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USAAMRDL-TR- 75-56B

ADVANCED HELICOPTER STRUCTURAL DESIGN INVESTIGATION
Volume II - Design Application Study for Free Planet Transmissions

Boeing Vertol Company
P.O. Box 16858
Philadelphia, Pa. 19142

March 1976

Approved for public release;
distribution unlimited.

Prepared for

EUSTIS DIRECTORATE
U. S. ARMY AIR MOBILITY RESEARCH AND DEVELOPMENT LABORATORY
Fort Eustis, Va. 23604

EUSTIS DIRECTORATE POSITION STATEMENT

This effort is one of two parallel contractual studies to define advanced structural configurations, advanced materials, and fabrication technology to satisfy requirements for a complete helicopter. The associated study program under the same title was conducted by Sikorsky Aircraft under the terms of Contract DAAJ02-74-C-0061.

Numerous design concepts, materials, and manufacturing techniques were investigated for various helicopter components (i.e., body group, main rotor, transmission, etc.). The best overall concepts were selected and integrated into a complete advanced helicopter design, with predictions of improved weight, cost, and aircraft performance.

Mr. L. Thomas Mazza, Technology Applications Division, served as project engineer, with Mr. E. Rouzee Givens directing the "Free Planetary Transmission Drive" study portion of the program.

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

The free planet gear drive was applied to the requirements of the Medium-Range Utility Transport Helicopter. Design studies showed potential advantages for the free planet drive as compared to conventional planetary systems used hitherto. Recommendations were made that existing free planet hardware be further tested to better define load-carrying capability and system reliability.

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SUMMARY

The analysis shown in Table 1 concludes that the free planet transmission as applied to the Medium-Range Utility Transport Helicopter (MUT) is equal or superior to a conventional transmission in all but cost and producibility. The Boeing Vertol advanced concept transmission showed significant advantage in this area, as well as in others.

The assessment of the free planet transmission is drawn from design studies and analyses shown in this report as well as from limited testing of a 500-hp unit performed by Curtiss-Wright Corp. Features of the advanced concept transmission which are pertinent to the evaluation are outlined in this report.

Further testing of existing free planet hardware is recommended. There is a potential for increased load-carrying capacity because of better load sharing between planets and for increased reliability because planet bearings are eliminated. These features should be evaluated before final conclusions are drawn.

Table 1. Concept Screening Analysis – Main Rotor Transmission

- OBJECTIVE: Select best transmission concept for MUT
- COSTS: Based on 1,000 units

		BASELINE	CONCEPT A	CONCEPT B
Rating factors will be comparative to baseline structure counterpart 2 = MUCH BETTER 1 = BETTER 0 = SAME −1 = POORER −2 = MUCH POORER CONFORMANCE FACTORS: BETTER (+) } REFERRED SAME (0) } TO POORER (−) } BASELINE		CONVENTIONAL TRANSMISSION	BOEING VERTOL ADVANCED CONCEPT TRANSMISSION • Stiff fail-safe housing • Reduced weight • Reduced cost • Reduced number of bearings • Close coupled hub • Structural monitoring system • Load-balanced planet carrier	CURTIS-WRIGHT FREE PLANETARY CONCEPT TRANSMISSION USAAMRDL Tech Report 74-27 defines general configuration
RATING ELEMENTS	STRUCTURAL EFFICIENCY	0	2	1
	FAIL SAFETY	0	0+	0+
	SAFETY	0	0+	0+
	PRODUCIBILITY/COST	0	1+	0–
CONFORMANCE ELEMENTS	DETECTION AVOIDANCE	0	0	0
	CRASHWORTHINESS	0	+	0
	REPAIRABILITY	0	0	+
	MAINTAINABILITY	0	+	+
	SURVIVABILITY	0	+	+
	GENERAL SAS CONFORMANCE	0	0	0
A/F CONCEPT RANKING		(3)	(1)	(2)

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PREFACE

This document is Volume II of the final report on the results of a preliminary design exercise entitled Advanced Helicopter Structural Design Investigation; Volume I is USAAMRDL Technical Report 75-56A, Investigation of Advanced Structural Component Design Concepts. The program was conducted by the Boeing Vertol Company for the Eustis Directorate, U. S. Army Air Mobility Research and Development Laboratory, under Contract DAAJ02-74-C-0066, from June 1974 through May 1975.

The work includes definition of a state-of-the-art aluminum baseline medium-range utility helicopter, redesign in advanced composites with advanced structural subsystems, and resizing of the advanced helicopter to perform the identical mission of the baseline helicopter.

Technical direction was provided by Mr. L. Thomas Mazza, with the free planetary transmission drive study directed by Mr. E. Rouzee Givens, both of the Eustis Directorate, USAAMRDL.

The study was conducted at the Boeing Vertol facility in Ridley Park, a suburb of Philadelphia, Pennsylvania. The principal Boeing contributors were Donald Hoffstedt, Program Manager; Sidney Swatton, Airframe Design; John Mack and William Rumberger, Transmission Design; Erwin Durchlaub, Structural Analysis; Frank Sauter, Cost Engineering; Arling Schmidt, Weights Analysis; Robert Pinckney, Manufacturing Technology; David Harding, R&M, Survivability/Vulnerability; John Schneider, Preliminary Design; and Harold Rosenstein, Performance and Sizing.

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Drive system structural efficiency was addressed principally through a mechanical design study of the main transmission; the concepts and ratings are shown in Table 1.

USAAMRDL authorized a separate study of a Curtiss-Wright free planetary drive transmission, sized for the MUT configuration, as part of this contract. This design study and the conclusions drawn are described in the section entitled FREE PLANET DRIVE SYSTEM.

FREE PLANET DRIVE SYSTEM

This portion of the design study effort was conducted to evaluate the potential advantages of applying the Curtiss-Wright free planet arrangement of gears to the MUT aircraft.

Specific free planet candidates were selected after considering various arrangements in a parametric study. Factors which determined the selection are discussed. Two free planet transmission configurations are compared against conventional planetary systems in the following areas:

1. Weight and efficiency
2. Cost and producibility
3. Configuration compatibility
4. Reliability and maintainability
5. Survivability and vulnerability

THE FREE PLANET DESIGN CONCEPT

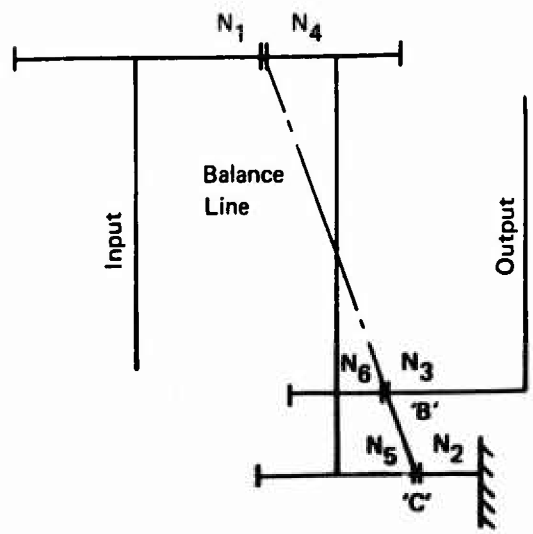
The free planet concept covers those arrangements of planetary gears wherein the planets are not constrained by rigid positioning on a carrier or "spider" structure. In the free planet concept the planetary gears are spaced on planetary spindles. The radial forces are reacted by support rings which react the gear separating forces and centrifugal forces of planet progression. The gear torque forces on the spindles are balanced in such a way that the spindles are teetered about the output gear reaction force; therefore, the spindle is free to center itself within the force system imposed on it without the restraints of the conventional planetary carrier and bearing.

Curtiss-Wright has designed and tested free planet transmission hardware in the 500-hp range for ratios of 7:1 and 20:1. (The 20:1 hardware identified as the FP 500 and FP 501 was designed and developed under contract USAAMRDL TR-74-27.) This hardware was visually inspected by a Boeing Vertol design engineer. Figure 1 shows the Curtiss-Wright hardware and schematic arrangement.

Curtiss-Wright provided consultation on the free planet concept to assist in the design effort at Boeing Vertol. The tooth numbers pertaining to the timing requirement that appear on the two free planet layouts were supplied by Curtiss-Wright.

SELECTION OF COMPARATIVE CANDIDATES

The process of selecting drive system configurations was guided by the following considerations:



Free Planet Schematic

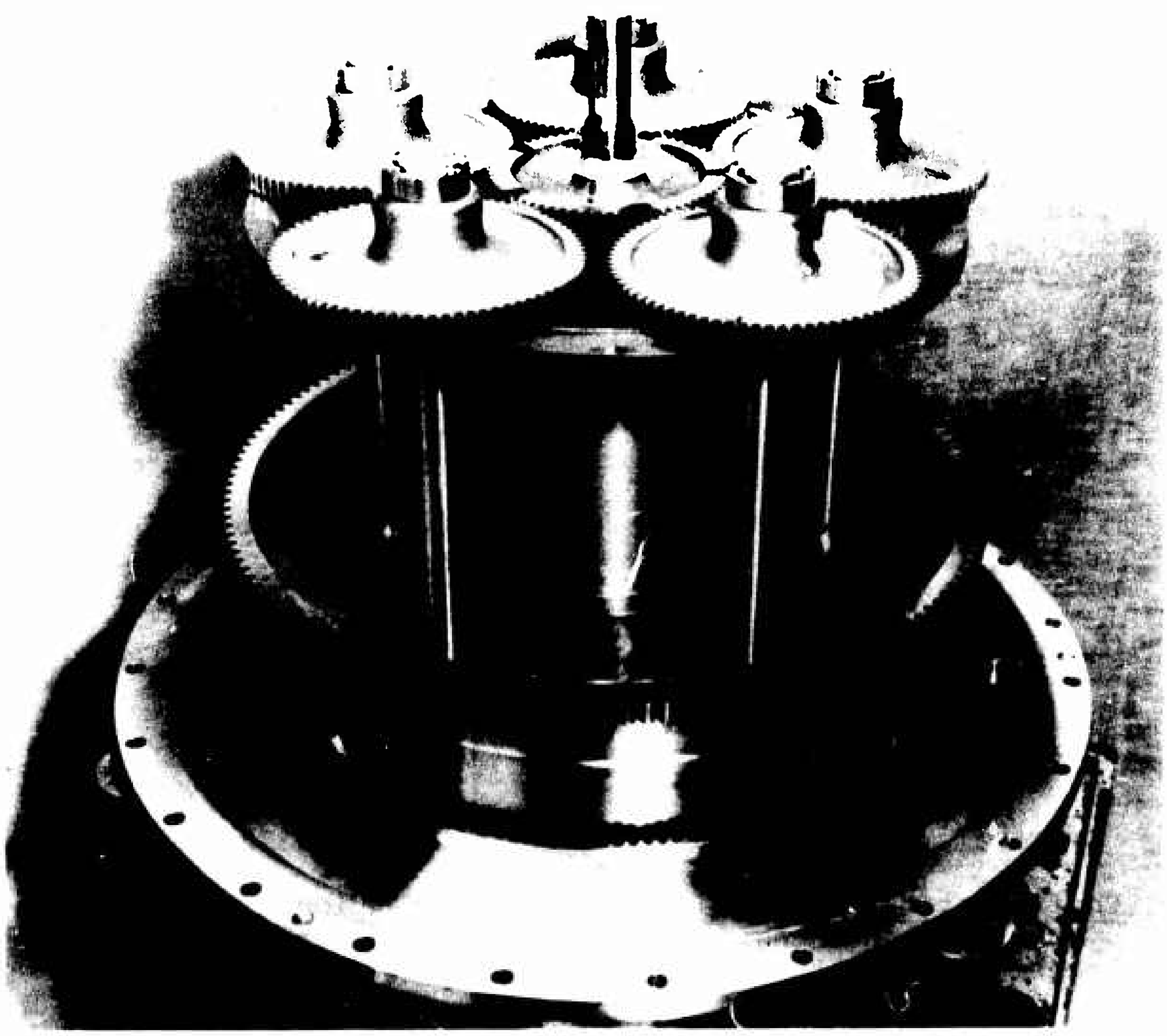


Figure 1. FP500 Series Free Planet Assembly (Curtiss-Wright).

1. Configurations embodying the free planet and the conventional planetary drive system arrangement shall be comparable in ratio and other major features.

2. The selected free planet arrangements must be configured to apply to the current MUT airframe structure described in this document.

3. The candidate free planet arrangement should be optimized for the least weight and cost while conforming to the restraints of the MUT application.

4. The same loads, stress levels, and materials criteria shall apply to both the free planet transmission and its conventional counterpart.

5. Designs shall be in accordance with Boeing Vertol design experience, and may incorporate design improvements derived from Boeing Vertol experience.

The Candidates

- Configuration I (Figure 2) is the conventional baseline transmission. The essential ratios are:

Engine box	2.7:1 (30,000 rpm in)
Main transmission spiral bevel	5.4:1
Planetary	5.5:1
Total rotor box	29.7:1 (372 rpm out)

- Configuration II (Figure 3) is an alternative to configuration I; it uses the free planetary system. The essential ratios are:

Engine box	2.0:1 (30,000 rpm in)
Main transmission spiral bevel	3.24:1
Planetary	12.9
Total rotor box	41.8 (372 rpm out)

- Configuration III (Figure 4) is a conventional transmission system which has engines driving directly into the rotor transmission. The essential ratios are:

Main transmission spiral bevel	4.69:1
Planetary	17.2:1
Total rotor box	80.7:1

- Configuration IV (Figure 5) uses a free planetary arrangement and has engines driving directly into the rotor transmission. The essential ratios are the same as those of configuration III.

Justification for Candidate Selection

A basis for a meaningful comparison of the free planet system is to compare it to the best choice in current drive system technology.

JUSTIFICATION OF ARRANGEMENT AND RATIO SELECTION

Figures 2 and 3 are the conventional baseline and free planet arrangements of a drive system which employs engine transmissions in order to improve the configuration by simplifying the structure and air intakes, and minimizing frontal area. The rationale for the particular ratio splits are as follows:

Engine Box

The engine boxes for configurations I and II are a scaled down Boeing Vertol configuration. Figure 6 is a 2.7:1 reduction box containing an overrunning clutch on the output side. Configuration I would have a ratio of 2.7:1 with a single engine rating of 1033 hp.

The engine box for configuration II (the free planet arrangement, Figure 3) has a reduction ratio of 2:1. This ratio was chosen to obtain a clutch and cross-shaft speed of 15,000 rpm, which represents a preferred upper limit for shaft and clutch speed. It is desirable to make minimum reduction in rpm up to the final output stage and thereby minimize weight. The 2:1 ratio versus the 2.7:1 ratio of configuration I permits an engine transmission and cross-shaft weight saving for configuration II, thereby capitalizing on the capabilities of the free planet concept in the ratio range of 10:1 to 20:1.

Configurations III and IV (Figures 4 and 5) have the engines driving directly into the rotor transmission, thereby eliminating the engine boxes.

Main Transmission Bevel Gear Reduction

The bevel gear ratio for configuration I is dictated by the maximum reduction practical in a single-stage planetary (5.5:1). Having already established the engine transmission ratio at 2.7:1, the main transmission bevel ratio then becomes 5.4:1.

Configuration II has a bevel gear ratio of 3.24:1. This comes as a result of a free planet ratio of 12.9:1.

Configurations III and IV have the same bevel gear of 4.68:1. This ratio is dictated by a planetary ratio of 17.2:1 (since it must accommodate the ratio difference between the engine and rotor transmission.)

Main Transmission Planetary Reduction

Configuration I layout (Figure 7) has a planetary ratio of 5.5:1. This was considered to be a maximum practical reduction for a single-stage planetary system.

Configuration II layout (Figure 8) employs the free planetary transmission which is the primary candidate to be considered here. Its ratio is 12.9:1.

Configurations III and IV layouts (Figures 9 and 10) have identical ratios of 17.2:1. This enables a more direct comparison of the free planet transmission and conventional two-stage planetary transmission. The 17.2:1 ratio was selected because Boeing Vertol experience has proven this ratio near optimum for two stages in the CH-47 and HLH studies.

THE FREE PLANET ARRANGEMENT

The free planet arrangement used in this study is a simple arrangement and involves fewer parts than other more complex two-stage free planet schemes. Since complexity is generally related to higher cost and since the ratio needed fell between 10:1 and 20:1, the single-stage free planetary arrangement was the only configuration considered practical for the MUT application. Within this general arrangement, however, three free planet configurations were considered for the final layout, as follows:

1. A 12.9:1 planetary with 4 planets

2. A 12.9:1 planetary with 5 planets

3. A 17.2:1 planetary with 4 planets

Selecting, Sizing and Arranging the Free Planet Elements

In order to understand the relationship of free planet elements, a parametric study was conducted. Figures 11 through 17 show the parametric layouts. Gear size, center distance, height, and weight were investigated. The parametric study was conducted primarily for a 12.5 and 17.5 planetary ratio within the limitations of diameter and weight imposed by the MUT structure and within the restraints imposed by gear geometry (tooth length/gear diameter). The results of that study dictated the following parameters for the final design layout for both the 12.5 and 17.5 ratios:

1. Four planet spindles
2. A 9.0-inch planet center distance
3. A 16.7-inch-diameter stationary ring gear
4. A 14.5-inch-diameter output gear

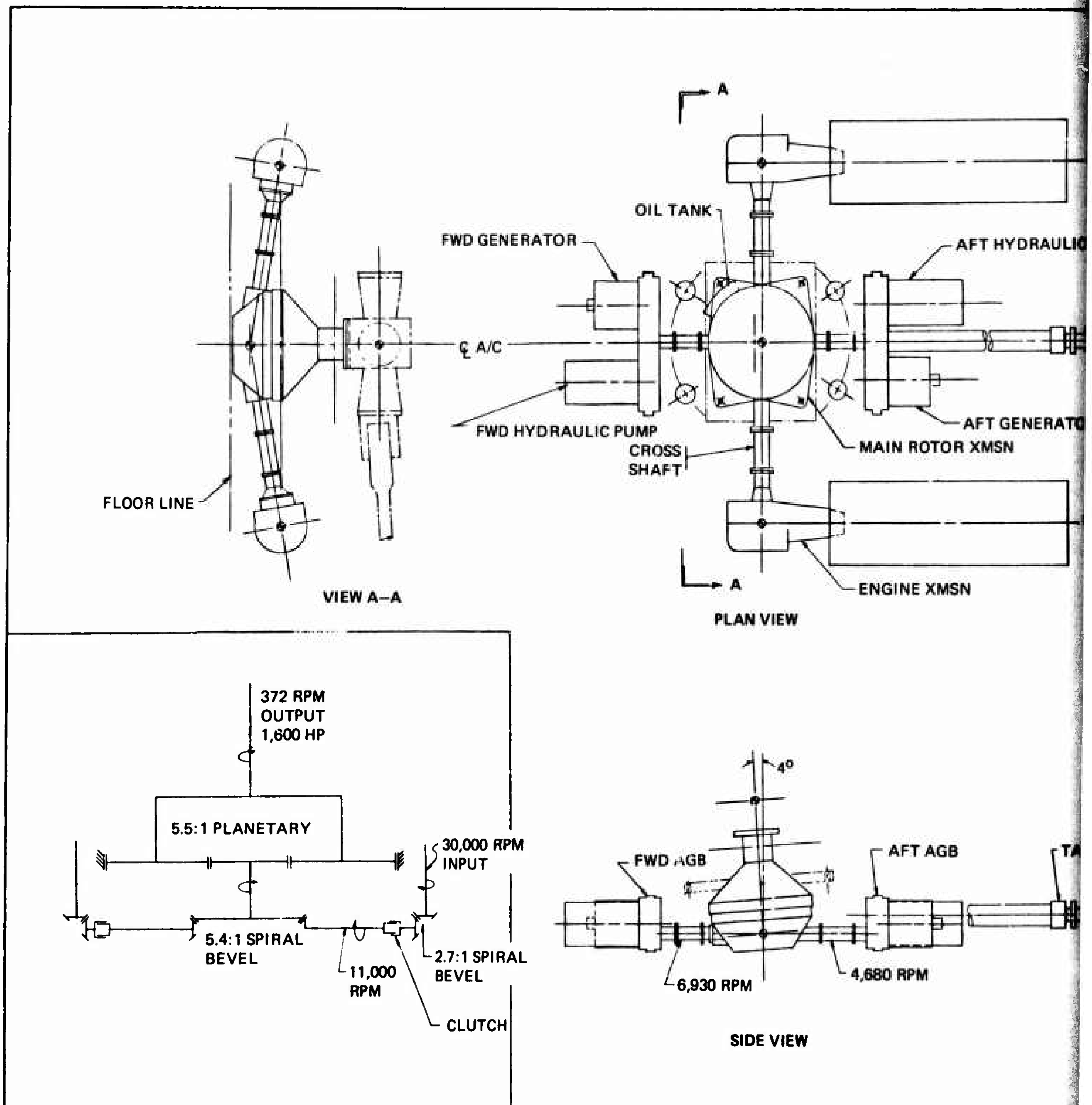
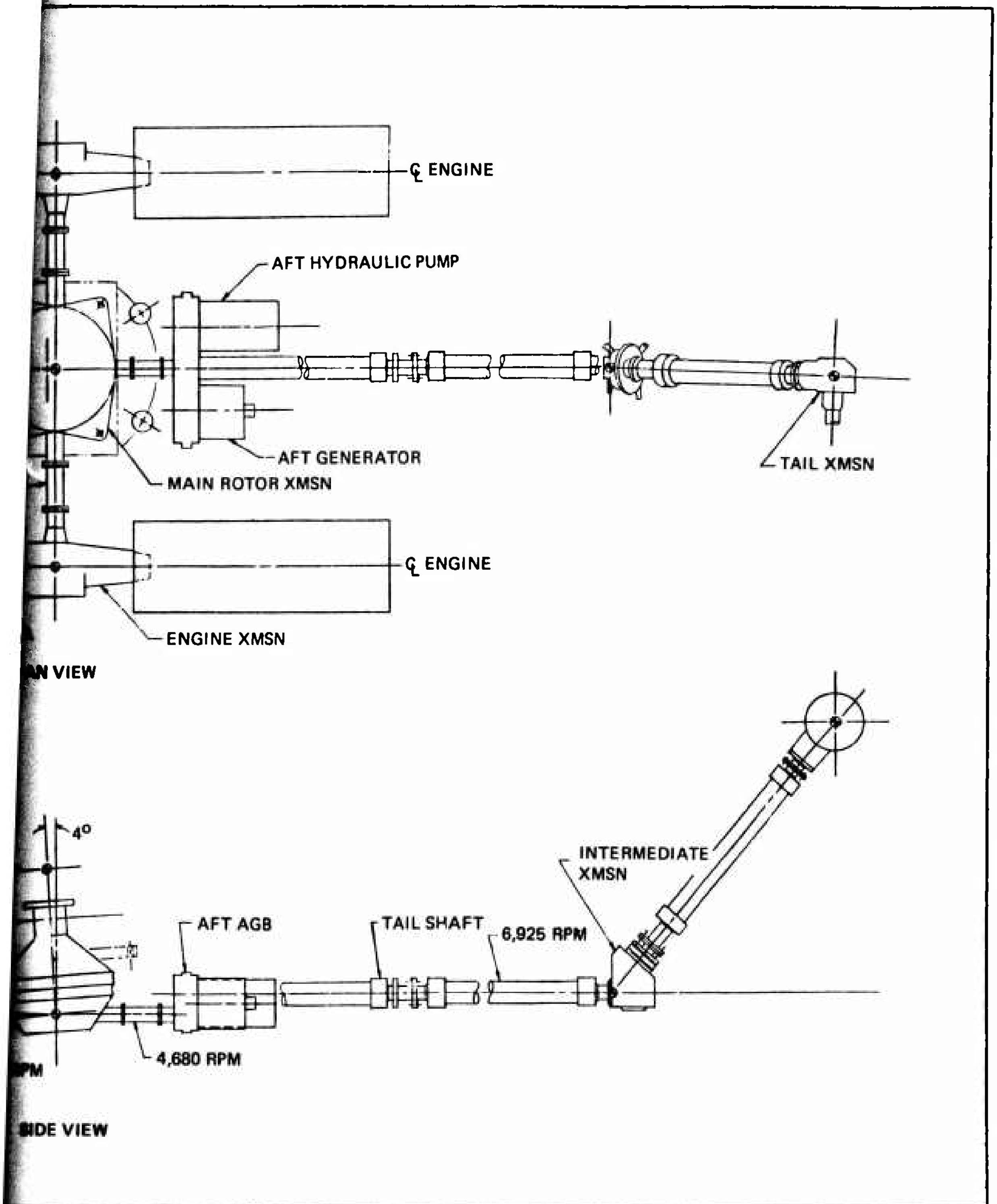


Figure 2. Conventional Transmission - Configuration No. 1.


