufacturing requirements if eccentricity limits associated with AGMA Class 11 gearing are maintained.

Analysis indicates that the maximum overload on any part of any planet will be in the order of 10 percent. This is of the same order as the mal-distribution normally attributed to conventional planetaries with many (more than three) planets, although no quantitative experimental data are available concerning load sharing in planetary gear systems.

Interstage Roller (Figures 53, 57, and 58)

The force F_R is transmitted to a cylindrical internal diameter in the output stage by means of a roller on the input stage planet shaft. The diameter relationships are such that only pure rolling is involved, this being the case when

$$\frac{\text{Roller O.D.}}{\text{Shaft I.D.}} = \frac{g}{(1 - x - g)\, x}$$

The actual roller diameter is selected to allow interposition of a nonrotating self-aligning sleeve bearing between the input planet shaft and the roller. The input shaft diameter is established by its structural requirements, and the roller width is established to maintain conservative surface contact stresses.

Retaining/Support Rings

Radial planet forces are absorbed and balanced out by rings which engage cylindrical surfaces on the planets. Since all planet diameters engaging a given ring are equal, the conditions approach that required for pure rolling motion.

The radial forces involved are the net of tooth separating forces and centrifugal forces. In the case of the output planet the tooth separating forces and lesser centrifugal forces are opposed as they are at the D mesh end of the input planets. Since the tooth separating forces predominate, support rings are used with the planets rolling on their O.D.'s. At the sun input end of the input stage, the tooth separating and centrifugal forces are additive and a retaining ring is employed with the planets rolling on its inside diameter.

All rings are sized with sufficient width to maintain acceptable contact stresses and with cross-sectional areas and section moduli to limit deflections and bending stresses to acceptable values.

Tail Rotor Drive

Power for the tail rotor is taken directly from a bevel pinion on one of the input bevel gears, as shown in Figure 53 and the upper view of Figure 55. This bevel pinion meshes with a gear of the proper size and orientation for the tail rotor drive shaft installation.

Accessory Drive

The power output to the accessory gearbox is shown in Figure 60. A spur gear mounted concentric with and driven by the planetary sun gear assembly meshes with a smaller gear to provide the desired speed increase. The output drive to the accessory gearbox is achieved by the use of a miter gear set.

Main Rotor Shaft

The main rotor shaft is concentric with and driven by the E mesh ring gear through a splined connection as shown in Figure 53. Advantage is taken of the relatively large central space allowed by the type of gearing employed to make the main rotor shaft and its supporting bearings relatively large in diameter in an effort to minimize weight and, if required, conserve the interior space for an oil cooler and blower. Retention of the main rotor shaft is provided by a combination of tandem ball bearings at the lower end to take lift and radial forces and a cylindrical roller at the upper end for radial forces.