℄ ENGINE
AFT HYDRAULIC PUMP
AFT GENERATOR
MAIN ROTOR XMSN
TAIL XMSN
℄ ENGINE
ENGINE XMSN
AN VIEW
4°
INTERMEDIATE
XMSN
AFT AGB
TAIL SHAFT
6,925 RPM
4,680 RPM
PM
SIDE VIEW

on No. 1.

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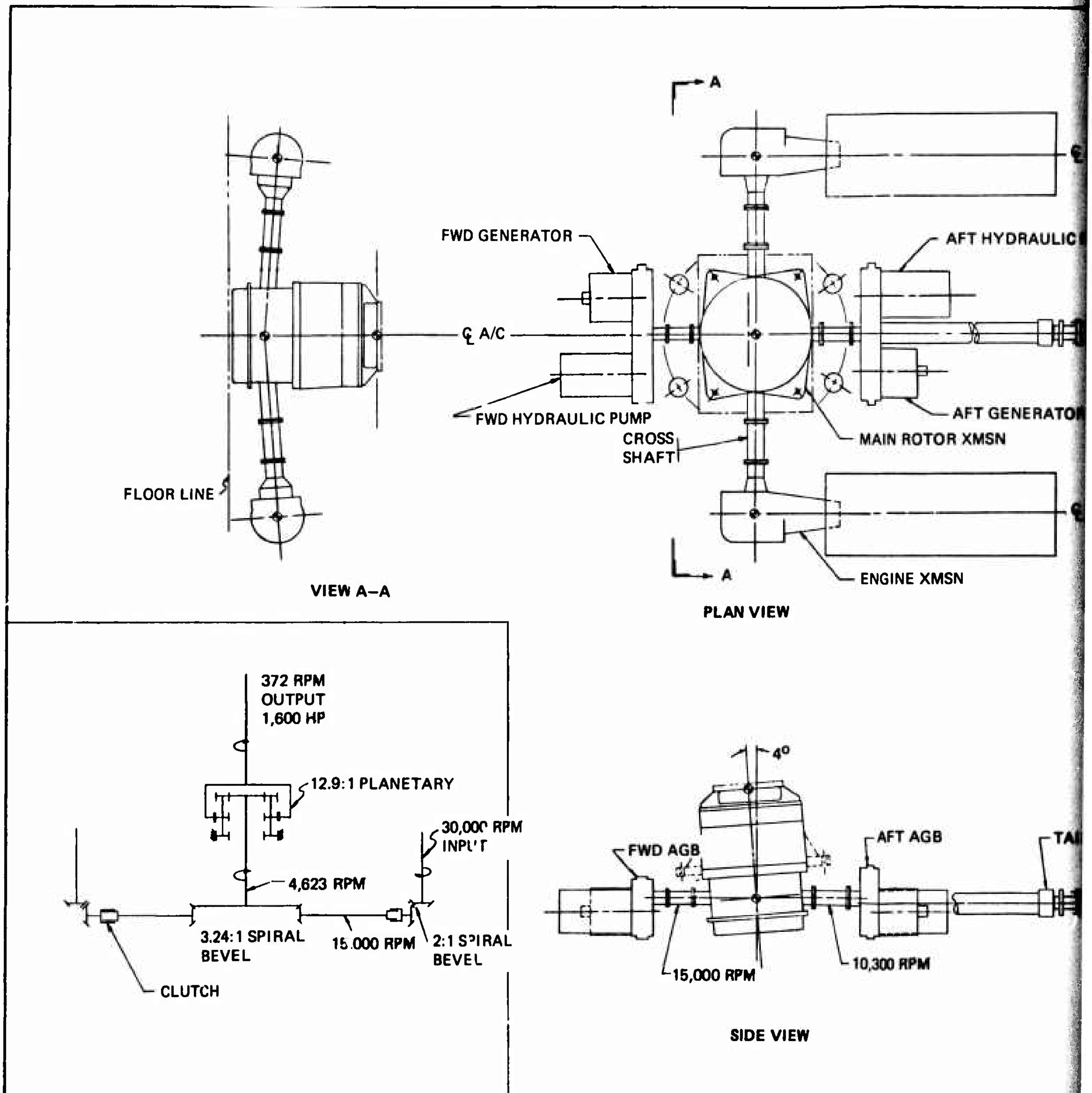
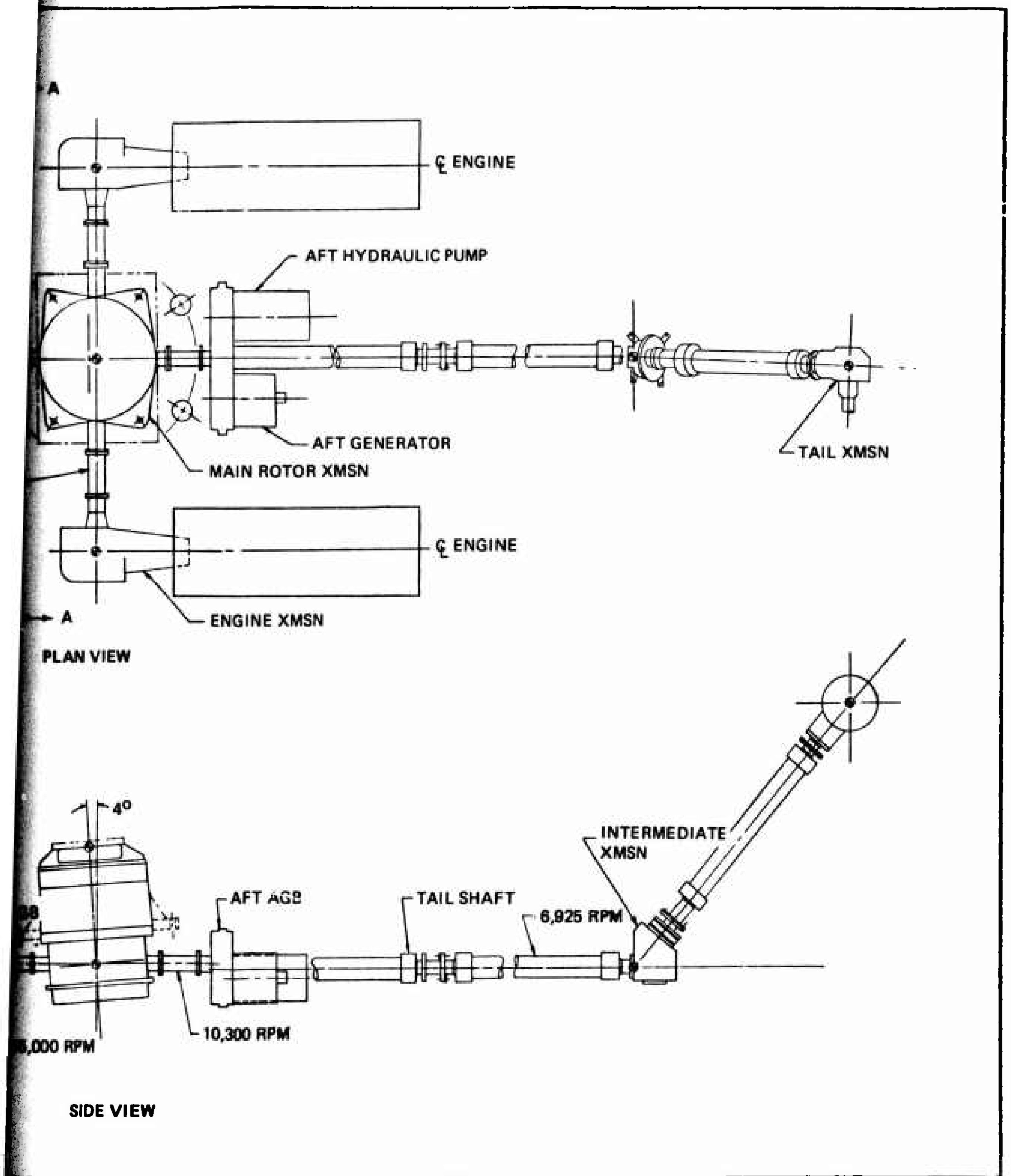


Figure 3. Free Planet Version of Configuration No. 1.


A
℄ ENGINE
AFT HYDRAULIC PUMP
AFT GENERATOR
TAIL XMSN
MAIN ROTOR XMSN
℄ ENGINE
ENGINE XMSN
A
PLAN VIEW
4°
INTERMEDIATE
XMSN
AFT AGB
TAIL SHAFT
6,925 RPM
10,300 RPM
,000 RPM
SIDE VIEW

No. 1.

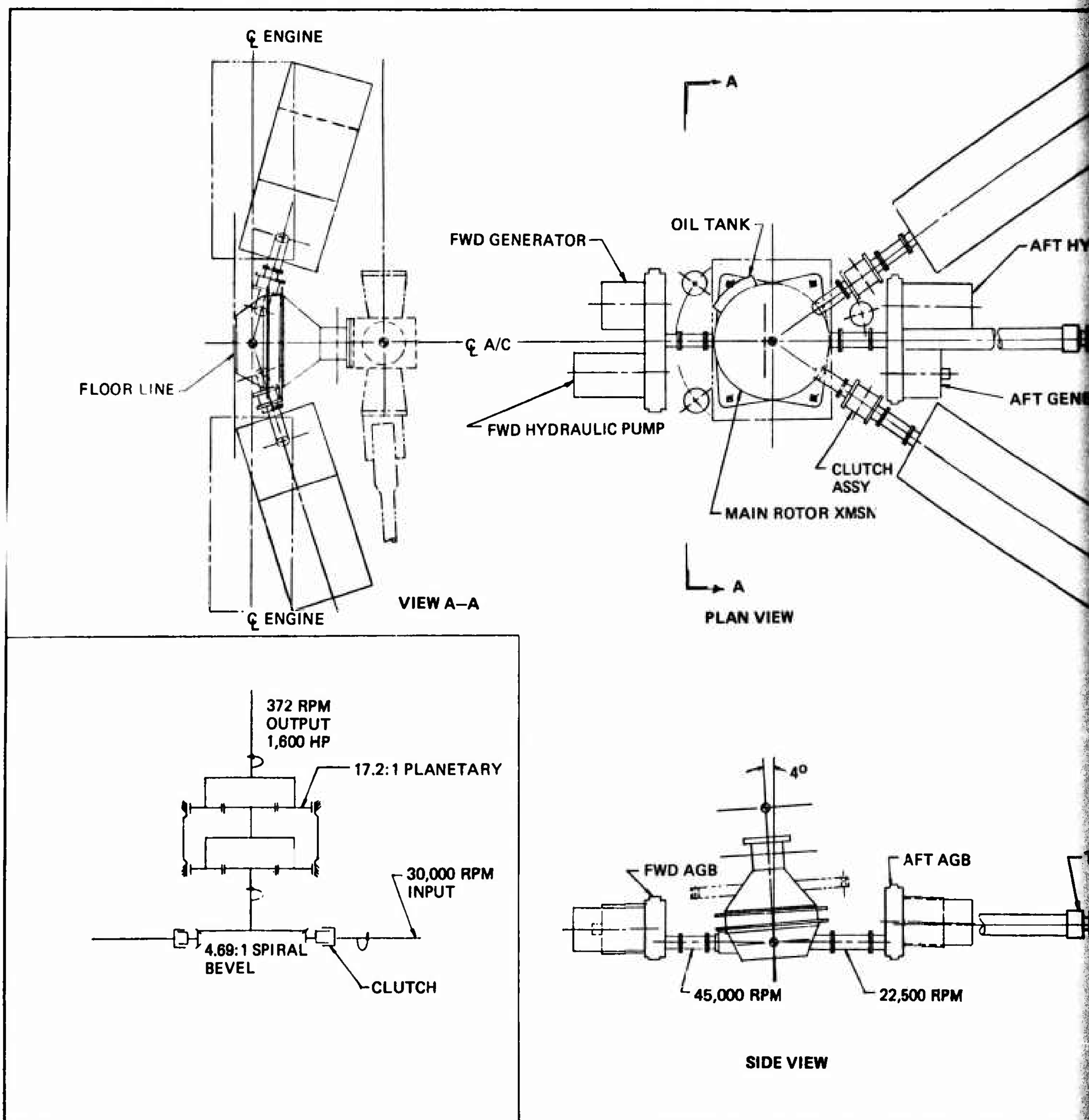
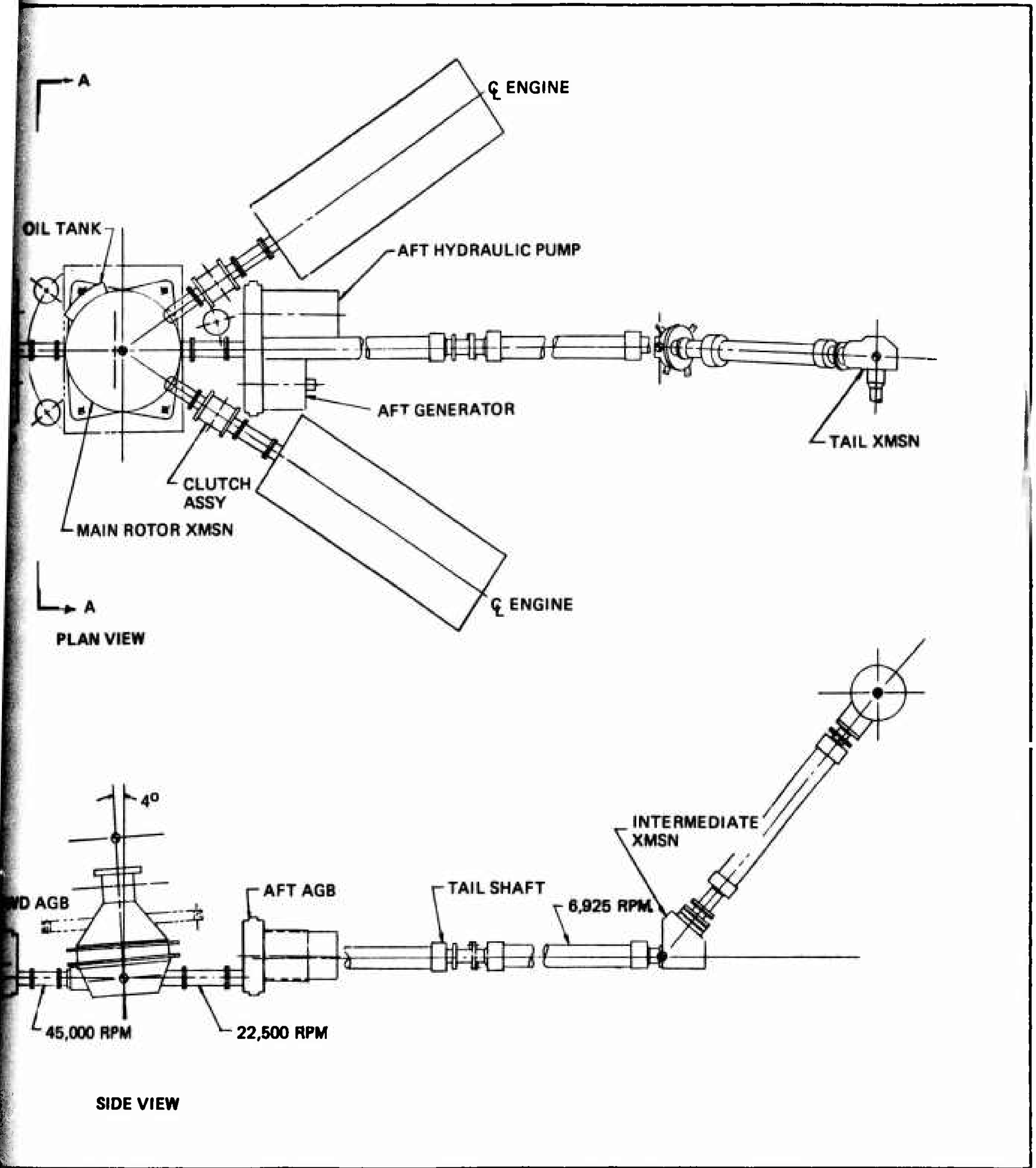


Figure 4. Conventional Version for Direct Engine Input.


A
℄ ENGINE
OIL TANK
AFT HYDRAULIC PUMP
AFT GENERATOR
TAIL XMSN
CLUTCH ASSY
MAIN ROTOR XMSN
℄ ENGINE
A
PLAN VIEW
4°
INTERMEDIATE XMSN
WD AGB
AFT AGB
TAIL SHAFT
6,925 RPM
45,000 RPM
22,500 RPM
SIDE VIEW

ngine Input.

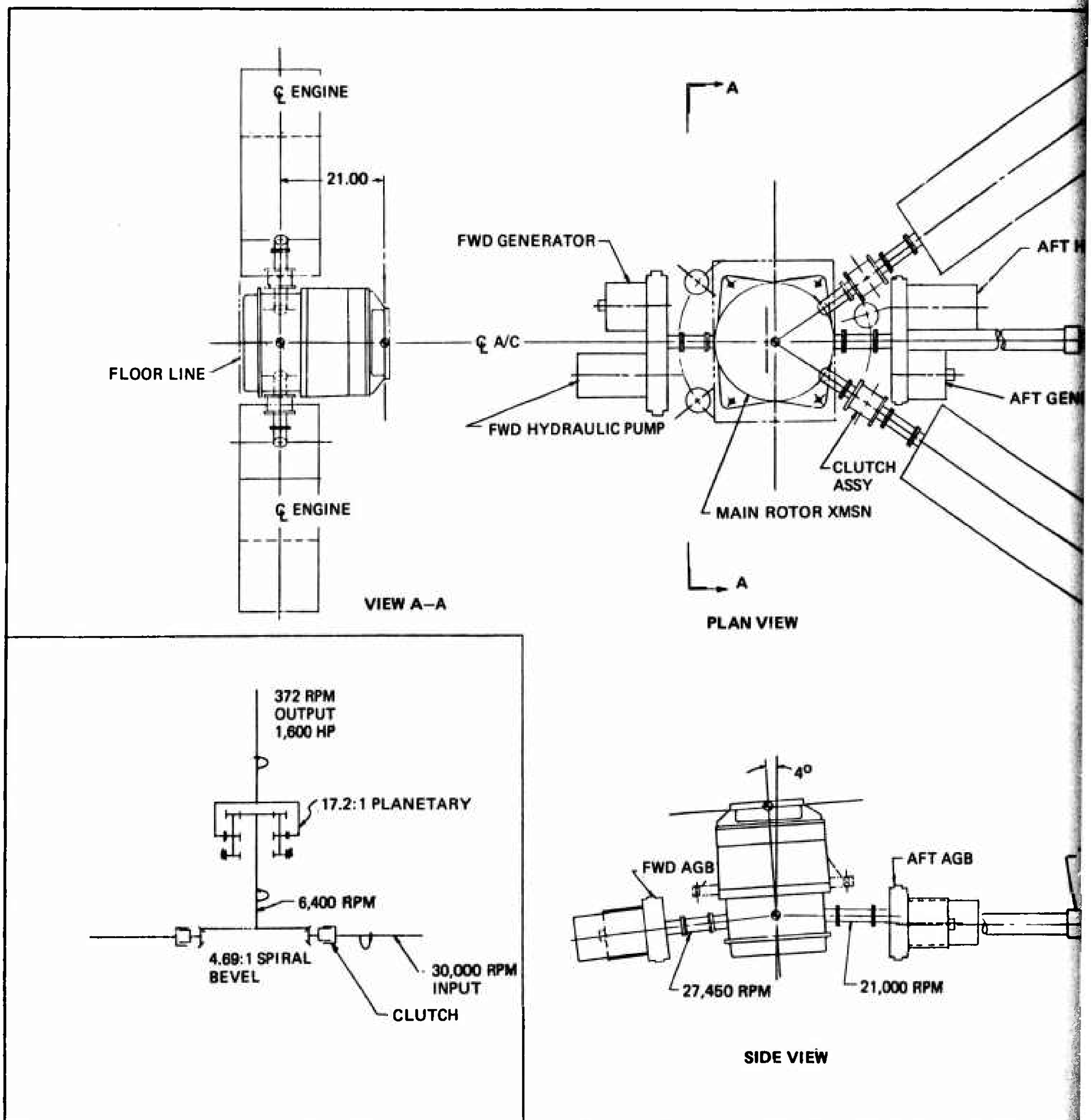
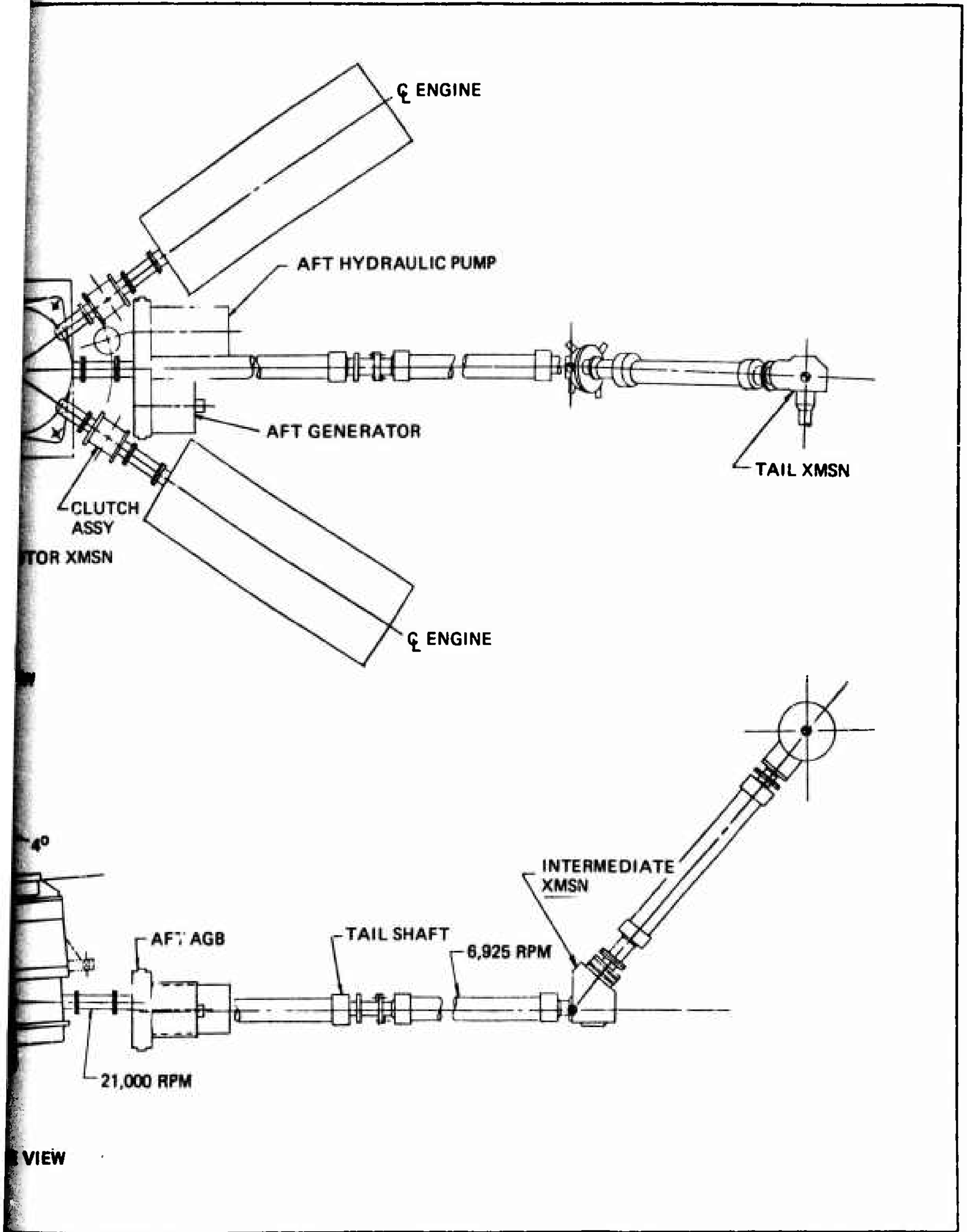


Figure 5. Free Planet Version for Direct Engine Input.


℄ ENGINE
AFT HYDRAULIC PUMP
AFT GENERATOR
TAIL XMSN
CLUTCH
ASSY
TOR XMSN
℄ ENGINE
4°
INTERMEDIATE
XMSN
AFT AGB
TAIL SHAFT
6,925 RPM
21,000 RPM
VIEW

ut.

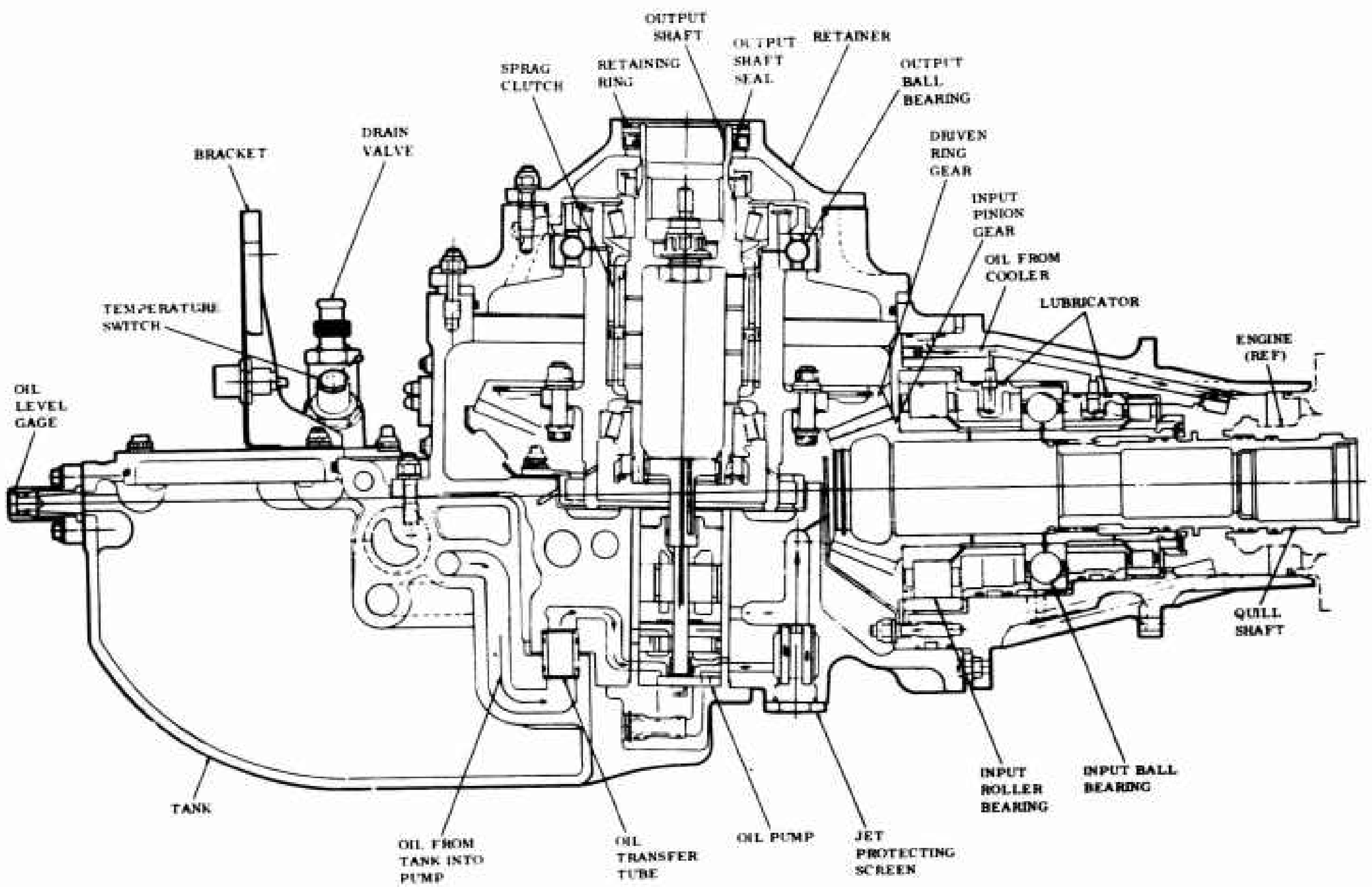


Figure 6. Engine Transmission with 2.7:1 Reduction.

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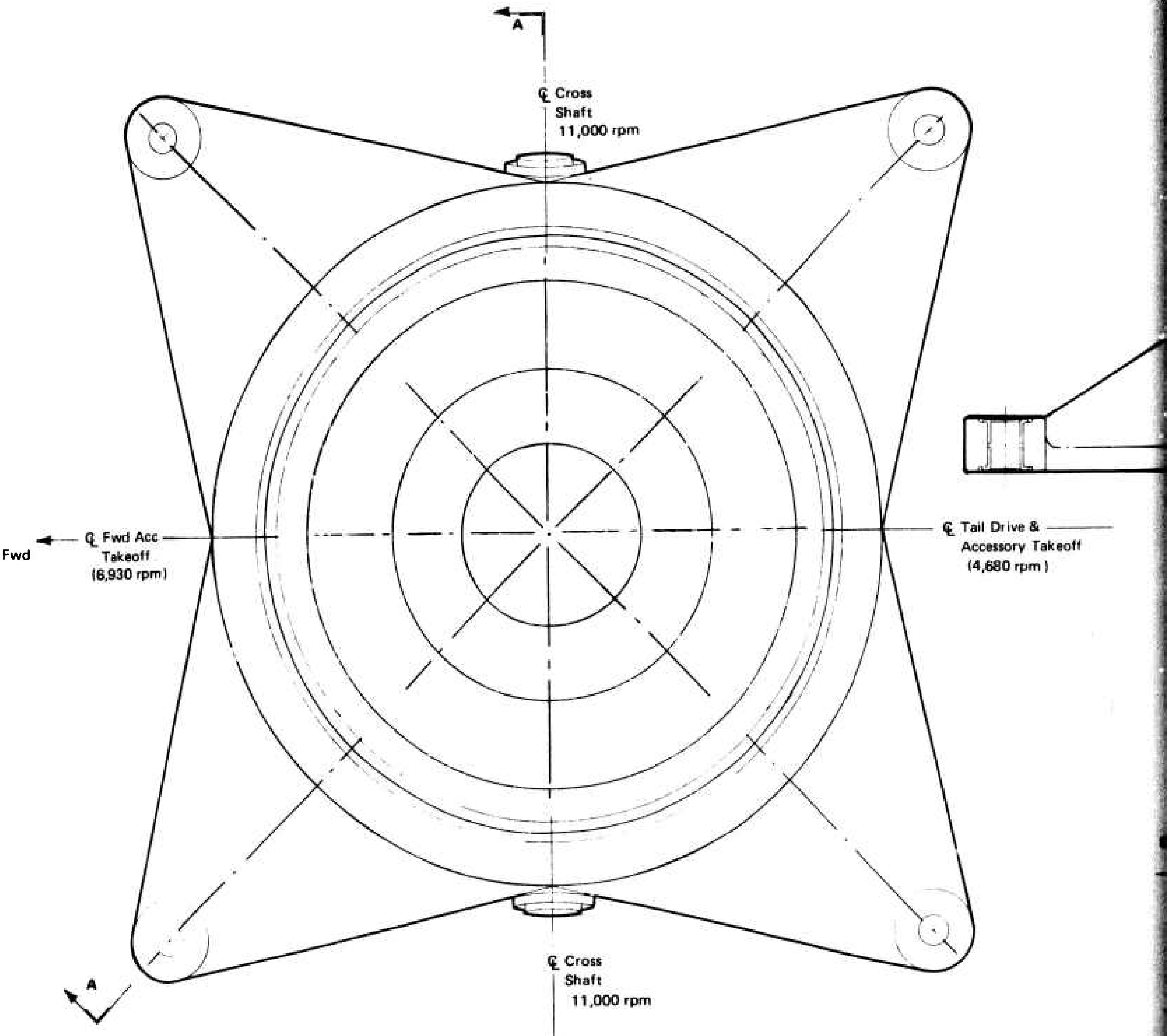
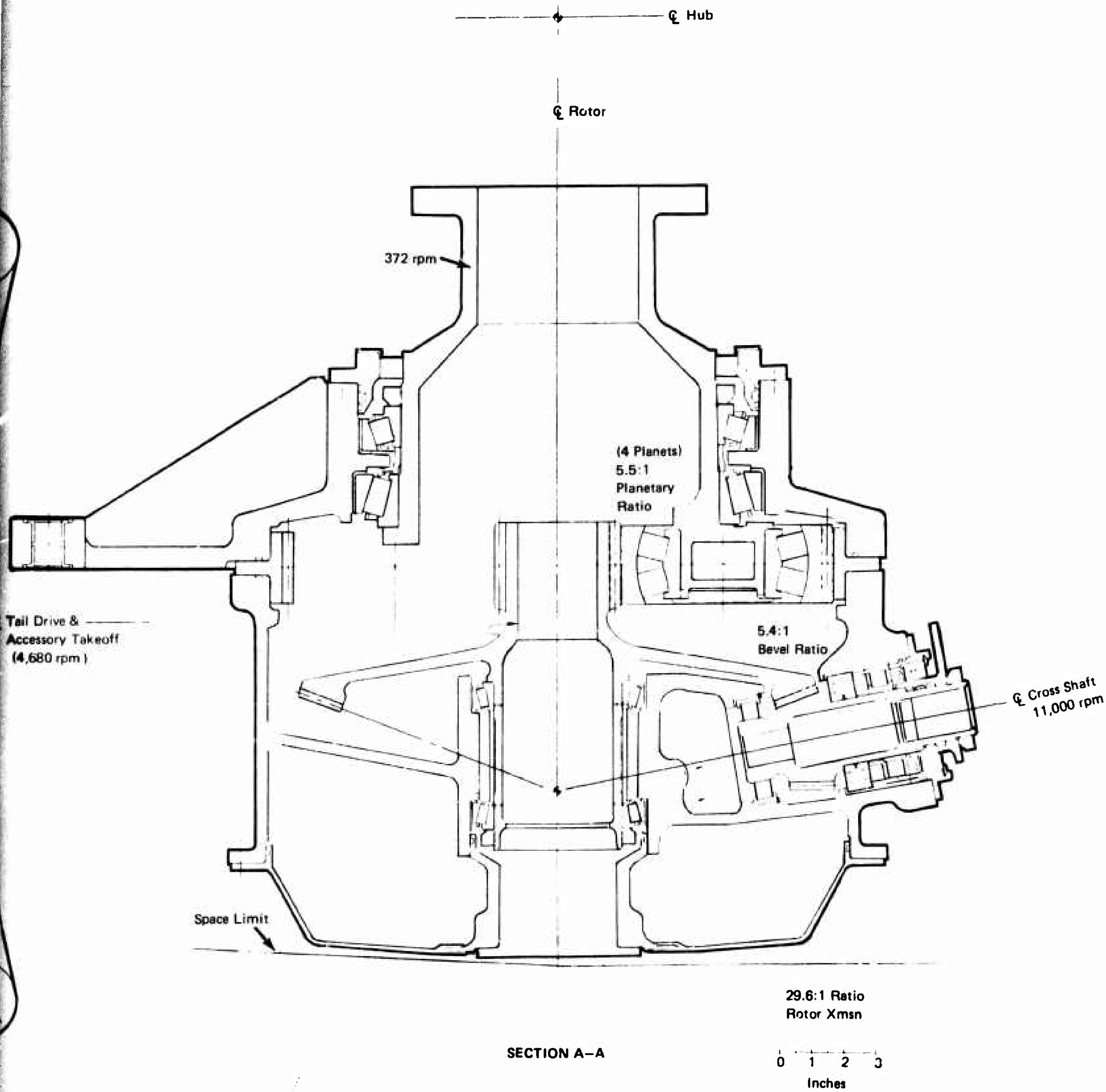


Figure 7. Free Planet Application Study Configuration I (5.5:1 Planetary Ratio, Conventional).


℄ Hub
℄ Rotor
372 rpm
(4 Planets)
5.5:1
Planetary
Ratio
Tail Drive &
Accessory Takeoff
(4,680 rpm)
5.4:1
Bevel Ratio
℄ Cross Shaft
11,000 rpm
Space Limit
29.6:1 Ratio
Rotor Xmsn
0 1 2 3
Inches

SECTION A–A

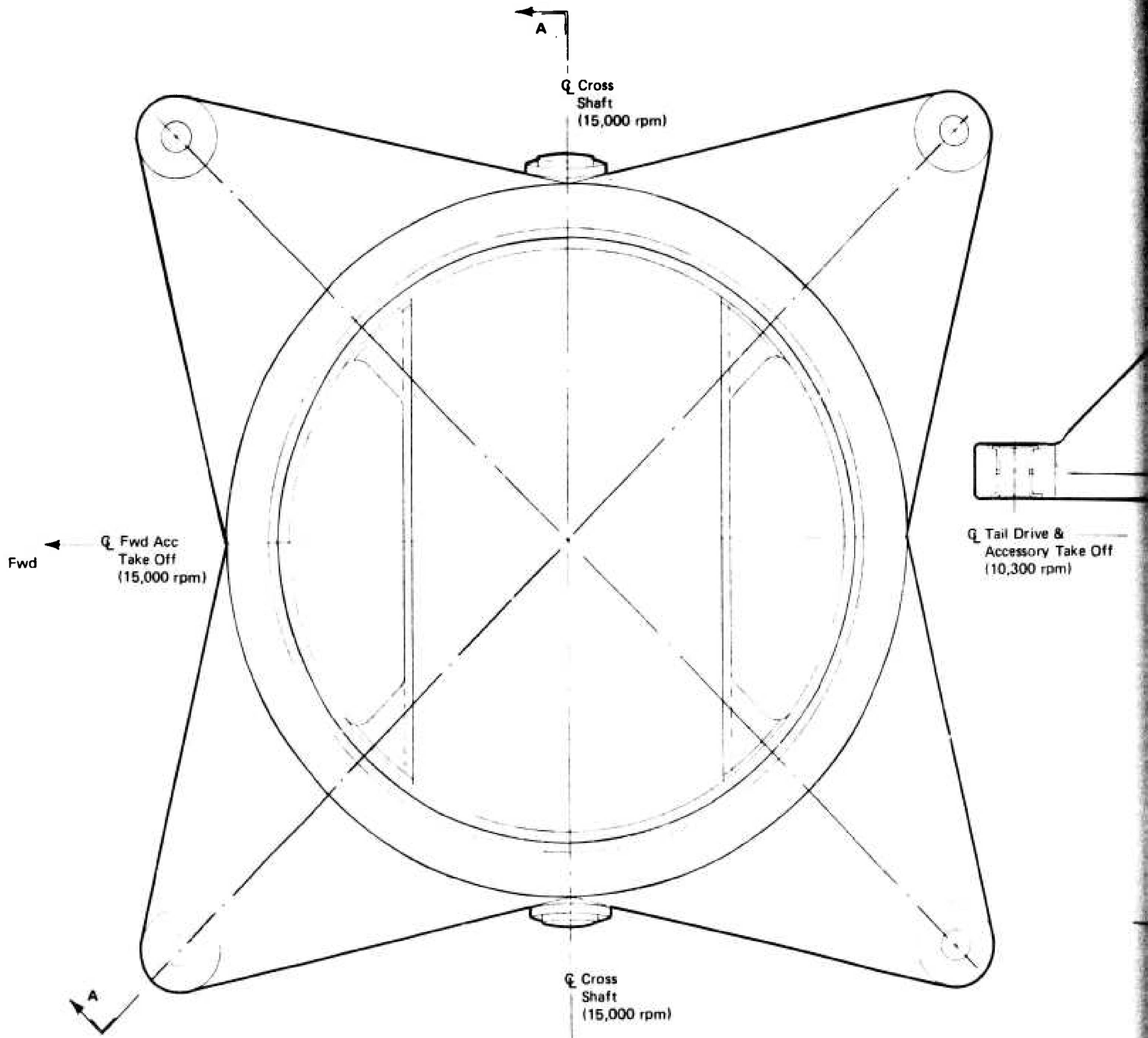
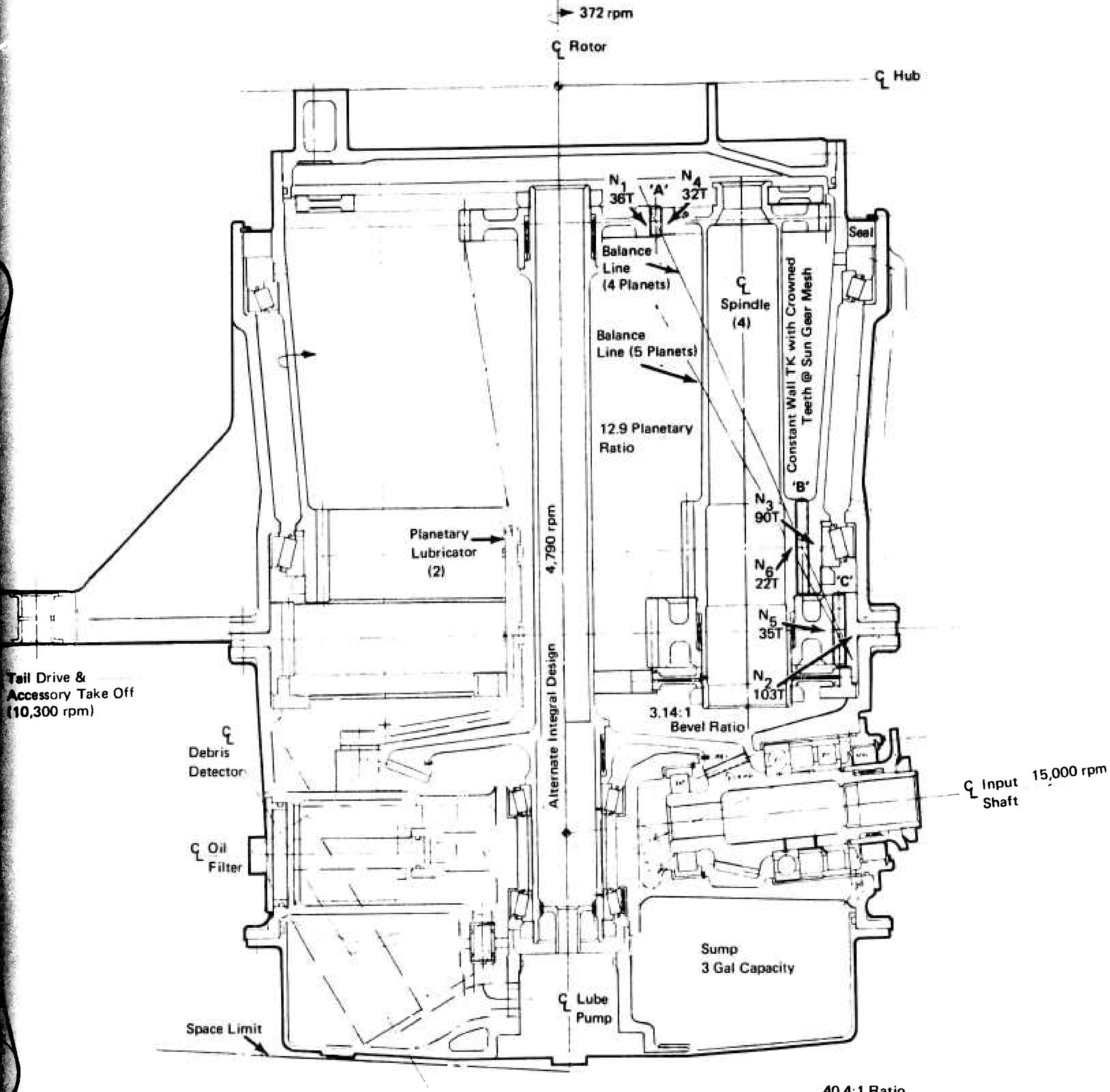


Figure 8. Free Planet Application Study Configuration II and IIa (12.9:1 Free Planetary).



SECTION A–A

40.4:1 Ratio
Rotor Xmsn

0 1 2 3
Inches

2

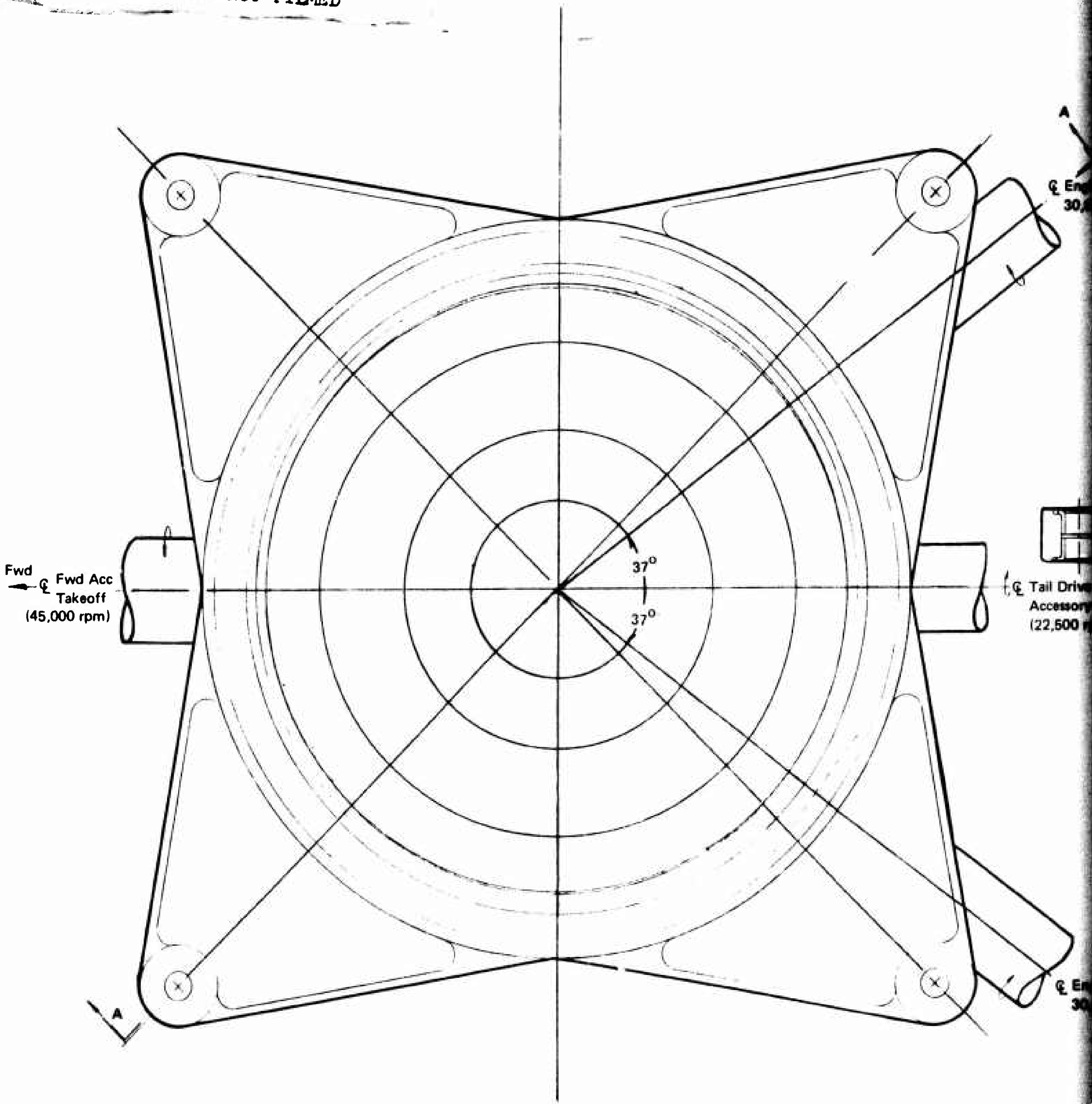
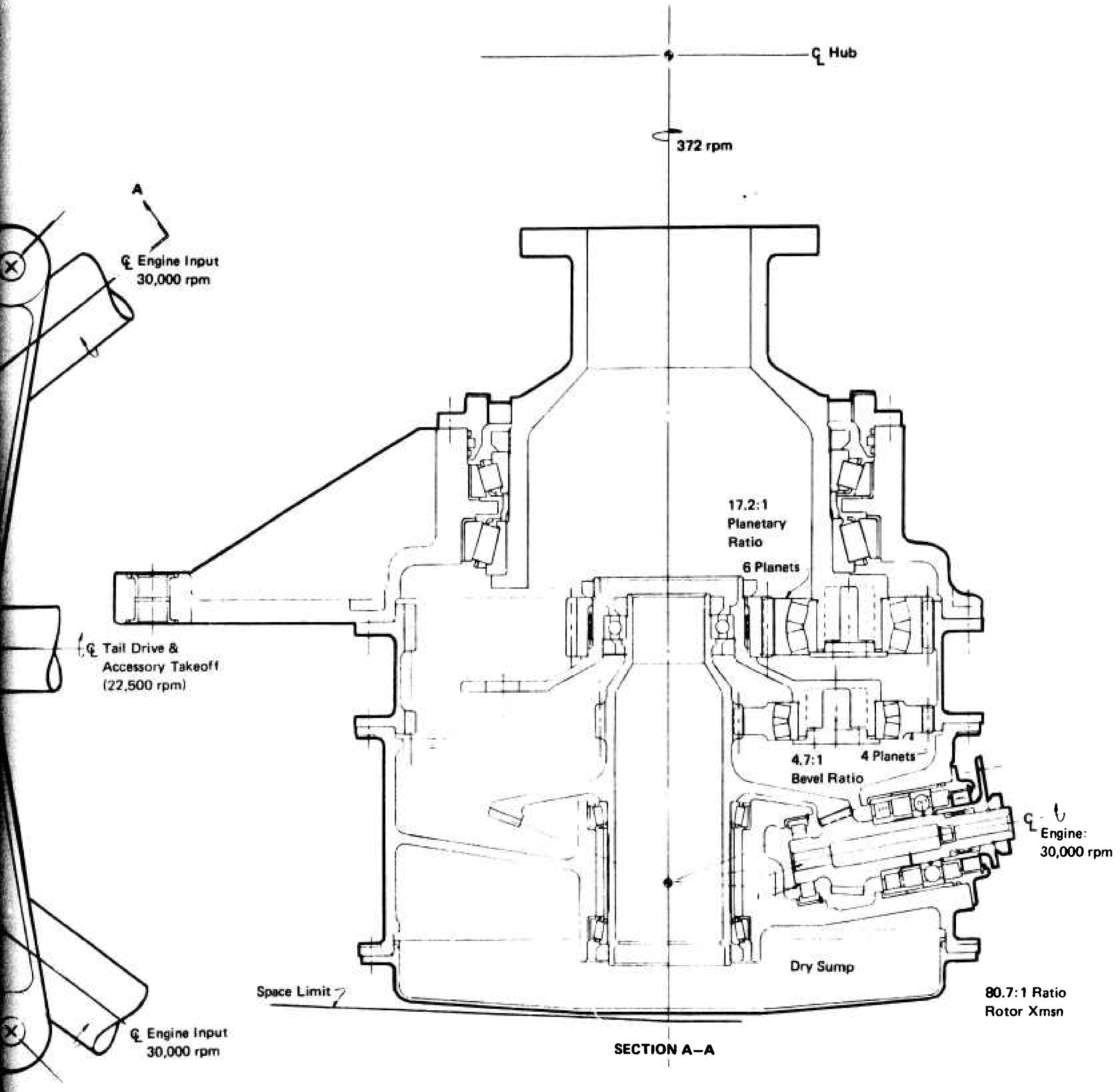


Figure 9. Free Planet Application Study Configuration III (17.2:1 Conventional Planetary).