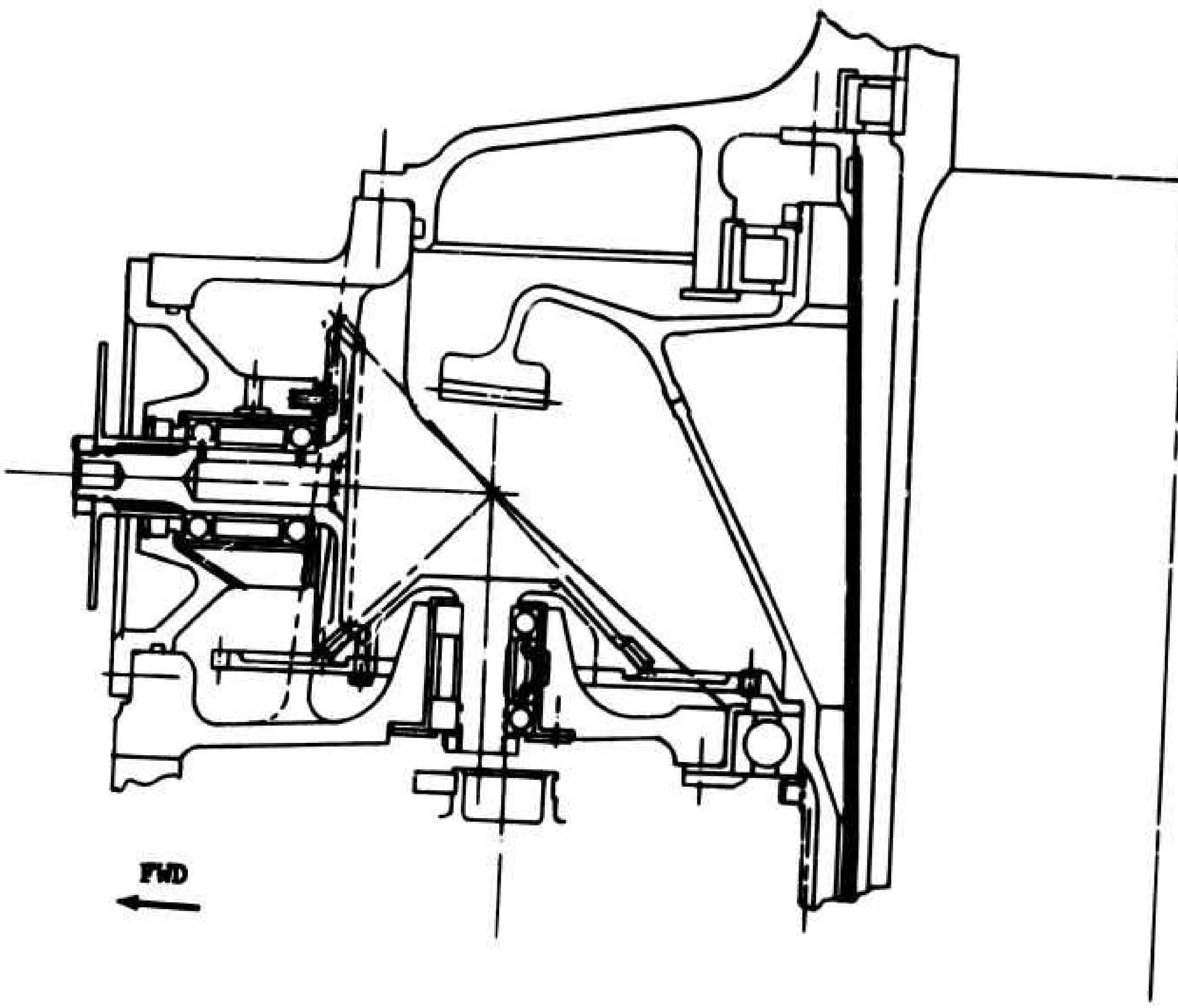

Figure 60. Accessory Drive Arrangement.

Selection of Free-Planet Proportions

The proportions selected for the gear ratios in the two stages of Figure 53 are a result of the following considerations:

1. The output stage is a differentially compound planetary in which the amount of recirculating power is related to the ratio. Both the C and E meshes each transmit an amount of power equal to the stage ratio minus 1 times the useful power, i.e., $RPR_2 = (R_2 - 1)$. The input stage, however, is a normal compound pinion stage where the amount of power transmitted by each mesh (the S and D meshes) is equal to 1 minus the reciprocal of the ratio of this stage, i.e., $RPR_1 = 1 - 1/R_1$. Consequently, for a given overall reduction ratio it is desirable to minimize the output stage ratio and maximize the input stage ratio if a minimum loss transmission is desired for highest overall efficiency.

   In order to get a maximum reduction in the input stage, the planet gear meshing with the sun gear is made as large as possible. Were it necessary to keep all sun planet gears in the same plane, they could be no larger than the C mesh output stage planet pinion. However, the relatively narrow sun planet face width makes it possible to stagger alternate planet pinions and overlap them to the point where they just clear the shafts to which the adjacent pinions are attached. The resulting small shift in the location of the balance line can be compensated for by a minor correction to the D mesh with little sacrifice in weight or space.

2. The extent to which the output stage ratio can be minimized is limited by two factors. A low reduction in this stage is associated with a small diameter E mesh planet. As this gear becomes smaller, the output force, F_R, from the input stage is increased. This gear size is limited by the allowable stresses in the diameter of the first-stage pinion shaft which must clear the inside diameter of the E mesh planet by an amount dictated by the relative diameters of the F_R transmitting roller and the shaft inside diameter in which it rolls. The difference in these diameters also tends to increase as the input stage ratio increases.