℄ Hub
372 rpm
A
℄ Engine Input
30,000 rpm
17.2:1
Planetary
Ratio
6 Planets
℄ Tail Drive &
Accessory Takeoff
(22,500 rpm)
4.7:1
Bevel Ratio
4 Planets
℄ Engine:
30,000 rpm
Dry Sump
Space Limit
80.7:1 Ratio
Rotor Xmsn
℄ Engine Input
30,000 rpm

SECTION A–A

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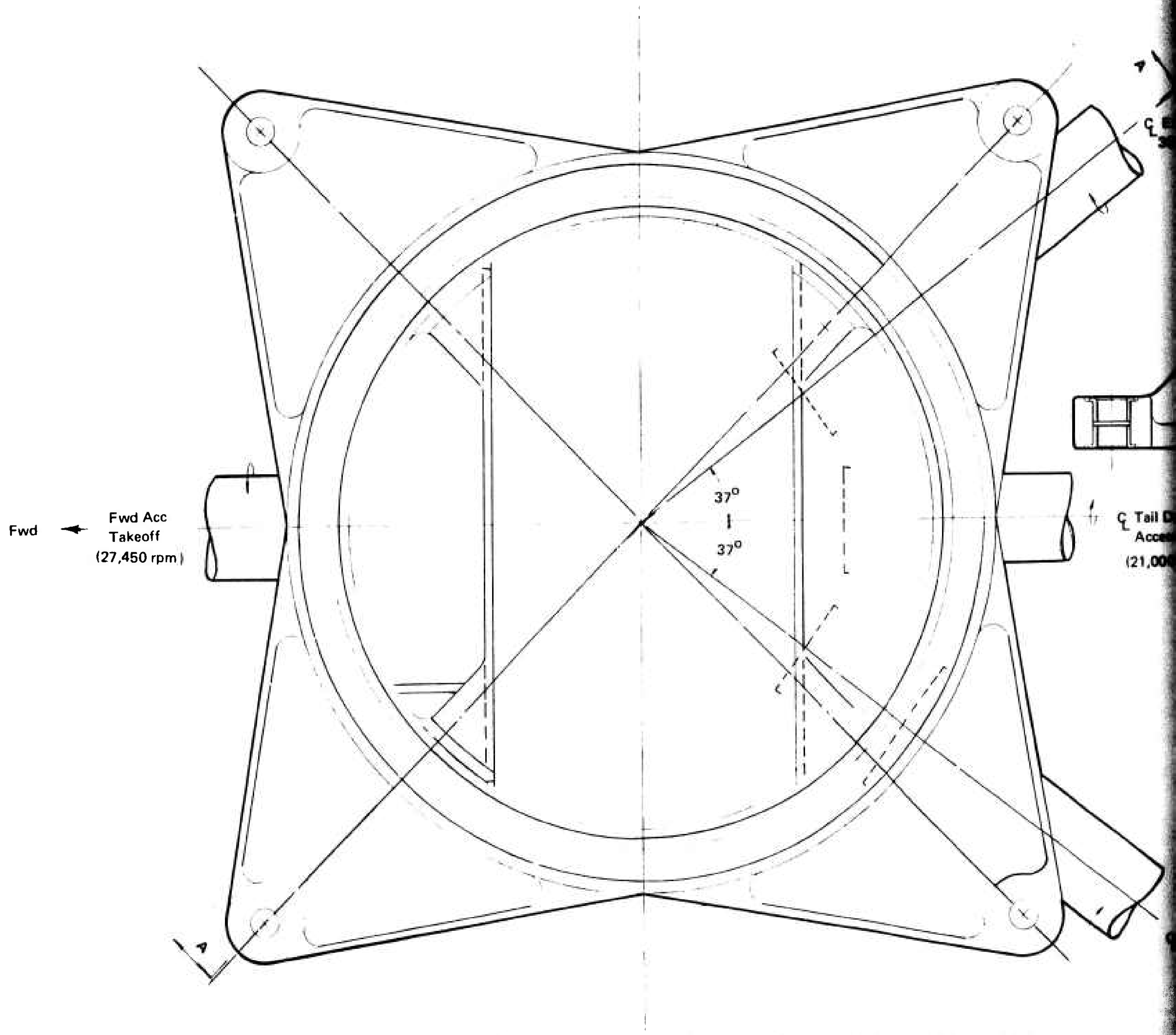
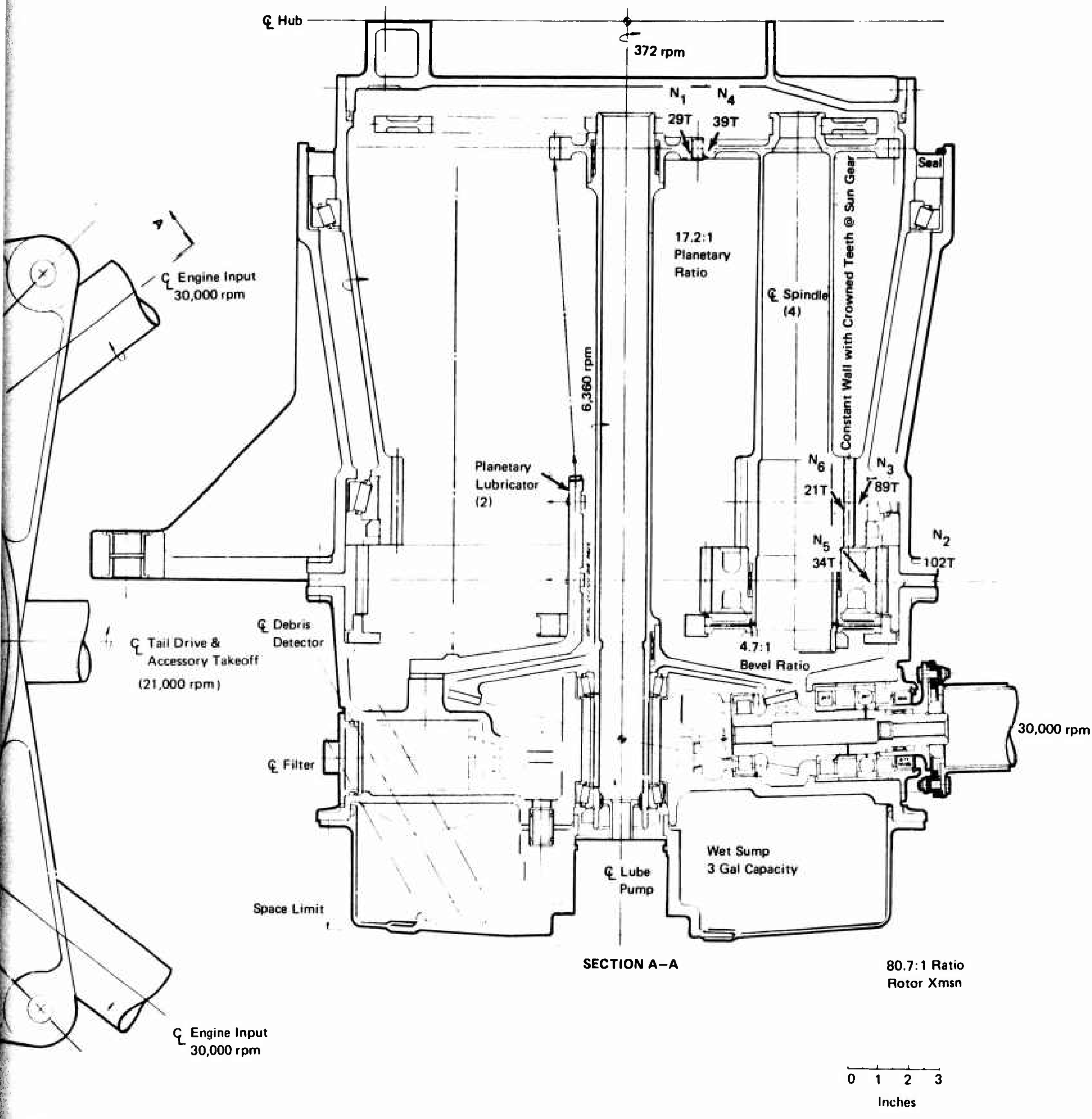


Figure 10. Free Planet Application Study Configuration IV (17.2:1 Free Planetary).



℄ Hub
372 rpm
N_1
N_4
29T
39T
Seal
17.2:1
Planetary
Ratio
℄ Spindle
(4)
Constant Wall with Crowned Teeth @ Sun Gear
A
℄ Engine Input
30,000 rpm
6,360 rpm
Planetary
Lubricator
(2)
N_6
21T
N_3
89T
N_5
34T
N_2
102T
℄ Debris
Detector
℄ Tail Drive &
Accessory Takeoff
(21,000 rpm)
4.7:1
Bevel Ratio
30,000 rpm
℄ Filter
Wet Sump
3 Gal Capacity
℄ Lube
Pump
Space Limit
SECTION A–A
80.7:1 Ratio
Rotor Xmsn
℄ Engine Input
30,000 rpm
0 1 2 3
Inches

Cut No. 1

N_1	=	4.167	–	25T
N_2	=	13.833	–	83T
N_3	=	11.667	–	70T
N_4	=	4.833	–	29T
N_5	=	4.833	–	29T
N_6	=	2.667	–	16T

Comments

- Initial Try & Baseline for Comparison

- Descend Balance Line by Reducing N_4 Dia & Increasing N_1 Dia

12.5:1 Ratio
5 Planets
9.0 in. C/D

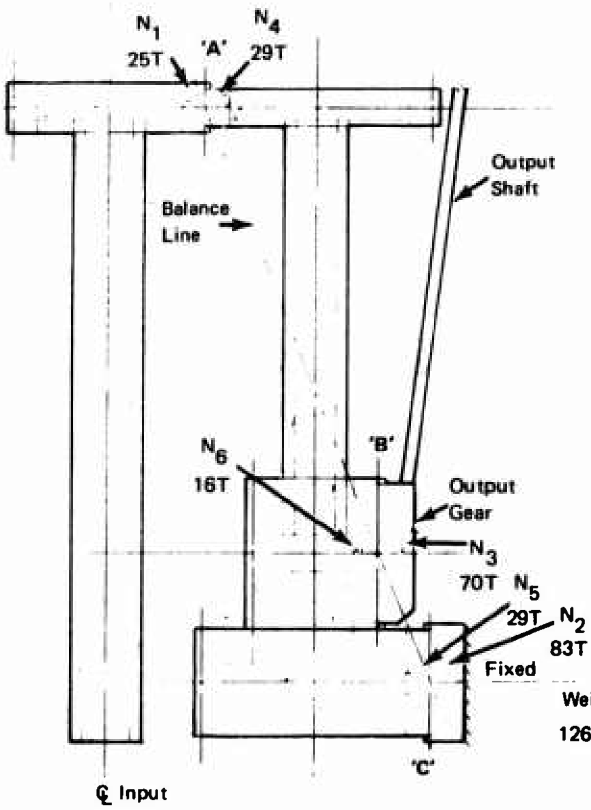
Cut No. 2

Change →	N_1	=	6.167	–	37T
Maintain →	N_2	=	13.833	–	83T
	N_3	=	12.667	–	71T
Change →	N_4	=	2.833	–	17T
	N_5	=	4.833	–	29T
	N_6	=	3.667	–	22T

Comments

- Increase in Weight & Height Since N Have More Influ on Height

12.5:1 Ratio
5 Planets
9.0 in. C/D



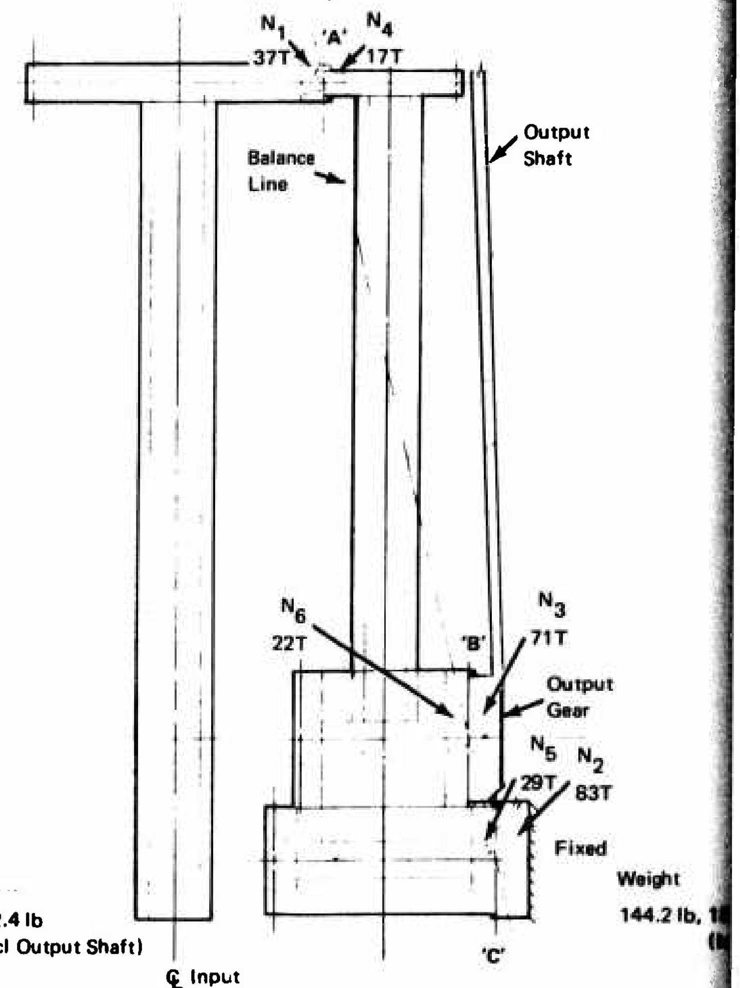


Figure 11. Free Planet Study Parametric Study (1,604 hp, 372 rpm).

No. 2

=	6.167	–	37T
=	13.833	–	83T
=	12.667	–	71T
=	2.833	–	17T
=	4.833	–	29T
=	3.667	–	22T

:1 Ratio
nets
in. C/D

Comments

- Increase in Weight & Height Since N_5 & N_6 Have More Influence on Height

- Increase N_2 Dia & Center Dist & Maintain N_1/N_4 Relationship of Cut No. 1

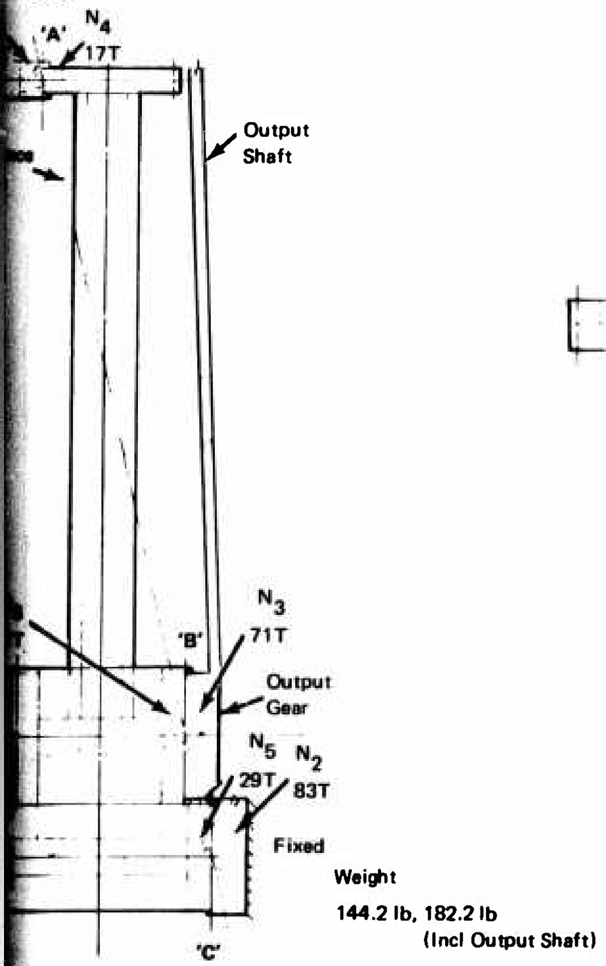
Cut No. 3

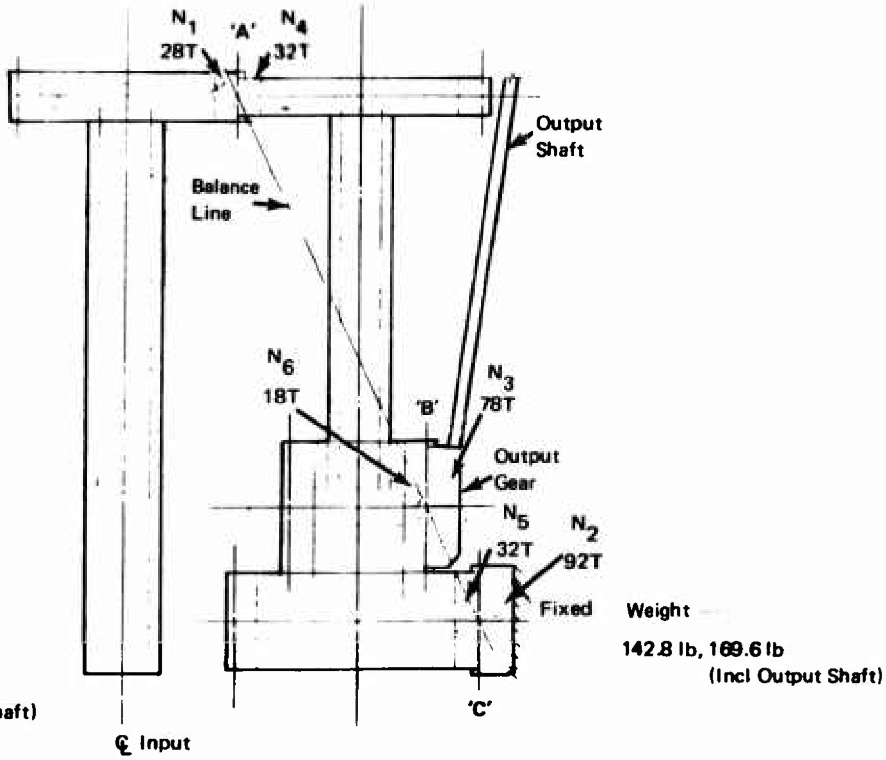
	N_1	=	4.667	–	28T
Change →	N_2	=	15.333	–	92T
	N_3	=	13.000	–	78T
	N_4	=	5.333	–	32T
	N_5	=	5.333	–	32T
	N_6	=	3.000	–	18T

12.5:1 Ratio
5 Planets
Change → 10.0 in. C/D

Comments

- Lower Height & Increase in Weight (Planet Gears), Larger Dia





2

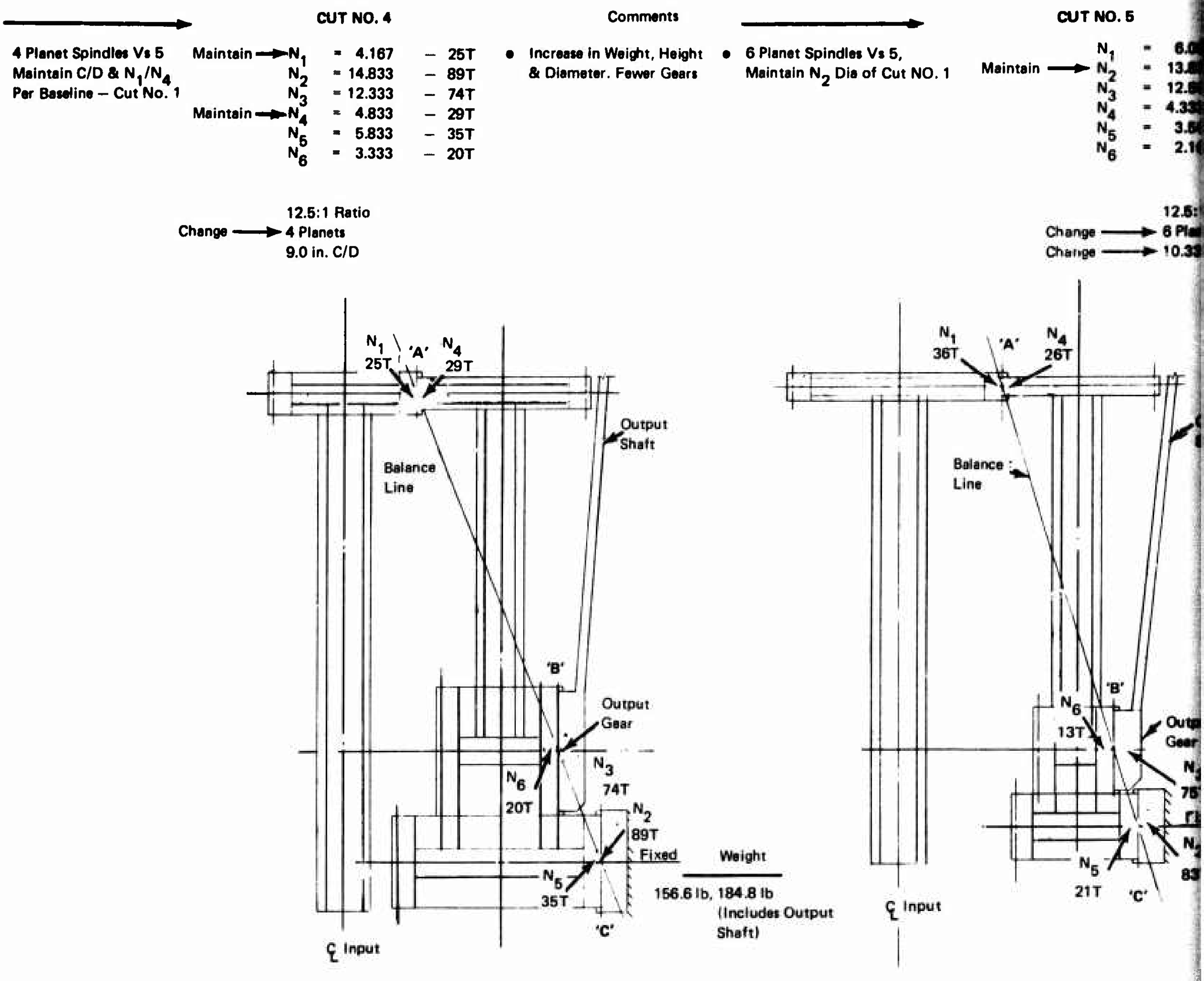


Figure 12. Free Planet Study Parametric Study (1,604 hp, 372 rpm).

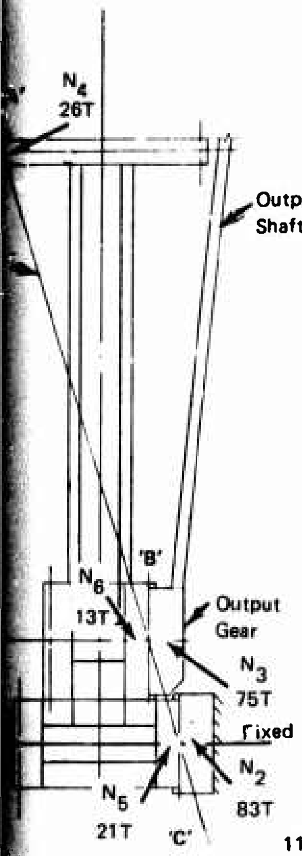
CUT NO. 5

	N_1	=	6.000	– 36T
tain →	N_2	=	13.833	– 83T
	N_3	=	12.500	– 75T
	N_4	=	4.333	– 26T
	N_5	=	3.500	– 21T
	N_6	=	2.167	– 13T

Comments

- Reduced Weight. Same Envelope. More Gears

12.5:1 Ratio
Change → 6 Planets
Change → 10.333 C/D



Weight

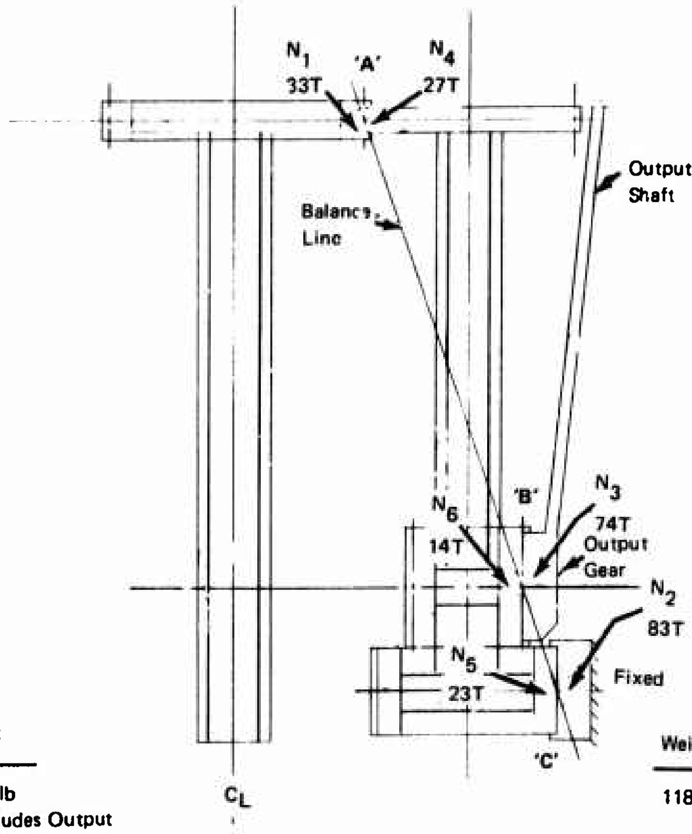
112.2 lb, 143 lb
(Includes Output
Shaft)

→

- Same as Cut No. 5, except reduce C/D

Maintain

12.5:1 Ratio
6 Planets
Change → 10.000 in. C/D



CUT NO. 6

N_1	=	5.500	– 33T
N_2	=	13.833	– 83T
N_3	=	12.333	– 74T
N_4	=	4.500	– 27T
N_5	=	3.833	– 23T
N_6	=	2.333	– 14T

Comments

- Reduced weight.
Reduced height.
More gears.
(Incr wt vs Cut No. 5)

Weight

118.1 lb, 147.4 lb
(Includes Output
Shaft)

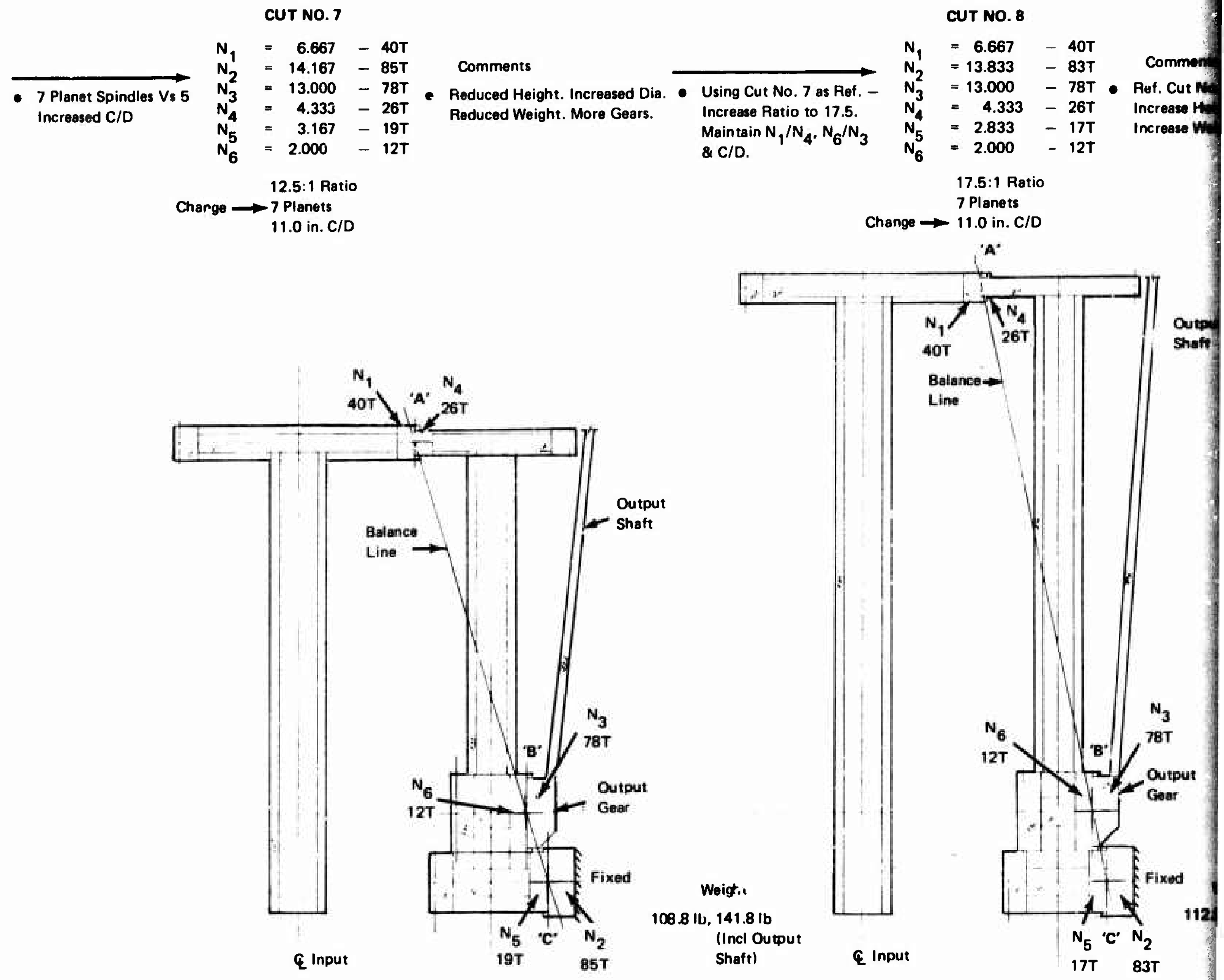


Figure 13. Free Planet Study Parametric Study (1,604 hp, 372 rpm).

CUT NO. 8

N_1 = 6.667 – 40T
N_2 = 13.833 – 83T
N_3 = 13.000 – 78T
N_4 = 4.333 – 26T
N_5 = 2.833 – 17T
N_6 = 2.000 – 12T

17.5:1 Ratio
7 Planets
ange → 11.0 in. C/D

Comments

- Ref. Cut No. 7 – Increase Height. Increase Weight.

- Using Cut No. 7 as Comparison – Maintain N_2. Reduce No. of Planet Spindles. (Gear Geom. of N_6 Requires C/D Decrease)

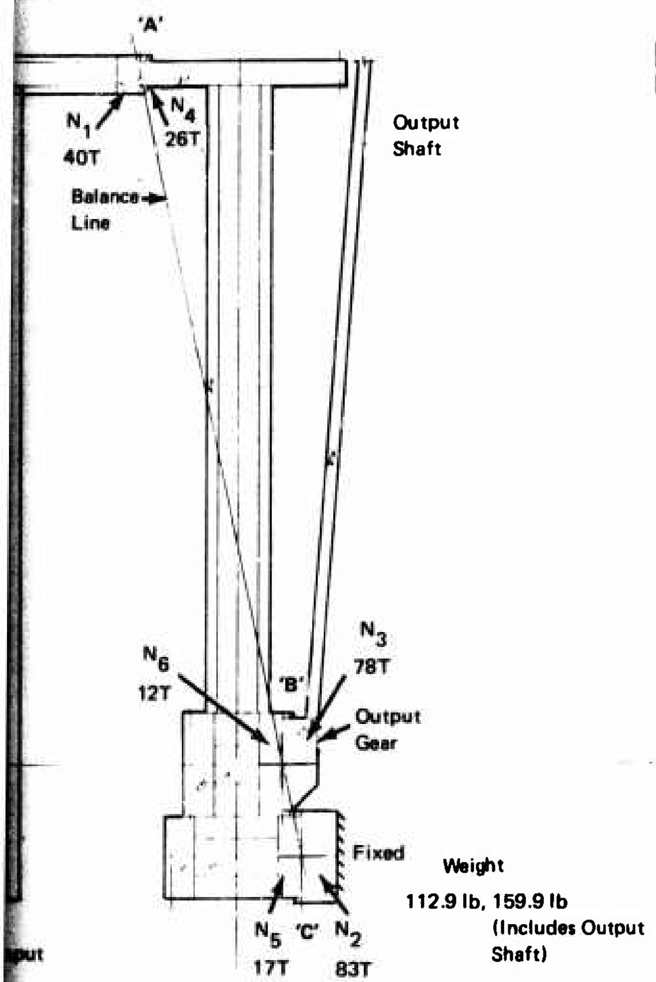
CUT NO. 9

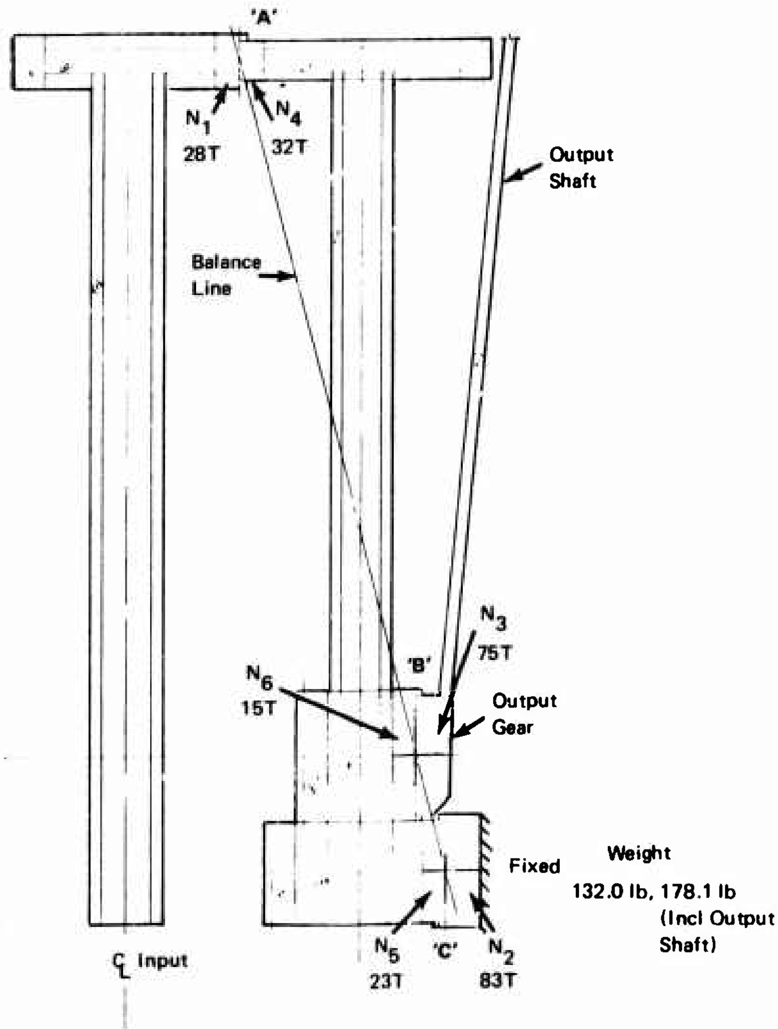
N_1 = 4.667 – 28T
N_2 = 13.833 – 83T
N_3 = 12.500 – 75T
N_4 = 5.333 – 32T
N_5 = 3.833 – 23T
N_6 = 2.500 – 15T

17.5:1 Ratio
Change → 5 Planets
10.0 in. C/D

Comments

- Ref. Cut No. 8 – Increase Height.





rpm).

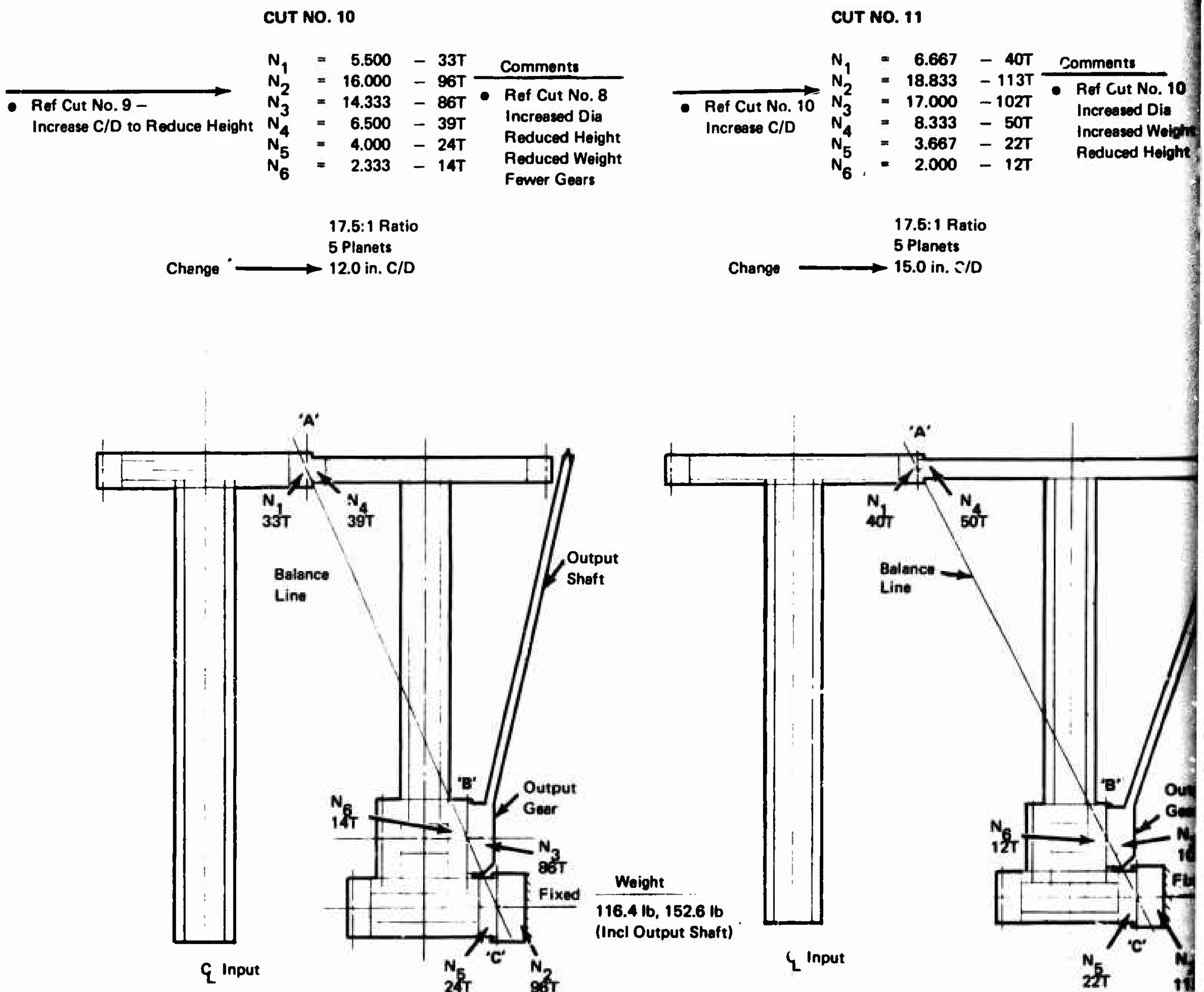


Figure 14. Free Planet Study Parametric Study (1,604 hp, 372 rpm).

NO. 11

=	6.667	– 40T
=	18.833	– 113T
=	17.000	– 102T
=	8.333	– 50T
=	3.667	– 22T
=	2.000	– 12T

Comments
- Ref Cut No. 10
  Increased Dia
  Increased Weight
  Reduced Height

17.5:1 Ratio
5 Planets
15.0 in. C/D

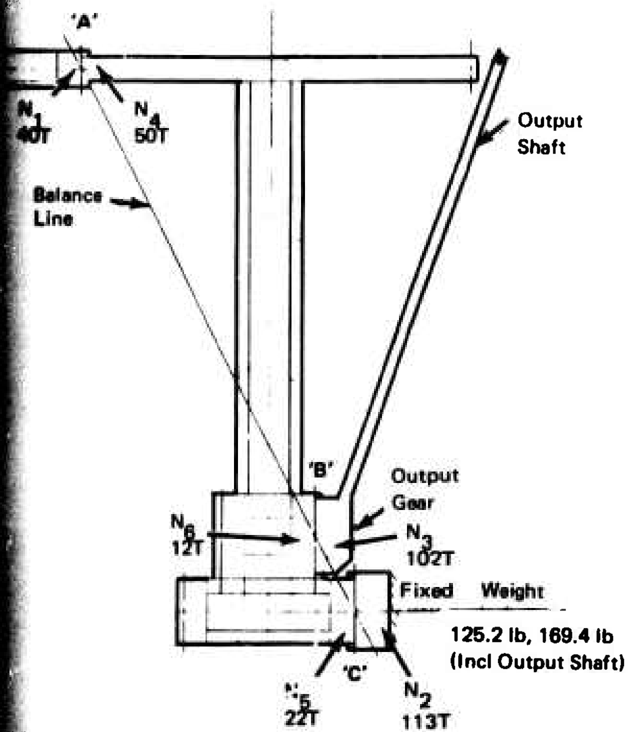
- Explore 27.0 Ratio

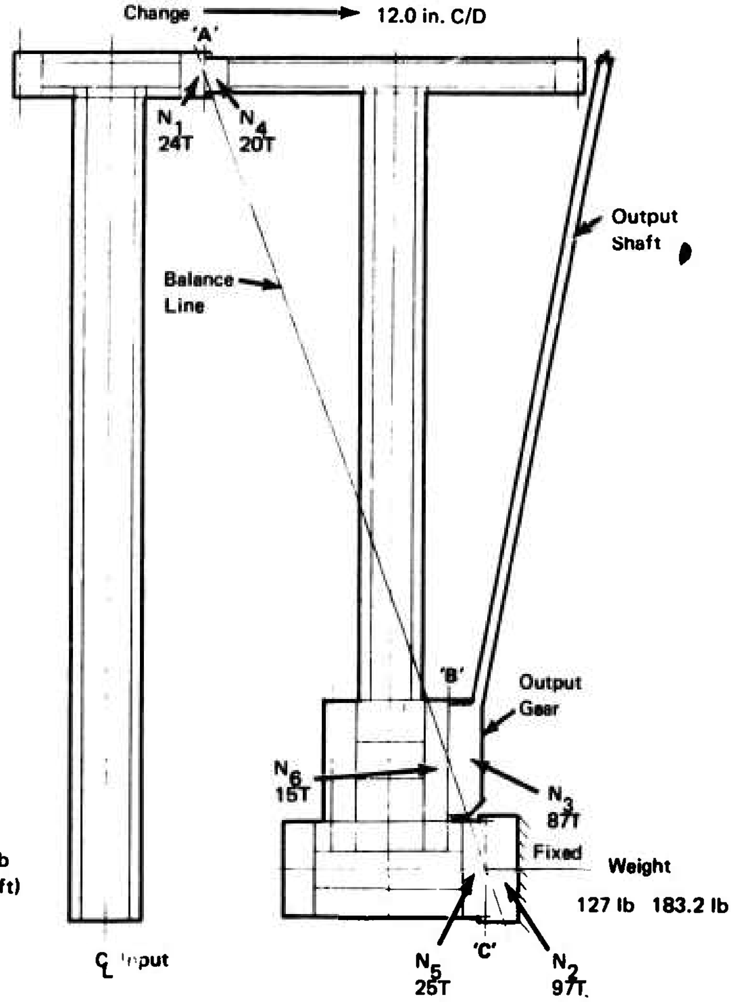
CUT NO. 12

N_1	=	4.000	– 24T
N_2	=	16.167	– 97T
N_3	=	14.500	– 87T
N_4	=	8.000	– 20T
N_5	=	4.167	– 25T
N_6	=	2.500	– 15T

Comments
- Increased Weight
  Increased Height
  Fewer Gears

Change → 27.0:1 Ratio
Change → 4 Planets
Change → 12.0 in. C/D





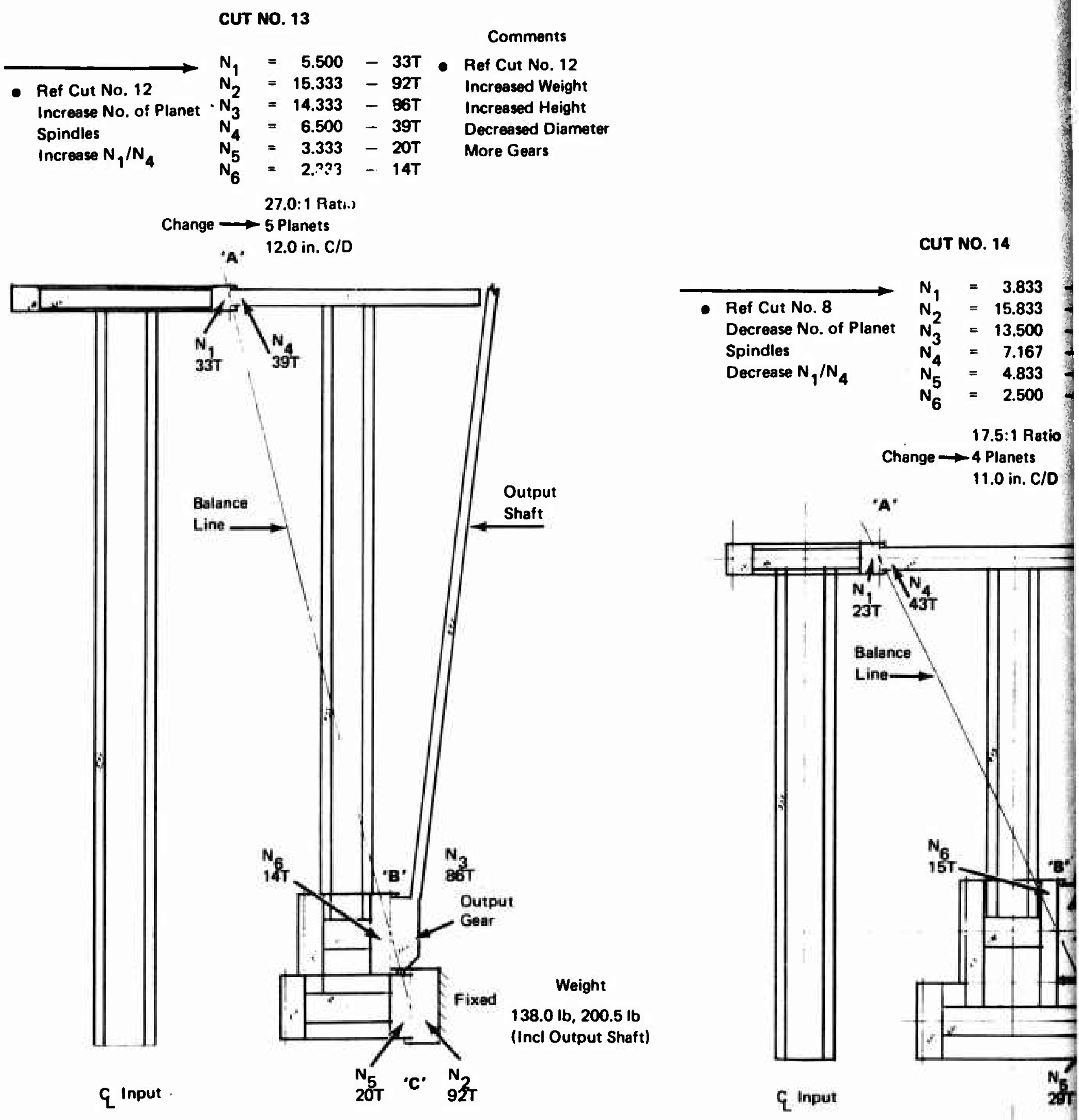


Figure 15. Free Planet Study Parametric Study (1,604 hp, 372 rpm).