Free-Planet Reduction Ratio Equations

Referring to the transverse views of Figure 58, the following reduction ratio equations can be derived:

$$RR_2 = RR_{\text{Planet Centerline to Output}} = \frac{1 - x + xz}{(1 - x)(1 - z)}$$

$$RR_1 = RR_{\text{Input to Planet Centerline}} = \frac{(1 - x - s + g)(1 - x)}{sg}$$

$$RR_1(RR_2) = RR_{\text{Overall}} = \frac{(1 - x - s + g)(1 - x + x z)}{s g (1 - z)}$$

BEARING AND GEAR DATA

Each engine drives a bevel pinion meshing with a combining bevel gear with a ratio of 7.9:1. This driven bevel gear drives the sun gear of the 12-planet fre -planet reduction. The ratio to the planet centerline of the input stage where its output is taken is 6.7:1, while the output stage ratio from this point to the movable E ring gear is 2.5:1 for an overall ratio of 134:1. Bearing data are contained in Table XLIV, while gear data are summarized in Table XLV.

TABLE XLIV
BEARING DATA

Bearing Location (Ref. Fig. 53)	Type of Bearing **	Size of Bearing (mm)	Speed (rpm)	B_{10} Life Unfactored (hours)
Bevel Pinion-Toe	RB	40 x 90 x 23	30,000	4,000
Bevel Pinion-Heel	RB	60 x 130 x 31	30,000	3,000
Bevel Pinion Thrust	BB	50 x 90 x 20	30,000	4,600*
Bevel Gear	RB	254 x 346 x 25	3,800	2,900
Bevel Gear	BB	282 x 334 x 24	3,800	3,200
Main Rotor Shaft	RB	222 x 270 x 21	224	10,000
Main Rotor Shaft	Duplex	362 x 440 x 39	224	3,000
Tail Rotor Drive	RB	50 x 80 x 16	6,000	10,000
Tail Rotor Drive	BB	50 x 80 x 16	6,000	7,000

*B_{10} life x 3 using M-50 steel

**RB = Roller Bearing, BB = Ball Bearing

EFFICIENCY SUMMARY

Main Rotor Drive-3600 hp Output

	Loss, hp
Frictional Losses	48.6
Churning Losses	54.7
Main Rotor Drive Losses	103.3

TABLE XLV
GEAR DATA

Gear	Number of Teeth	Pitch Dia. (in.)	Bending Stress (psi)	Hertz Stress (psi)	Pitch Line Velocity (fpm)	Face Width (in.)	Misc
Input Bevel Pinion	25	2.86	28,100	209,000	22,500	1.70	Pr.Angle = 25° Sp.Angle = 20° Sh.Angle = 96°
Input Bevel	198	22.65	11,500	209,000			
Sun	108	10.88	25,000	101,000	9,200	0.50	
Sun Planet	91	9.22	25,000	101,000	9,200	0.50	
D Fixed Ring	228	23.62	20,900	130,000	3,400	1.25	
D Planet	34	3.52	27,400	130,000	3,400	0.95	
C Fixed Ring	250	25.00	27,700	106,000	3,600	1.61	
C Planet	49	4.90	27,700	106,000	3,600	1.61	
E Output Ring	246	22.78	27,700	140,000	3,300	3.42	
E Planet	29	2.68	27,700	140,000	3,300	3.42	
Tail Rotor Drive Pinion	25	2.51	18,100	125,000	19,600	0.500	Sh.Angle = 14½°
Tail Rotor Drive	125	12.5	23,300	125,000	19,600	0.375	

Tail Rotor Drive - 320 hp Output

Gear Mesh	Loss, pct	Loss, hp
2 bevel	1.0	3.2
churning	0.75	2.4
	Tail Rotor Drive Losses	5.6

Accessory Drive - 123 hp

Gear Mesh	Loss, pct	Loss, hp
2 bevel	1.0	1.23
2 spur	1.0	1.23
churning	0.75	0.92
	Accessory Drive Losses	3.38

Input Power Required

Main Rotor Drive	3600 + 103.3 = 3703.3
Tail Rotor Drive	320 + 5.6 = 325.6
Accessory Drive	123 + 3.4 = 126.4
Total Power Required	4043 + 112.3 = 4155.3

$$\text{Transmission Efficiency} = \left(1 - \frac{113.3}{4155.3}\right) 100 = 97.3 \text{ percent}$$

WEIGHT SUMMARY

A detailed weight analysis was conducted on the parts which are utilized in the proposed design. The weights of lubrication pumps, filters and miscellaneous items were only estimated. A summary of the weights is given in Table XLVI.