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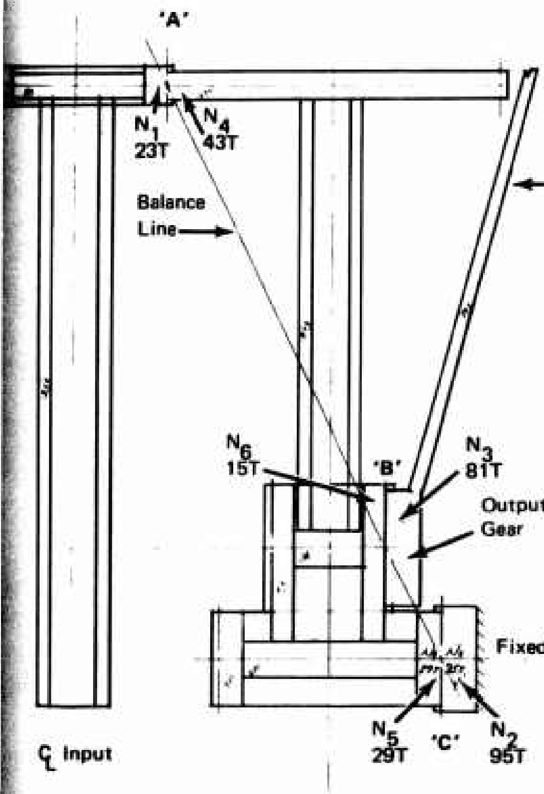
Cut No. 8
crease No. of Planet
ndles
crease N_1/N_4

CUT NO. 14

N_1 = 3.833 – 23T
N_2 = 15.833 – 95T
N_3 = 13.500 – 81T
N_4 = 7.167 – 43T
N_5 = 4.833 – 29T
N_6 = 2.500 – 15T

Comments
- Fewer Gears
  Decreased Height
  Increased Diameter

17.5:1 Ratio
Change → 4 Planets
11.0 in. C/D



Weight
125.8 lb, 159 lb
(Incl Output Shaft)

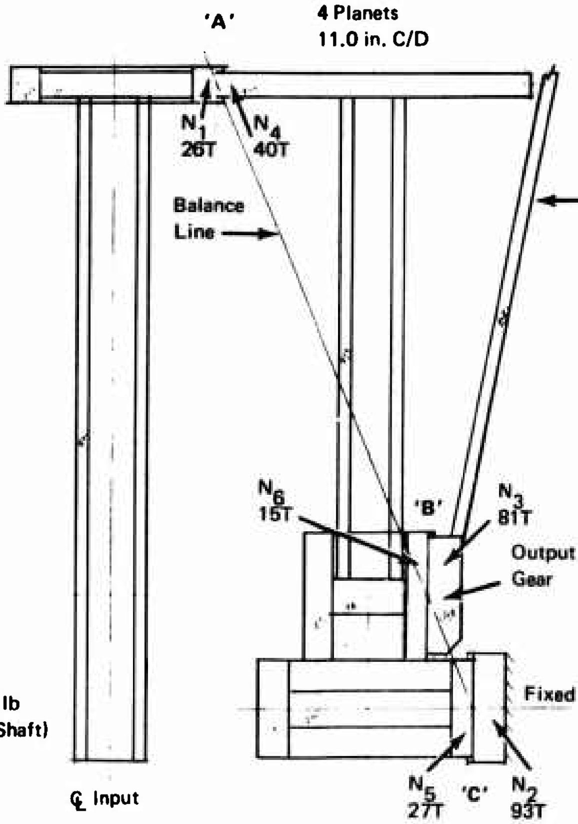
CUT NO. 15

- Increase N_1/N_4

N_1 = 4.333 – 26T
N_2 = 15.500 – 93T
Maintain N_3 = 13.500 – 81T
N_4 = 6.667 – 40T
N_5 = 4.500 – 27T
Maintain N_6 = 2.500 – 15T

Comments
- Increased Height
  Decreased Dia @ N_4

17.5:1 Ratio
4 Planets
11.0 in. C/D



Weight
122.0 lb, 158.3 lb
(Incl Output Shaft)

rpm).

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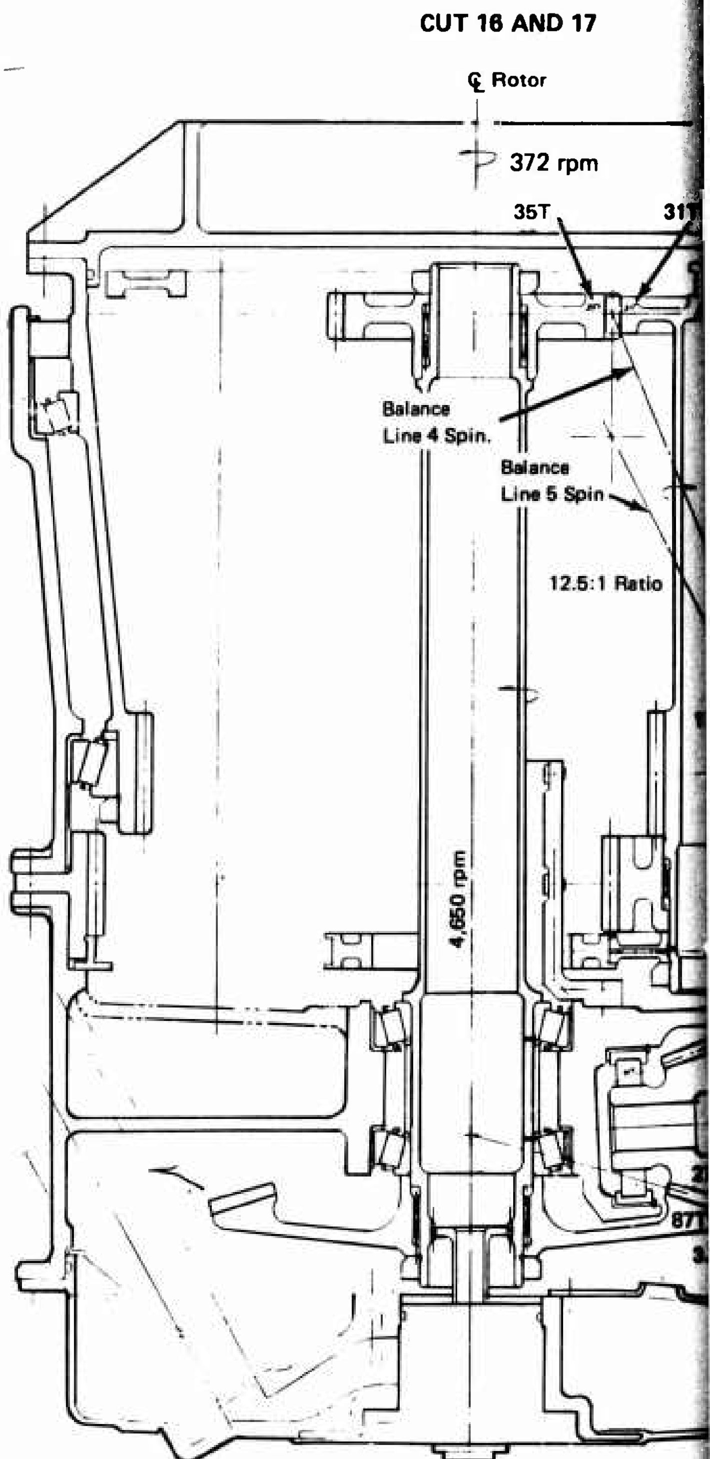
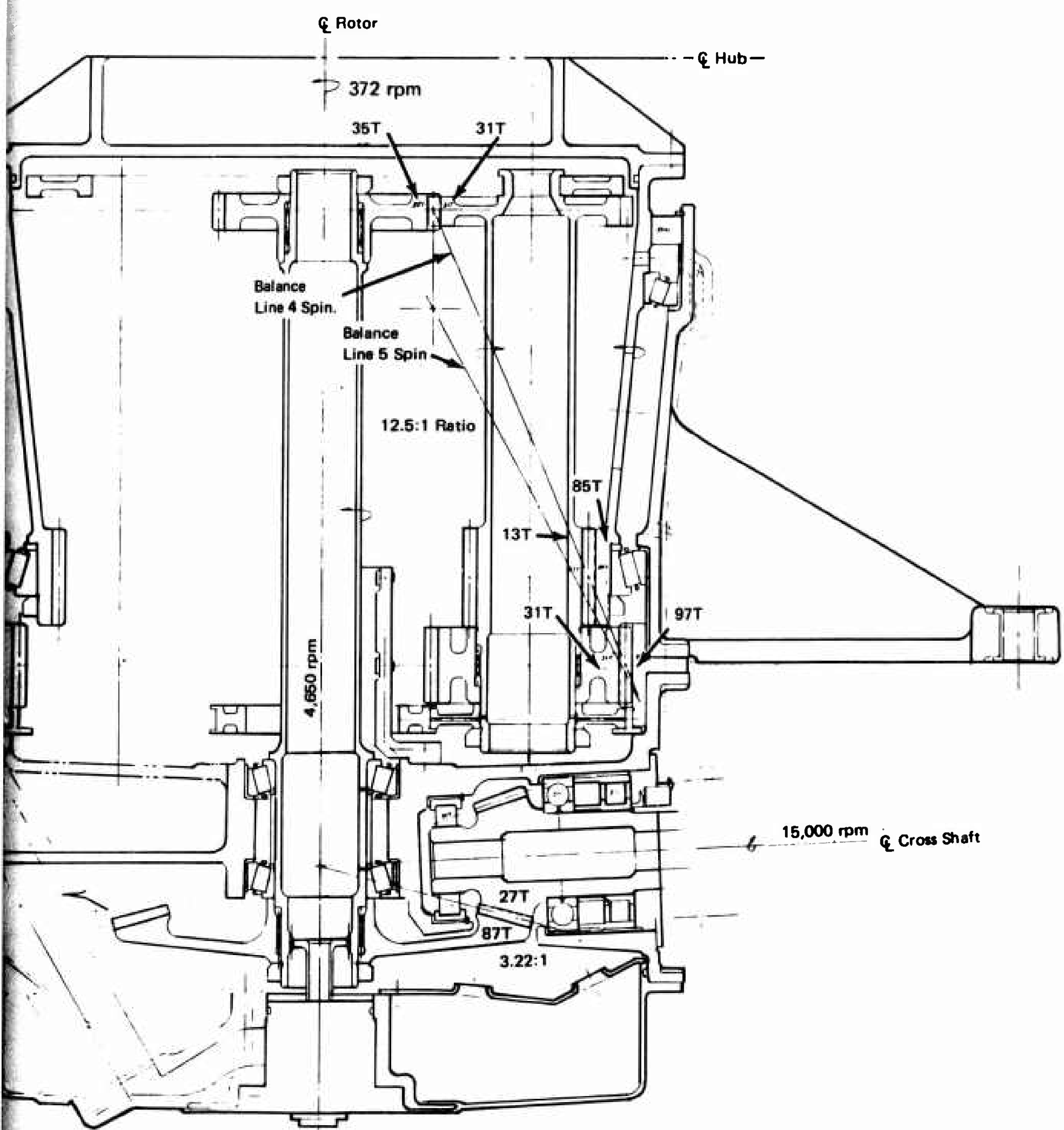


Figure 16. Free Planet Study as Applied to MUT 12.5 Planetary Ratio.

CUT 16 AND 17



12.5 Planetary Ratio.

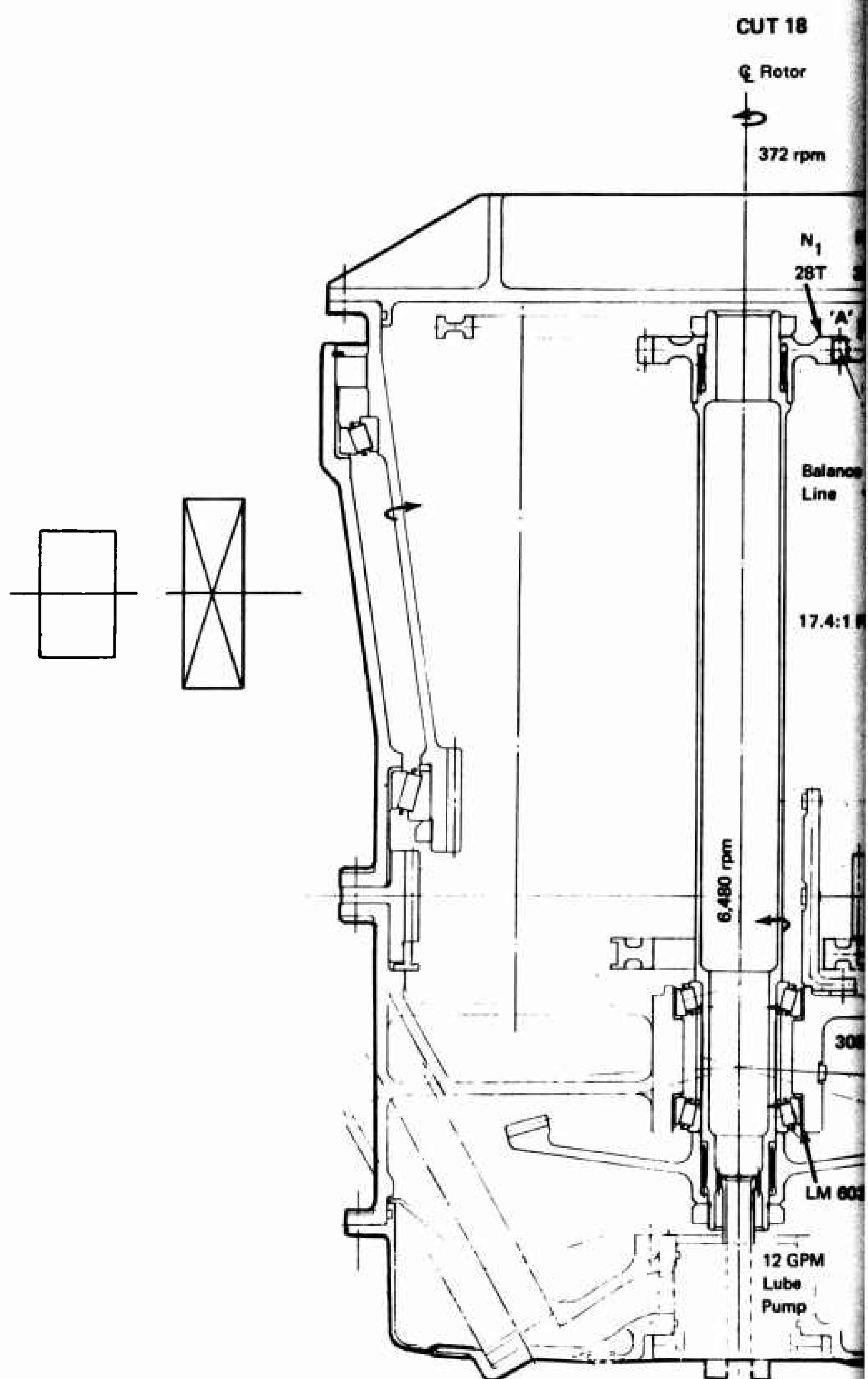
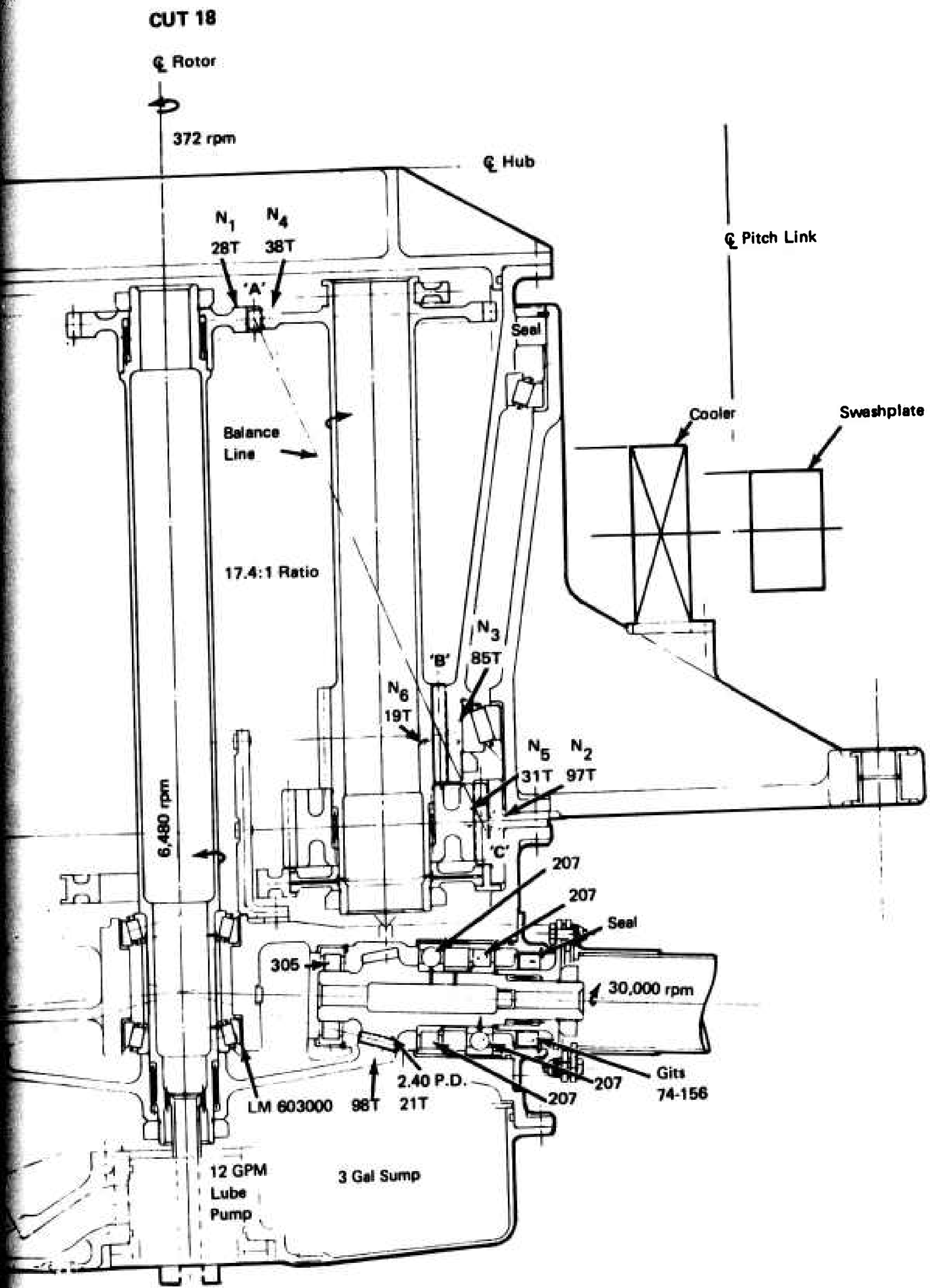


Figure 17. Free Planet Study as Applied to MUT 17.4 Planetary Ratio.


CUT 18
℄ Rotor
372 rpm
℄ Hub
℄ Pitch Link
N_1
28T
N_4
38T
'A'
Seal
Balance
Line
Cooler
Swashplate
17.4:1 Ratio
N_3
85T
'B'
N_6
19T
N_5
31T
N_2
97T
'C'
6,480 rpm
207
207
Seal
305
30,000 rpm
2.40 P.D.
21T
98T
LM 603000
207
207
Gits
74-156
12 GPM
Lube
Pump
3 Gal Sump

Ratio.

2

The Parametric Layout Study

The parametric layouts of Figures 11 through 17 were executed in order to determine the optimum relationship for diameter, height, number of planet spindles, and weight. Two ratios were investigated in some depth, with a third looked at briefly. Certain envelope restrictions were imposed for the arrangement study. These restrictions were dictated by the relationship of airframe structural elements and available space as determined in the structural configuration studies conducted for MUT.

The main rotor transmission was restricted to a structural compartment 30 inches wide (buttline measurement), 22.5 inches long (station measurement) and 28.5 inches high. A further restriction was that engine centerline location was established 19.5-inches down from the rotor hub centerline. A 20.0-inch dimension was allowed for "splayed engine" arrangement (configurations III and IV). In general, the main housing of the free planet area was restricted to a 20-inch diameter by a 15.5- to 16.5-inch height.

Some general design guidelines observed during the parametric study are as follows:

1. Meshes B and C (see Figure 18) have the greatest effect on the geometry of the free planetary since their relationship controls the slope of the "balance line" and thus the height of the gear arrangement.

2. An increase in the planet spindle center distance can reduce weight (cut 5 versus cut 6, Figure 12) or increase weight (cut 10 versus cut 11, Figure 14) depending on the optimization point for a given ratio requirement.

3. An increase in the number of planet spindles can decrease the weight (cut 1 versus cut 4, Figures 11 and 12) or increase the weight (cut 12 versus cut 13, Figures 14 and 15; cut 17 versus cut 18, Figures 16 and 17) depending on the point of optimization.

4. An increase in ring gear diameter generally results in reduced height and increased weight (cut 1 versus cuts 10 and 11, Figures 11 and 14).

5. For moderate changes in ratio, slight variations in mesh A (N_1/N_4) can affect the total ratio with the least configuration change.

6. Small changes in planet diameter have a significant effect on weight (cut 5 versus cut 6, Figure 12).

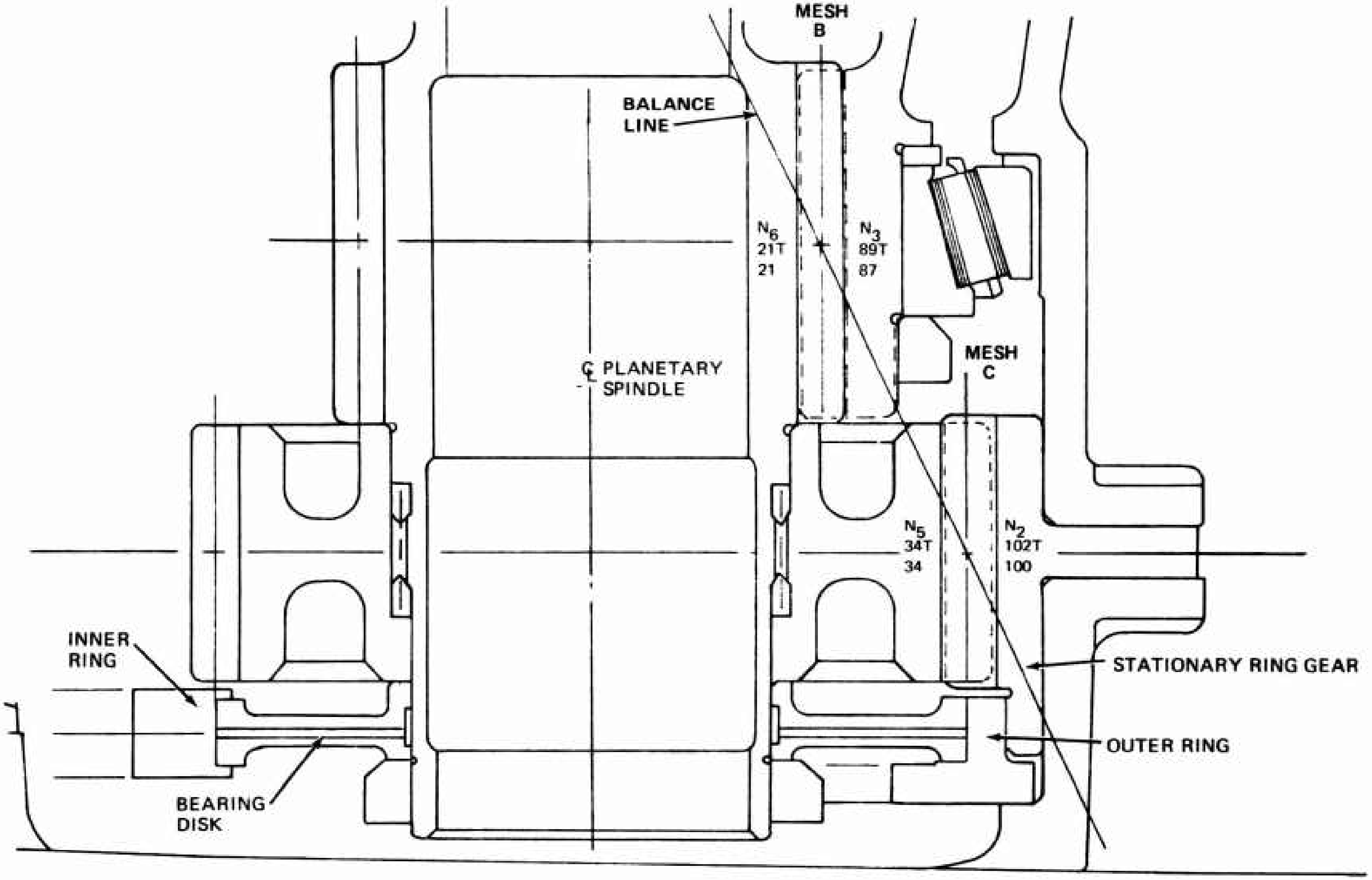


Figure 18. Meshes B and C of Free Planet Arrangement.

DISCUSSION OF THE SELECTED FREE PLANET DESIGN

The following factors determined the final selection of the design elements:

1. Arrangement and parts count - A variety of free planet arrangements are possible. The selection of configurations II and IV is based on the following considerations:

   a. The general configuration selected represents the design which received the most attention and hardware development by Curtiss-Wright.

   b. The presented design has the fewest parts and appears to be the easiest to produce.

   c. In the free planet concept the number of planet spindles significantly affects the height of the planetary arrangement. This is because the face width for meshes B and C determines the slope of the balance line (see Figure 18).

      For a given ring gear diameter and center distance, a greater number of planet spindles permits smaller face widths and, thus, lowers the height of the planetary by virtue of the lower inclination of the balance line. However, cost and reliability drive the design toward fewer planet spindles.

   d. The selected arrangement for this study was a four-spindle design. For comparison, configuration II also shows the balance line for a five-spindle arrangement. The five-spindle arrangement offers no weight advantage over the four-spindle arrangement shown, and the 2-1/2-inch height reduction does not justify its selection.

      Configuration IV is also a four-spindle arrangement since the higher reduction first stage does not permit a five-spindle arrangement because of insufficient space.

2. Weight - When reviewing the parametric study it was found that little weight difference existed between designs whose proportions of height and diameter were compatible to the configuration needs.

3. Rotor shaft support - It was apparent from initial layouts that the position of the free planet output shaft offered an opportunity to mount the rotor shaft bearings in a manner which would capitalize on the long output shaft length. (It may be desirable however to move the lower rotor shaft bearing up from directly under the output gear and to sacrifice shaft support for greater gear and bearing freedom.)

4. Planet support rings - One of the requirements of the free planet design is that support rings impose radial restraints to the separating and planetary centrifugal forces. An acceptable amount of ring deflection for the design shown would be about 0.001 inch. A deflection this small is not easily achieved without the addition of weight to provide stiffening for the rings. An even number of planets minimizes support ring deflection. Also, minimizing of support ring diameter helps reduce deflection. Because of these support ring considerations, a large bearing disc diameter was chosen to help offset the requirement for a large inner support ring at the ring gear mesh (see Figure 18). This reduced the diameter of the inner ring and made use of the stationary ring gear stiffness to augment outer ring stiffness requirements.

   Curtiss-Wright cautioned against forcing the planet spindle to ride on the pitch line of the stationary ring gear since this would restrict planet spindle freedom. It should be noted however that the design presented in configurations II and IV has a net force toward the center ring, and the intent of the design is to provide sufficient outer ring clearance to preclude spindle restraint. During the condition when applied torque is zero (helicopter autorotation), only centrifugal forces would be involved and, consequently, the outer ring engagement would take place. It was for this condition that the additional stiffness of the stationary ring was needed.

5. Simplified spindle support - The need for a rolling-element type bearing to support the planet spindle weight was eliminated using a concept previously demonstrated at Boeing Vertol. The concept provides a hardened rubbing strip to support the planet spindles. An oil catch ring approximately 0.005 inch below the rubbing strip traps the oil that is swept across the bearing surface by the progressive rotation of the planetary gears.

DESIGN REQUIREMENTS

The following loads and stress levels governed the design:

Loads

Rotor horsepower	1,604
Rotor rpm	372
Rotor torque	272,000 in.-lb.
Rotor moment	145,000 in.-lb.
Rotor drag	2,160 lb.

Gear Material Stress Allowables

Spur gears:	
Compression	180,000 psi
Bending	42,000 psi
Spiral bevel gears:	
Compression	260,000 psi
Bending	40,000 psi

TECHNICAL SUPPORT

The gear analysis for this program was conducted using Boeing Vertol's computer-aided gear design/analysis system. This computer program is generally consistent with the applicable AGMA standards; however, several factors not considered in these standards were also treated, including the following:

Specific sliding (slip ratio)

Sliding velocity

Entraining velocity

EHD lubricant film thickness

The bearing analysis uses a computer program which calculates the effects of speed, load distribution, and other factors on bearing life and performance. The analysis is based on the support structure being considered as part of the system, which is necessary since the bearings on a shaft are mechanically coupled. Through iterative techniques, this analysis accurately predicts fatigue life and performance.

Stress analysis was also conducted for the various elements of the free planet design. Samples of gear, bearing, and stress calculations are presented in Appendix A.

EVALUATION OF THE FREE PLANET VERSUS THE CONVENTIONAL TRANSMISSION

In order to evaluate and assess the free planet concept, the following areas were investigated:

Weight and efficiency

Cost and producibility

Configuration compatibility

Maintainability/reliability

Vulnerability/survivability

1. Weight and efficiency - The layouts of configurations II and IV were used to calculate the free planetary weight. A semianalytical method was used to determine the weights of the conventional planetary transmission. The method calculates the weight of each section or stage, using the following parameters:

   Surface compressive stress

   Gear ratio

   Support or combining factor

Special features factor

Design horsepower

Design rpm

The method was derived from more than 30 actual aircraft transmissions, and is generally accurate to within 5 percent. To a large extent the accuracy depends on the judgment exercised on the choice of the combining factor for integrating the various drive elements. For this exercise, the derived weights are based on the Boeing Vertol UTTAS transmissions using the combining value from the UTTAS gears. Table 2 summarizes the drive statistics and weight for each configuration.

The efficiency rating for each configuration is considered equal, although some benefit would be derived from elimination of the engine boxes for configurations III and IV.

Curtiss-Wright measured efficiency in the FP 500 and FP 501 free planet test program using both dynamometer and heat rejection measurements. Measured free planet efficiencies of 98.6 percent compare favorably with the efficiencies for the conventional transmission compared in this study.

2. <u>Cost and producibility</u> - In considering the differences in dollar cost and the ability to manufacture or produce the free planetary gearing, it is necessary to determine if any particularly critical processes are required for free planetary components (in comparison to conventional transmission components). During our investigation, no unusual manufacturing processes were identified which would be different from those required to produce the conventional transmission. The only difference from conventional Boeing Vertol planetaries is the tooling required to index the three planet gears mounted on a common spindle. This imposes no unusual requirement since the indexing requirement of 0.003 inch, which was determined to be satisfactory, is similar to compound planetaries.

Since no producibility penalty occurs for the free planet transmission, the cost was compared on a part-for-part basis. Using the current CH-47 production costs as a guide, a cost index factor was applied to each part. Table 3 shows the part count and cost index value. The cost index comparison shows the free planet design having a higher cost than the single-stage planetary of configuration I and a lower cost than the two-stage planetary of configuration III.

Table 2. Drive System Weight, Distribution and Ratios

	ENGINE BOX	ROTOR BEVEL	PLANETARY 1ST	PLANETARY 2ND	FWD AGB	TAIL T.O.	ALIGN SET	ANGLE ENGINE ROTOR	ANGLE AGB ROTOR	ANGLE OF ROTOR	ROTOR SHAFT & MOUNT	ENGINE SHAFTS	TOTAL WEIGHT ROTOR BOX & ALIGNMENT GEARS
CONFIG I	2.7:1 30,000 RPM 76 LBS	5.4:1 11,000 RPM 89 LBS	5.5:1 2,037 RPM 175 LBS	–	3.4:1 6,930 RPM 16 LBS	2.3:1 4,680 RPM 34 LBS	1.48:1 6,925 RPM 33 LBS	80°	86°	94°	121 LBS	15 LBS	559 LBS
CONFIG II	2.0:1 30,000 RPM 69 LBS	3.24:1 15,000 RPM 58 LBS	12.9:1 4,623 RPM 275* LBS	–	3.25:1 15,000 RPM 9 LBS	2.23:1 10,300 RPM 18 LBS	1.49:1 10,300 RPM 30 LBS	86°	86°	94°	** 24 LBS	14 LBS	497 LBS
CONFIG IIa (5 PLANETS)	2.0:1 30,000 RPM 69 LBS	3.24:1 15,000 RPM 58 LBS	12.9:1 4,623 RPM 307* LBS	–	3.25:1 15,000 RPM 9 LBS	3.23:1 10,300 RPM 18 LBS	1.49:1 10,300 RPM 30 LBS	86°	86°	94°	** 24 LBS	14 LBS	529 LBS
CONFIG III	20 LBS (CLUTCH)	4.69:1 30,000 RPM 38 LBS	4.70:1 6,400 RPM 51 LBS	3.66:1 1,362 RPM 144 LBS	1.5:1 45,000 RPM 9 LBS	3.5:1 22,500 RPM 9 LBS	3.2:1 22,500 RPM 21 LBS	90°	86°	94°	121 LBS	15 LBS	428 LBS
CONFIG IV	20 LBS (CLUTCH)	4.69:1 30,000 RPM 38 LBS	17.2:1 6,400 RPM 295* LBS *INCL. ROTOR SHAFT	–	6.22:1 27,450 RPM 5 LBS	3.3:1 21,000 RPM 9 LBS	4.101:1 21,000 RPM 19 LBS	9[illegible]	93°	94°	** 24 LBS **MOUNTING ARMS ONLY	15 LBS	425 LBS

TABLE 3. PART COUNT AND COST INDEX

PART	CONFIGURATION I (SINGLE-STAGE PLANETARY)		FREE PLANET		CONFIGURATION III (CH-47 TYPE)	
	PART COUNT	COST INDEX	PART COUNT	COST INDEX	PART COUNT	COST INDEX
Gears						
Bevel	3	15	3	15	3	15
Spur	5	5	13	13	12	12
Ring	1	8	2	18	2	10
Bearings	14	10	8	6	21	15
Carrier	1	10	0	-	2	-
Rings and Disks	0	-	7	4	0	-
TOTAL	24	48	33	56	40	72
Rating		10		8		7

3. Configuration compatibility - Since the free planet concept as configured for this application is much higher or taller than its conventional counterparts, the question of its suitability for the MUT application was therefore in question. However, we can compensate for the added length by integrating the hub into the rotor box. In configurations II and IV the rotor hub mounts directly on top of the gear box, thus eliminating the rotor shaft extension. The swashplate assembly and its associated linkages surround the main housing of the rotor box.

   The suitability of this configuration to MUT airframe compatibility was evaluated by the configuration designer. As follows, a rating of 1 to 10 was given as a measure of how well the various configurations suited the MUT airframe design concept.

   - Configuration I Rating: 6 (see Figure 2)

     1. Low position of input pinions base (relative to configuration II) reduces access to drive shafting.
     2. Oil tank position complicates access for transmission removal and forces bigger offset of forward structure lateral beam relative to forward transmission bolt hole.

   - Configuration II Rating: 10 (see Figure 3)

     1. The position of the input gear resulting from hub integration permits easier access to shafting.
     2. Parallel engine placement gives minimal interference with primary structure (buttline) BL 15 deck beams.
     3. Engine cross shafts and drive shafts to tail rotor and forward and aft AGB boxes conveniently fit into the structural arrangement, and also afford a balanced four-point actuator system which avoids asymmetrical loading on structure.

   - Configuration III Rating: 1 (see Figure 4)

     1. Angled engine arrangement must be a minimum of 37 degrees (drive shaft relative to aircraft centerline) for engine not to interfere with primary BL 15 deck beams.

2. Engines protrude from each side of the fuselage contour. (Increases drag count; complicates fairings; produces poor appearance.)

3. Low position of transmission base prevents use of wet sump configuration.

4. Oil tank position complicates access to remove transmission and forces bigger offset of forward structural lateral beam relative to forward transmission bolt hole.

5. This configuration would require major structural modifications (yielding inefficient load paths). Low parallel engine placement causes deck and beam lines to be lowered.

   If engines are tilted upwards at rear, as shown on View A-A of Figure 4, they will interfere with rotor blades.

6. Clutch assembly is located in such a position that it lies within the clearance hole in BL 15 beam (difficult access).

7. Three-point actuator system gives high load on rear single actuator, which is not on aircraft centerline.

- Configuration IV Rating: 3 (see Figure 5)

1. Angled engine arrangement must be a minimum of 37 degrees (drive shaft relative to aircraft centerline) for engine not to interfere with primary BL 15 deck beams.

2. Engines protrude from each side of fuselage contour. (Increases drag count; complicates fairings; produces poor appearance.)

3. Clutch assembly is located in such a position that it lies within the clearance hole in BL 15 beam (difficult access).

4. Three-point actuator system gives high load on rear single actuator, which is not on aircraft centerline.

4. Maintainability/Reliability Evaluation - For this study, a reliability index has been defined which addresses only the removal rate of the various configurations, but which has been qualitatively modified for such considerations as failure detectability, progression rate, and failure consequence.

a. Reliability critical characteristics - Table 4 identifies the most significant variables among the various candidate drive system designs. These parameters have been listed in terms of decreasing importance from a reliability point of view.

In order to calculate a reliability index, it is necessary to quantify failure rate variations due to a different number of gears and bearings in the various configurations.

Historically, bearings have caused twice as many transmission/gearbox removals as gears (Reference Table 38 of USAAMRDL TR 73-58). Furthermore, in planetary transmissions this ratio runs as high as 3-5.5 to 1, as shown in Table 5.

However, since there are usually about 1.5 times as many bearings in a planetary transmission as there are gears, one would expect more bearing failures than gear failures. In this study a factor of 2 (3 divided by 1.5) will be employed for gear reliability (i.e., on a piece-count basis, each gear is twice as reliable as each bearing).

The following reliability index results if configuration I is set as a baseline with a reliability equal to 100 (and only bearings and gears are considered).

Configuration	Bearing/Gear Reliability Index
I	100
II	103
III	62
IV	103

(Note: The higher the number the greater the reliability.)

The redistribution of bearings and gear quantities in the free planet design of configuration II almost counterbalances configuration I from a reliability point of view.

The higher planetary gear reduction ratios and the higher speeds of configurations II through IV make them a higher reliability risk than configuration I. However, since it is difficult to

Table 4. Reliability Significant Factors

FACTOR	I SCALED DOWN UTTAS	II FREE PLANET	III 2 STAGE PLANETARY DIRECT ENG INPUT	IV FREE PLANET DIRECT ENG INPUT
FREE PLANET	NO	YES	NO	YES
NUMBER BEARINGS	14	10	21	10
NUMBER GEARS	9	18	17	18
ENGINE TRANSMISSION	YES	YES	NO	NO
PLANETARY REDUCTION RATIO	5.5:1	12.9:1	17.2:1	17.2:1
CLUTCH RUNNING SPEED RPM	11,000	15,000	30,000	30,000
TYPE SUMP	DRY	WET	DRY	WET

Table 5. CH-47 Failure Distribution by Component Class and Transmission Type

	ENGINE TRANSMISSION		COMBINING TRANSMISSION		FORWARD TRANSMISSION		AFT TRANSMISSION		TOTALS	
	QTY	PCT	QTY	PCT	QTY	PCT	QTY	PCT	QTY	PCT
BEARINGS	422	16.83	123	13.53	1,504	43.46	1,309	39.81	3,358	33.03
GEARS	143	5.70	59	6.49	280	8.09	446	13.56	928	9.13
LUBE SYSTEM	147	5.86	208	22.88	276	7.97	202	6.14	833	8.19
RETENTION AND MOUNTING HARDWARE	1,127	44.97	444	48.84	1,294	37.39	1,072	32.57	3,937	38.73
NONROTATING STRUCTURE	77	3.07	18	1.98	62	1.79	133	4.04	290	2.85
SHAFTS	358	14.28	59	6.49	44	1.27	66	2.00	527	5.18
CLUTCHES	232	9.25	–	–	–	–	59	1.79	291	2.86
	2,506	99.26	911	100.21	3,460	99.97	3,287	99.91	10,164	99.97

quantify this risk, free planet configurations II and IV would be preferred from a main transmission bearing/gear reliability point of view.

Table 5 also shows that attachment hardware contributes to a considerable number of failures. Recent modification of the planet retaining hardware has greatly reduced this failure rate. Since the free planet design eliminates this type of planet retention hardware, we would anticipate even greater improvement in this area when the free planet concept is used.

In considering the reliability of drive system designs using direct engine input to the transmission (configurations III and IV), it is necessary to quantify engine transmission reliability, and stratify that reliability to isolate the clutch problem.

CH-47 clutch experience (refer to Tables 41 and 42 of USAAMRDL TR-73-58) indicates that elimination of the engine transmission, by means of direct engine input, provides a reduction in drive system failure rate. However, the exceedingly high clutch running speeds of configurations III and IV negate some of this anticipated benefit.

Using speed as a factor to modify the reliability predictions for configurations III and IV, the clutch modules yield the following reliability indices (based solely on the engine transmission/clutch modules, and again giving configuration I a baseline value of 100).

Configuration	Engine Xmsn/Clutch Module Reliability Index
I	100
II	95
III	170
IV	170

A reliability index must also be considered for the effect of providing a wet sump.

For CH-47 transmissions, about 3 percent of the removals have been generated by failures of items which would not have been critical with a wet sump. Thus, a reliability evaluation of the four configurations considering only the variable sump type, yields the following values:

Configuration	Type Sump Reliability Index
I	100
II	103
III	100
IV	103

Thus, combining the three reliability indices into one summing reliability index (and normalizing to a configuration I baseline of 100) yields the results summarized in Table 6.

TABLE 6. NUMERICAL RELIABILITY EVALUATION

Configuration	Bearing/ Gear Index	Engine Xmsn/ Clutch Module Index	Type Sump Index	Total Reliability Index
I	100	100	100	100
II	103	95	103	100
III	62	170	100	111
IV	103	170	103	125

b. Reliability qualitative considerations - Several areas of concern were identified while performing the reliability evaluation, which are difficult to quantify rigorously, but which at least warrant consideration. These factors are as follows:

(1) Higher accident potential of gear failure than bearing failure

(2) Free planet gear spindle failure modes

(3) Free planet support ring failure modes

The following paragraphs address these factors.

(1) Higher accident potential of gears - Although gears have historically been more reliable than bearings, they have, in general, exhibited more critical failure modes than bearings. Gear tooth breakage, web cracks, and flange cracks (often fretting induced) are several of the more common critical gear failure modes.

Thus, the free planet design which in essence replaces bearings with gears could present a slightly higher accident risk. However, this risk will be minimized if an adequate debris monitoring system and conservative bending stress allowables are employed.

(2) Free planet gear spindle failure modes - Through-the-part cracking of the planet gear spindle on the free planet design would be an undetectable failure mode. Thus, design criteria must minimize (via conservative stress allowables) the potential for this type of failure.

It has been suggested if a spindle crack were to occur, that planet would relieve itself by putting the other three gears in an overstress situation. This could cause tooth surface distress failures, which are detectable. A failure sequence such as this is possible if the broken planet could be restrained from jamming the assembly.

(3) Free planet support ring failure modes - Through-the-part cracks of this component would be undetectable, consequently, they must be precluded by conservative design practices and high quality control. Spalling on these rings, generated by high surface stresses, could form a through-failure origin if the condition were not detected.

5. Vulnerability/Survivability Evaluation - Table 7 summarizes the vulnerability/survivability study. The free planet configurations show a 10-percent improvement over the conventional type transmission. The actual improvement would become identifiable only through experience with free planet transmission hardware. It should be noted however that the long length of the sun gear shaft is not a benefit from a vulnerability standpoint.

A significant point of comparison, of course, is the elimination of the planet bearings in the free planet transmission. However, since our experience indicates that planet bearings are not as critical as other high-speed drive system bearings, no significant survivability benefits are apparent from eliminating these bearings. This would be especially true where drive systems incorporate a backup lubrication system.

Table 7. Vulnerability/Survivability Evaluation — Transmission Comparison

FACTOR	BASELINE	FREE PLANET	CRITICALITY FACTOR	VULNERABILITY FACTOR HIGHER NUMBER = MORE VULNERABLE	
				CONFIG I BASELINE	CONFIG II FREE PLANET
PRESENTED AREA (SQ IN)	300	360			
NONDUPLICATED AREA (SQ IN)	210	230	.05	10.5	11.5
ROTOR SHAFT BEARING AREA (SQ IN)	38	36	1.0	38	36
PLANET SPINDLES (SQ IN)	–	32	1.0	–	32
PLANET BEARINGS (SQ IN)	51	–	1.0	51	–
PLANET GEARS (SQ IN)	51 (S)	20 (S) 62 (US)	.15 (S) .25 (US)	7.5	18.5
TOTAL				107	98
TOTAL*				96.5	86.5

S = SHIELDED
US = UNSHIELDED

NOTE: Planet Bearing direct impact and lube loss figured into criticality factor.

*Corrected for systems having backup lube.

BOEING VERTOL ADVANCED CONCEPT TRANSMISSION

Boeing Vertol has conceived an advanced transmission design lighter and more producible than the baseline state-of-the-art configuration. While this design is considered proprietary at this time, features pertinent to the study comparison are outlined below.

General arrangement: Bevel pinion inputs combine on a bevel gear and flow through a single-stage planetary. The rotor hub support is fully integrated with the planetary. Rotor shaft and upper cover are eliminated. The transmission housing carries rotor loads to the airframe. Overall dimensions, in particular height, are significantly reduced as compared to the baseline; a comparison sized for UTTAS requirements is shown in Figure 19. The low height allows a gravity (wet) sump and simplified lubrication system.

Planetary: Ratio is the same as the baseline design (5.5:1). Gear elements are arranged so as to balance planet loads and so reduce bending and deflections in the planet posts and bearings. The effect will be to increase gear and bearing reliability and life and to reduce the weight of the planetary system. The planet gears are high-contact-ratio profile, providing increased load sharing and hence greater capacity.

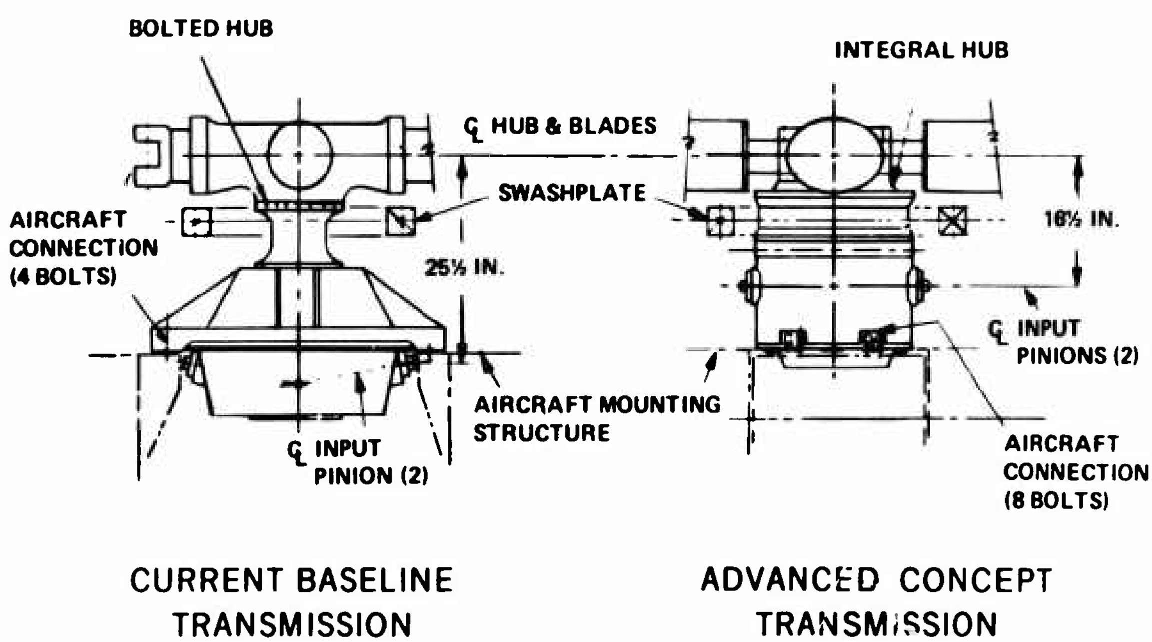


Figure 19. Main Transmission Configuration Comparison

CONCLUSIONS AND RECOMMENDATIONS

The summary evaluation (Table 8) indicates that free planet Configuration II achieved higher ratings in all areas except cost and producibility when compared to baseline system I. Configurations III and IV, providing high-ratio final drives in conventional and free planet arrangements, were discarded from the study because of basic problems of configuring the aircraft to accept the necessary engine placement.

When Configuration II is compared to the advanced transmission also using an integrated hub, the advanced transmission ranks higher. The reasons for this are that the advanced concept is lighter, has fewer gears, is more compact, and is thus equal or better in configuration compatibility.

The improvement potential for the free planet concept appears to lie in its ability to realize added load capacity, or added reliability, from improved load distribution. This possibility may overcome the space envelope which the design requirements dictate. The degree of improvement from better load sharing can only be understood after extensive testing of the free planet concept in a rotor transmission configuration. Risks associated with the free planet concept can be minimized by thoroughly evaluating existing hardware. Such an evaluation should include overload testing to make an initial assessment of improvement potential.