TABLE XLVI
WEIGHT SUMMARY,
MAIN GEARBOX WITH FREE-PLANET PLANETARY

Item		Weight - lb
Magnesium Housings		190
Bevel Drive Splined Adapters, Overrunning Clutches, Pinions, Sun-Bevel Gear, Steel Housings, Bearings, Liners, Clamp Nuts, Gimbal Mount, etc.		160
Planetary Elements		
Input Planets	107	
Output Planets	92	
Ring Gears	69	
Rings	12	
		280
Reaction and Drive Tubes to Planetary Ring Gears		99
Rotor Shaft		166
Rotor Shaft Support Bearings, Seals, Liners, Clamps and Nuts		69
Tail Rotor and Accessory Drive Elements Gears, Bearings, Seals, Liners		18
Lubrication Pumps, Filter, Jets and Fittings		30
Miscellaneous Hardware		20
Total Dry Weight		1032

LIST OF SYMBOLS

b Axial distance between centers of PS mesh and roller, in.

c Axial distance between roller and C gear centers, in.

C* Output planetary stage gear mesh to ground

d Axial distance between roller and D gear centers, in.

d Interstage roller diameter, in.

D* Input planetary stage gear mesh to ground

e Axial distance between roller and E gear centers, in.

E* Output planetary stage mesh to output

E Modulus of elasticity

F Force, lb

g Ratio $\frac{PDPD}{PDRC} = \frac{\text{Pitch Diameter, D Mesh Planet}}{\text{Pitch Diameter, C Mesh Ring Gear}}$

P Planet gear

R Ring gear

s Ratio $\frac{PDS}{PDRC} = \frac{\text{Pitch Diameter, Sun Gear}}{\text{Pitch Diameter, C Mesh Ring Gear}}$

S* Input planetary stage sun gear

x Ratio $\frac{PDPC}{PDRC} = \frac{\text{Pitch Diameter, C Mesh Planet}}{\text{Pitch Daimeter, C Mesh Ring Gear}}$

z Ratio $\frac{PDPE}{PDPC} = \frac{\text{Pitch Diameter, E Mesh Planet}}{\text{Pitch Diameter, C Mesh Planet}}$

PD Pitch diameter, in.

RPR Recirculating power ratio

RR Reduction ratio

Subscripts

1 Input stage or component

2 Output stage or component

R Roller

*Also used as subscripts to identify meshes

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13. ABSTRACT

This report presents transmission system concepts that have been developed during a design investigation into single-rotor pure and compound helicopter drive trains.

Advanced technology gas turbine engines which utilize high output shaft speeds require greater speed reduction in helicopter drive trains. This study has considered the effects of these engines upon various drive train configurations. Accepting turbine power at free turbine speed and incorporating all speed reductions in the main gearbox have permitted consideration of several new reduction concepts that offer increased redundancy and efficiency over more conventional gearbox designs.

Specific areas considered in this study include torque dividing mechanisms supplying power to multiple load paths, redundant drives, and the use of epicyclic gearing for initial speed reduction.

This study indicates that a selected pure helicopter transmission system, with 4000 horsepower input, can be fabricated for approximately 1600 pounds with an efficiency of 97.4 percent. A compound helicopter transmission system exhibits a minimum efficiency of 96.7 percent with a resulting system weight of 1839 pounds.

Recently, the free-planet planetary concept for parallel-axis speed reduction has been advanced for use in helicopter drive trains. A preliminary study incorporating this concept in a main gearbox is included.

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14. KEY WORDS	LINK A		LINK B		LINK C	
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Torque dividing mechanism						
Redundant drives						
Epicyclic gearing						
Free-planet planetary						