Table 8. Evaluation Index

	DRIVE WEIGHT*	PLANETARY WEIGHT	COST/ PRODUCIBILITY	CONFIGURATION COMPATIBILITY	RELIABILITY MAINTAINABILITY	SURVIVABILITY/ VULNERABILITY
CONFIG I (SINGLE-STAGE PLANETARY)	559 LB	175 LB	7	6	100	9
CONFIG II (FREE PLANET)	497 LB	275 LB	8	10	100	10
CONFIG III (2-STAGE PLANETARY)	428 LB	195 LB	10	1	111	
CONFIG IV (FREE PLANET)	425 LB	295 LB	8	3	125	–
ADVANCED TRANSMISSION	450 LB	170 LB	10+	10	100	10

*This weight does not include tail shafting, or accessory and tail gear boxes.

NOTE: The higher the index number the better the rating.

APPENDIX A

FREE PLANET STRESS, GEAR, AND BEARING CALCULATIONS

• FREE PLANET STRESS CALCULATIONS

$$\text{Ratio} = \frac{\left(1 + \dfrac{N_2N_4}{N_5N_1}\right)}{\left(1 - \dfrac{N_2N_6}{N_5N_3}\right)} = \frac{1 + \dfrac{100 \times 31}{34 \times 35}}{1 - \dfrac{100 \times 21}{34 \times 87}} = \frac{3.62}{0.29} = 12.4$$

DESIGN LOADS – TORQUE

HP = 1,604 @ 372 rpm

$$\text{Rotor Torque} = \frac{1{,}604 \times 63{,}000}{372} = 271{,}600 \text{ in.-lb}$$

$$\text{Input Shaft Torque} = \frac{271{,}600}{12.4 \times 3.15} = 6{,}940 \text{ in.-lb}$$

Input Shaft rpm = 372 x 12.42 x 3.15 = 14,550 rpm

DESIGN LOADS – ROTOR SYSTEM

Thrust = 10,300 lb @ 8 lb/sq ft = 1,287

$$\text{Rotor Diameter} = \sqrt{\frac{4}{\pi} \times 1{,}287} = 40.5 \text{ ft}$$

$$@\ 758 \text{ ft/sec } \omega = 37.4 \text{ rad/sec} \times \frac{60}{24} = 357 \text{ rpm}$$

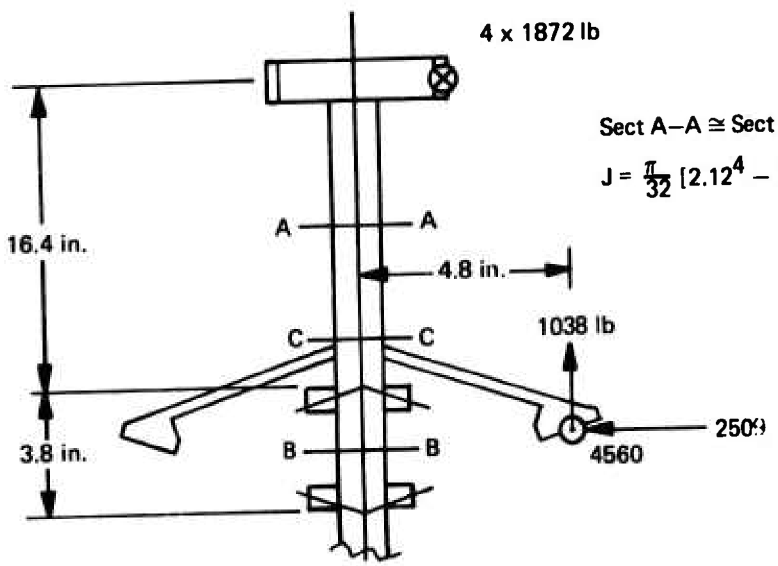
$$\text{Design Moment} = 260{,}000 \text{ in.-lb} \times \frac{40.5}{49} \times \frac{10{,}300}{15{,}300}$$

$$= 145{,}000 \text{ in-lb}$$

$$\text{Drag} \cong 1.5 \times \frac{8}{57} \times 10{,}300 = 2{,}160 \text{ lb}$$

- **SUN GEAR SHAFT**

Free Planetary



Sect A–A ≅ Sect B–B

$J = \frac{\pi}{32}\,[2.12^4 - 1.8^4] = .952$

$$\text{Shaft Torsion } f_s = \frac{Tc}{J} = \frac{21{,}870 \times 1.06}{.952} = 24{,}350 \pm 2{,}922 \times K_t$$

Low, Diameter Could Be Reduced to 1.8 in.

Note: This Modification was Made.

<u>CC</u>

Slope at Upper Bearing Defines Deflection at Sun Gear

Determine Force that Centers Sun

$$\theta_{\text{Upp. Brg.}} = \frac{NL}{3EI} = \frac{6{,}000 \times 3.8}{3 \times 29 \times 10^6 \times .48} = .0005 \text{ in./in.}$$

$$\delta_{\text{Sun}} = 16.4 \times .0005 = .0082 \text{ in.}$$

$$P_{\text{Sun}} = \frac{3EI\sigma}{L^3} = \frac{3 \times 29 \times 10^6 \times .48 \times .0082}{16.4^3} = 77.6 \text{ lb}$$

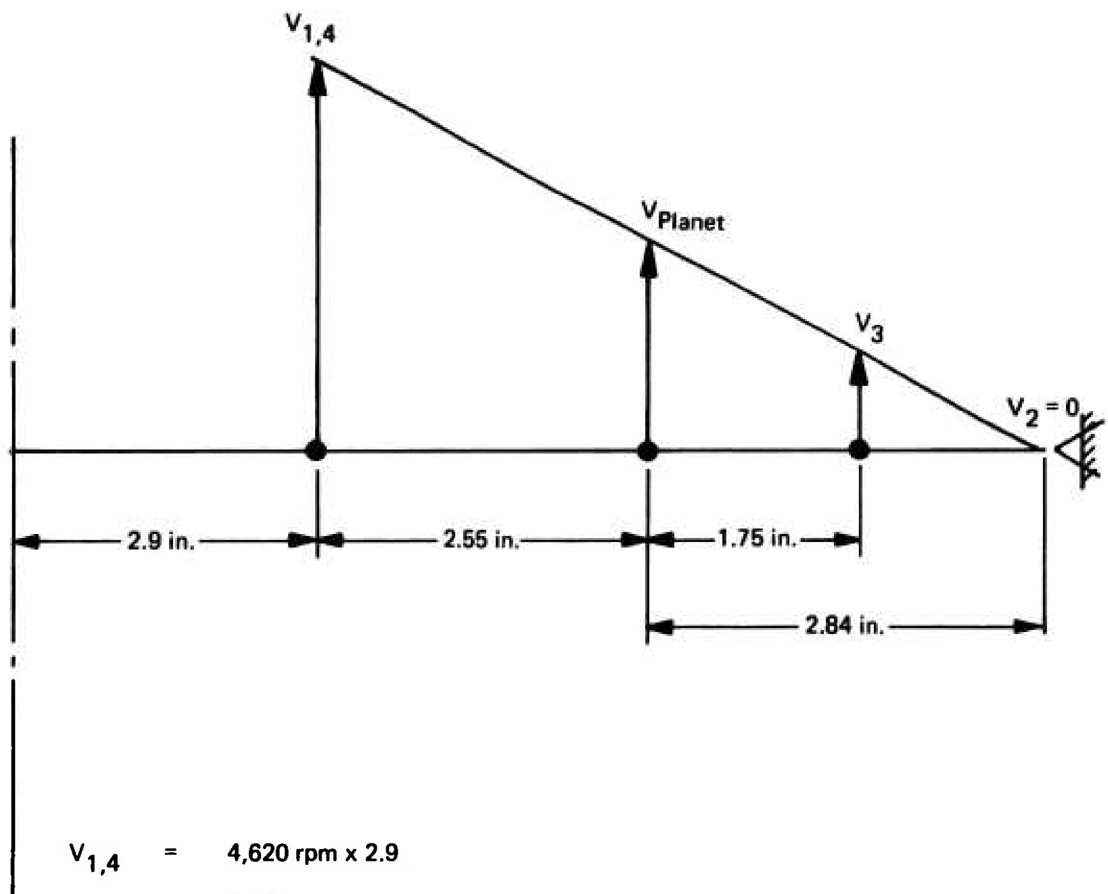
$$M_{CC} = 77.6 \times 16.4 = 1{,}272 \text{ in.-lb}$$

$$\text{Shaft Bending } f_b = \frac{Mc}{I} \quad \frac{1{,}272 \times 1.06}{.48} = \pm 2{,}809 \text{ psi}$$

Low, Diameter Could Be Reduced to 1.8 in.

Note: This Modification was Made.

- **PLANET GEAR KINEMATICS (RPM)**



$$V_{1,4} = 4{,}620 \text{ rpm} \times 2.9$$

$$V_{Planet} = \frac{2.84}{(2.55+2.84)} \times 4{,}629 \times 2.9 = 7{,}060$$

$$\text{Rpm} = \frac{V_{PLANET}}{(2.9+2.55)} = 1{,}380 \text{ rpm}$$

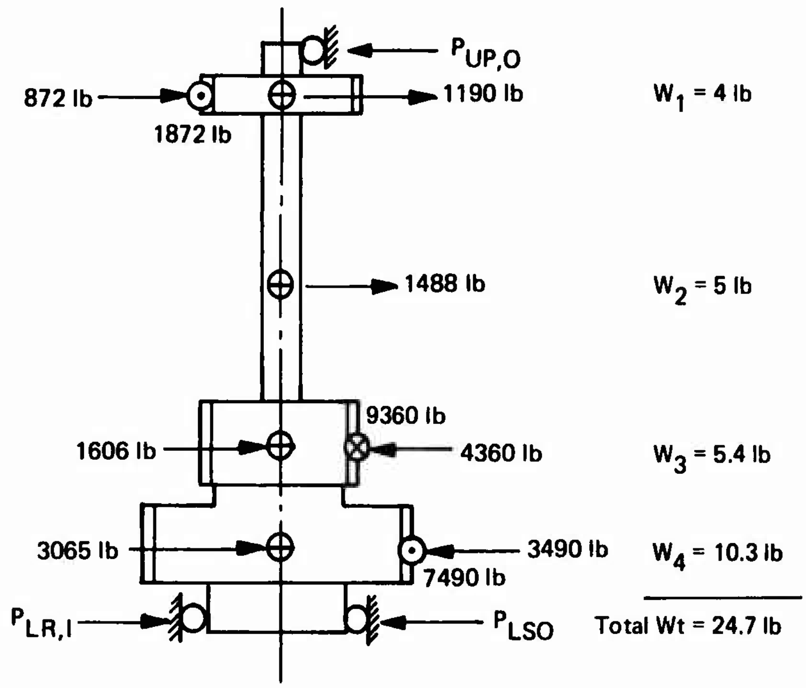
$$\text{CF Planet} = \frac{W}{386} \times R \times \omega^2 = \frac{W}{386} \times 5.5 \times \left(1{,}380 \times \frac{2\pi}{60}\right)^2$$

$$= 297.6\ W$$

$$\text{Rpm}_{Output} = \frac{V_3}{2.9+2.55+1.75} = \frac{\frac{2.84-1.75}{2.55+2.84}}{7.2} \quad V_1 = \frac{.202}{7.2} \times 4{,}620 \times 2.9$$

$$= 375 \text{ rpm}$$

- **PLANET GEAR SHAFT**



<u>RING LOADS DUE TO W_N</u>

$$P_{LR,I} = \frac{3{,}490 \times 12.6 + 4{,}360 \times 10.2 - 872 \times .6}{14} = 6{,}280 \text{ lb}$$

$$P_{UP,O} = 6{,}280 - 3{,}490 - 4{,}360 + 872 = -698 \text{ lb}$$

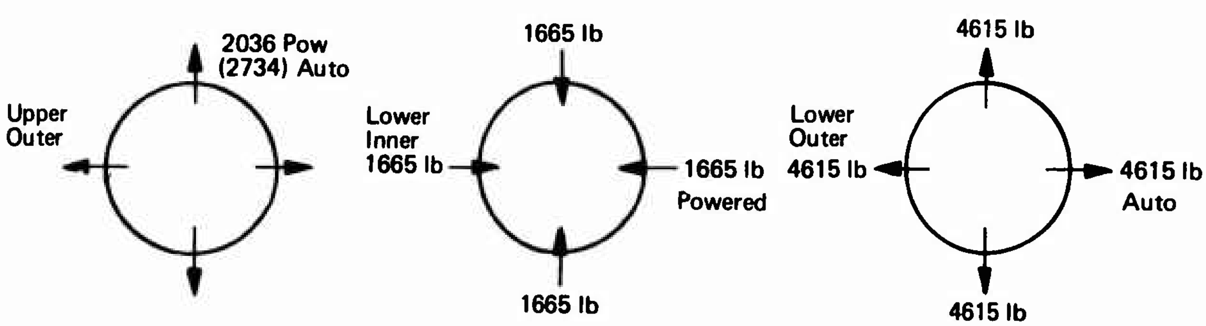
<u>RING LOADS DUE TO CF</u>

$$P_{UP,O} = \frac{1{,}190 \times 13.45 + 1{,}488 \times 8.4 + 1{,}606 \times 3.8 + 3{,}065 \times 1.2}{14}$$

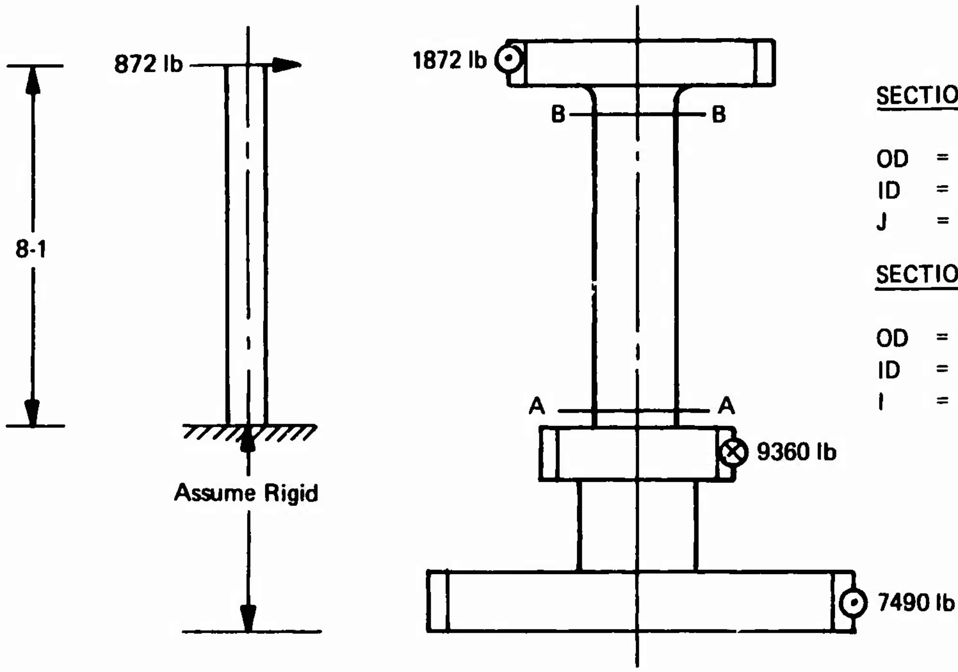
$$= 2{,}734 \text{ lb}$$

$$P_{LR,O} = 1{,}190 + 1{,}488 + 1{,}606 + 3{,}065 - 2{,}734 = 4{,}615 \text{ lb}$$

- **RING LOAD SUMMARY**



- **PLANET SHAFT**



SECTION BB

OD = 2.40 in.
ID = 2.15 in.
J = 1.16 in.

SECTION AA

OD = 2.93 in.
ID = 2.10 in.
I = 2.66 in.

PLANET SLOPE AT SUN MESH (DRIVE DIRECTION)

$$\theta = \frac{PL^2}{2EI} = \frac{1{,}872 \times 8.1^2}{2 \times 29 \times 10^6 \times 2.66} = .000796 \text{ in./in.}$$

Note: Crowing of sun gear teeth is an alternate to increased stiffness

SHAFT STRESS AT SECTION AA

$$f_b = \frac{Mc}{I} = \frac{(1{,}872 + 872) \times 8.1 \times 1.465 \times 1.5}{2.66} \pm 13{,}800 \text{ psi} \qquad (K_t = 1.5)$$

SHAFT TORSION STRESS AT SECTION BB

$$f_s = \frac{Tc}{J} \quad \frac{1{,}872 \times 2.6 \times 1.2}{1.16} = 5{,}035 \text{ psi}$$

- **GEAR STRESS CHECK**

Note: This poi tion of analysis was also checked on computer program.

SUN PLANET

$$f_t = \frac{W_t P_D}{F x Y_K} = \frac{1{,}872 \times 5}{.8 \times .45} = 26{,}000 \text{ psi}$$

PLANET-STATIONARY RING

$$f_t = \frac{W_t P_D}{F x Y_K} = \frac{7{,}490 \times 5}{2 \times .45} = 41{,}600 \text{ psi}$$

PLANET – OUTPUT RING

$$f_t = \frac{W_t P_D}{F x Y_k} = \frac{9{,}360 \times 5}{2.7 \times .45} = 38{,}518 \text{ psi}$$

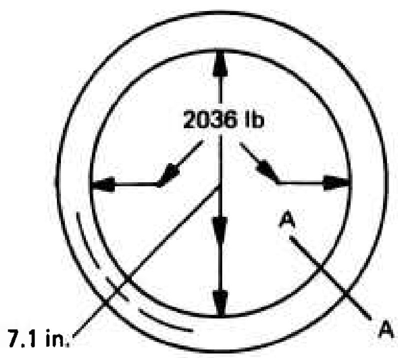
CONTACT STRESSES

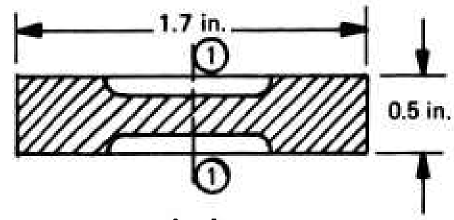
$$S_{C@\text{ Sun Planet}} = 3{,}180 \left(\frac{W_t}{F \text{ SIN } 2\phi} \times \frac{R_1 + R_2}{R_1 \; R_2}\right)^{1/2}$$

$$= 3{,}180 \left(\frac{1{,}872}{.8 \times .766} \times \frac{2.44 + 2.15}{2.44 \times 2.15}\right)^{1/2}$$

$$= 3{,}180 \quad (2{,}650)^{1/2} = 163{,}000 \text{ psi}$$

- **UPPER PLANET RING SUPPORT**





A–A

$$\text{Max Ring Moment} = (.3183 - .1817)\ WR$$

$$= .1366 \times 2{,}036 \times 7.1$$

$$= 1{,}975 \text{ in.-lb.}$$

$$I_{1\ 1} = \frac{1}{12} \times .5 \times 1.7^3 = .205 \quad I/C = \frac{.205}{.85} = .241$$

$$\text{Max Ring Stress} = \frac{Nc}{I} = \frac{1{,}975}{.241} = 8{,}200 \text{ psi}$$

Since Min Stress = −8,200 psi

Salt = ± 8,200 psi, No Problem (< 25,000 psi)

CHECK OF RING CF STRESS

$$\omega^2 = \left(1{,}380 \times \frac{2\pi}{60}\right)^2$$

$$\text{CF Load} = MR\omega^2 = \frac{\pi \times 7.1 \times .85 \text{ Area} \times .3}{386} \times .4 \times 7.1 \times 20{,}884$$

$$= .0147 \times 2.85 \times 20{,}884 = 872 \text{ lb}$$

$$f_t = \frac{872}{2 \times .85} = 513 \text{ psi}$$

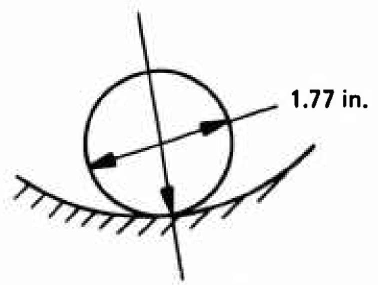
- **UPPER PLANET RING SUPPORT**

RING DEFLECTIONS

$$\delta = (.149 - .137)\ \frac{WR^3}{EI}$$

$$= \frac{.012 \times 2{,}036 \times 7.1^3}{29 \times 10^6 \times .201} = .0015 \text{ in.}$$

CONTACT STRESSES (Hertz)



t = 0.5 in.

$$P = 2{,}036 \text{ lb} \qquad p = \frac{2{,}036}{.5} = 4{,}072$$

$$Sc = .591 \sqrt{pE\ \frac{D_1 - D_2}{D_1 \times D_2}}$$

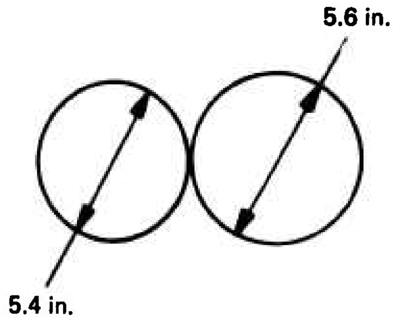
$$= .591 \times \sqrt{4{,}072 \times 29 \times 10^6 \times \frac{(12.6 - 1.77)}{12.6 \times 1.77}}$$

$$= .591 \times 239 \times 10^3 = 141{,}500 \text{ psi}$$

Comparable to Curtiss-Wright

- **LOWER PLANET RING SUPPORTS**

<u>INNER RING</u>

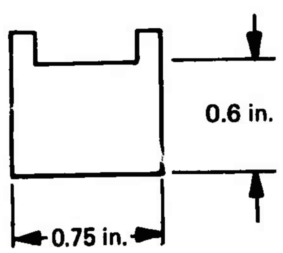


$$P = 1{,}665 \text{ lb} \qquad P = \frac{1{,}665}{.5} = 3{,}330 \text{ lb/in.}$$

$$Sc = .591 \sqrt{\frac{3{,}330 \times 29 \times 10^6 \times (5.4 + 5.6)}{5.4 \times 5.6}}$$

$$= 110{,}000 \text{ psi}$$

<u>INNER RING DEFLECTIONS</u>



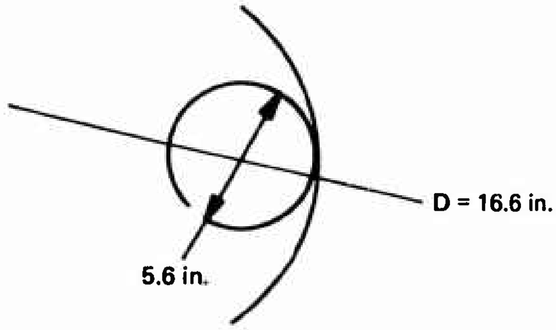
$R = 2.4$ in.

$I = \frac{1}{12} \times .75 \times .6^3 = .0135$

$$\delta = \frac{.012WR^3}{EI} = \frac{.012 \times 1{,}665 \times 2.4^3}{29 \times 10^6 \times .0135}$$

$$= .000705 \text{ in.}$$

OUTER RING

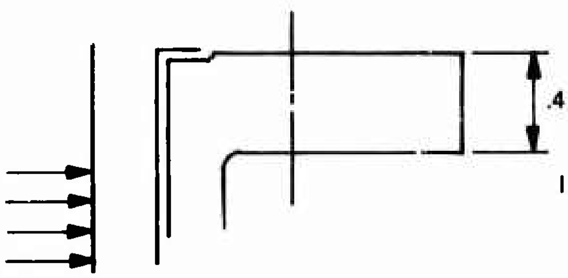


$P = 4{,}615 \text{ lb}^{*} \qquad p = \dfrac{4{,}615}{.5} = 9{,}230 \text{ lb/in.}$

$$Sc = .591 \sqrt{\frac{9{,}230 \times 29 \times 10^6 \times (16.6 - 5.6)}{16.6 \times 5.6}}$$

$$= .591 \times 178 \times 10^3 = 105{,}180 \text{ psi}$$

OUTER RING DEFLECTIONS



$$I_{Flange} = \frac{1}{12} \times .4 \times 1.6^3$$

$$= .136$$

$$\delta = \frac{.012 \times 4{,}615 \times 9.6^3}{30 \times 10^6 \times .136} = .012 \text{ in.}$$

No

*Auto Rotation Only

- **LOWER PLANET RING SUPPORTS**

<u>OUTER RING STRESSES</u>

$$M = .136 \times 4{,}615 \times 9.6 = 6{,}051 \text{ in.-lb}$$

$$f_b = \frac{Mc}{I} = \frac{6{,}051 \times .8}{.136} = 35{,}600 \text{ psi}$$

Note: Design was modified to provide greater outer ring support stiffness.

LIST OF ABBREVIATIONS AND SYMBOLS

AGB	accessory gearbox
f_b	bending stress
f_s	shear stress
f_t	tensile stress
hp	horsepower
rpm	revolutions per minute
S_C	compressive stress
xmsn	transmission
δ	deflection
θ	slope
ω	angular velocity